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AMC PAMPHLET AMCP 706-356

ENGINEERING DESIGN HANDBOOK

AUTOMOTIVE SERIES

AUTOMOTIVE SUSPENSIONS

HEADQUARTERS, U.S. ARMY MATERIEL COMMAND APRIL 1967

HEADQUARTERS
UNITED STATES ARMY MATERIEL COMMAND
WASHINGTON, D. C. 20315

AMC PAMPHLET
No. 706-356

14 April 1967

ENGINEERING DESIGN HANDBOOK

AUTOMOTIVE SERIES
AUTOMOTIVE SUSPENSIONS

This pamphlet is published for the information and guidance of all concerned.

(AMCRD-R)

FOR THE COMMANDER:

K. H. BAYER
Major General, USA
Acting Chief of Staff

OFFICIAL:

STANLEY J. SAWICKI
Colonel, GS
Chief, Administrative Office

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PREFACE

The Engineering Design Handbook Series of the Army Materiel Command is a coordinated series of handbooks containing basic information and fundamental data useful in the design and development of Army materiel and systems. The handbooks are authoritative reference books of practical information and quantitative facts helpful in the design and development of Army materiel so that it will meet the tactical and the technical needs of the Armed Forces.

This handbook is one of the Automotive Series of the Engineering Design Handbook Series. The Automotive Series covers the broad field of automotive equipment, with particular emphasis on the physical, operational, and climatic requirements imposed by the military environment. Combat and tactical land vehicles of all types—wheeled, tracked, and others—are included in this series.

The purpose of these handbooks is to provide assistance to designers of military automotive equipment and technical guidance to military and civilian personnel who are responsible for preparing specifications for this equipment, or who are responsible for ensuring their fulfillment.

The suspension system, one of the major functional elements of the automotive assembly, is the subject of this handbook. Its treatment is divided into three major categories, namely, Part One, *Fundamental Discussions,* Part Two, *Land Locomotion*, and Part Three, *Suspension Systems, Components, and Design Data.* The material presented is a compilation of data and design information gathered from many sources. Because of the vast scope of this subject, its treatment has been limited to one of condensation and summary. Proofs and mathematical deviations are omitted, for the most part. Only final results, qualifying information, and interpretation useful to the design process are given. For a more complete treatment of any topic, the reader is urged to consult references listed at the end of each chapter.

When reference is made in this handbook to military specifications, regulations, or other official directives, it is done so to inform the reader of the existence of these documents. In this respect, the user is cautioned to make certain that he uses editions which are current at the time of use.

This handbook was prepared by the Military Systems Group, Machine Design Section, Mechanical Engineering Research Division* of the IIT Research Institute, assisted by the Soil Mechanics Section of the Solid Mechanics Division, for the Engineering Handbook Office of Duke University, prime contractor to the Army Research Office-Durham.

The authors wish to acknowledge the excellent cooperation rendered by the U. S. Army Tank-Automotive Command and the Development and Proof Services of Aberdeen Proving Ground (now of

* Rudolph J. Zastera, Project Leader and Technical Editor.

the Test and Evaluation Command) in providing reports, data, and other information beneficial to the preparation of this handbook.

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Comments and suggestions on this handbook are welcome and should be addressed to Army Research Office-Durham, Box CM, Duke Station, Durham, North Carolina 27706.

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PART ONE
FUNDAMENTAL DISCUSSIONS

CHAPTER 1

INTRODUCTION*

SECTION I GENERAL DISCUSSION

Military tactical doctrine has always stressed the importance of mobility—particularly off-the-road-mobility, soft ground mobility, snow mobility, and high-speed-over-rough-terrain mobility. Obviously, if we can operate effectively over terrain that our enemy cannot negotiate—or move with greater speed, or maneuver faster over the same terrain—we will have a decided advantage over him. We can evade his attacks at will by rapid dispersion or by making use of terrain that is denied him; we can execute lightning counterattacks from unexpected directions before he has time to regroup; and we can exploit our successes with rapid follow-up tactics

Early armies moved and fought primarily on foot. Most terrain could be negotiated satisfactorily by the foot soldier although his load-carrying capabilities, endurance, and speed were quite limited. The use of animals improved his load-carrying capabilities and endurance but had little effect upon his overall rate of march. Flashing cavalry charges were made possible, however, by the short distance speed capabilities of horses. With the advent of motorized vehicles, mechanized transport replaced animal transport—first only in logistic operations but later in reconnaissance and combat operations as well.

When first introduced, military vehicles were merely adaptations of the then current commercial vehicles. Demands by the using public for improved riding comfort and greater speed led to the improvement of roads and to increased vehicle performance, but vehicle performance was evaluated in the relatively ideal environment of the highway. As the development of roads and vehicles progressed simultaneously, vehicle performance became increasingly dependent upon good roads; and the suitability of commercial vehicles for military operations decreased. In unfavorable terrain, military vehicles became roadbound, making them vulnerable to enemy interdiction and increasing the total time spent by a march column on the road.

The nuclear battlefield imposes even greater requirements for increased mobility, particularly a high degree of cross-country mobility and high off-road speed. The tactical concepts envisioned by military planners calls for an extreme dispersion of troops. Task force groups, comprised of widely scattered units to minimize the possibilities of nuclear counterattack, will be required to mass rapidly on a given target for a concerted strike and separate as rapidly as possible to their dispersed positions. Thus, if the area a unit is required to hold is materially increased due to the dispersion necessary, the requirements for logistic support and communication will also increase. These increased requirements can only be met by vehicles capable of much higher speeds and a high degree of off-road mobility.

The fullfillment of these demands for vastly improved mobility depends upon the vehicles' suspension systems. Engines and power trains develop the required propulsive efforts; frames,

* Written by Rudolph J. Zastera of the IIT Research Institute, Chicago, Illinois.

bodies, and hulls provide the required support and housing for components, cargoes, and crews; but the suspension systems must act as the feet and legs of the vehicles.

As feet and legs, the suspension system has a comparable function and importance to the vehicle as has its natural counterpart to a living animal. In general, it is required to support the weight of the loaded vehicle body above the ground, act to distribute the weight selectively, develop tractive effort with the ground for locomotion, make continuous space adjustments between the moving vehicle bodies and ground irregularities, act to control any deflection of the vehicles from their selected courses, provide a means for changing the courses of the moving vehicles, and protect the vehicles and their contents from excessive shocks due to ground irregularities. Thus, it is evident that the suspension system is the chief operator in determining the mobility of a vehicle and any improvements or increases of a vehicle's mobility must come as a result of improvements to the suspension system.

SECTION II PURPOSE AND FUNCTION

1-1 IMPROVE MOBILITY

A popular, but erroneous, opinion regarding the functional objective, or purpose, of the suspension system—one that is generally shared by casual or lay observers—presumes it to be to cushion the vehicle body and its contents from uncomfortable shocks as the vehicle traverses irregular terrain. Although this is one of the means by which the suspension system achieves its functional objective, it is not in itself that objective. A studied consideration of the question will reveal that one of the chief reasons for wanting to cushion the vehicle body and its contents from terrain induced shocks is to permit increased vehicle speed (Refs. 1 and 2). A vehicle with an absolutely rigid suspension system can traverse a rough terrain with relative comfort if the speed is very low and is kept below some maximum. As the speed is increased above this maximum, the vertical and angular accelerations of the vehicle become intolerable to the structure, the cargo, and to the occupants, causing the speed to be reduced to an acceptable comfort level. The introduction of an elastic support system between the vehicle body and the ground cushions the vehicle against the detrimental effects of the ground irregularities and permits a speed increase to a new maximum.

Furthermore, as the rigidly suspended vehicle travels over rough ground, it experiences large, erratic vertical accelerations. Each such vertical acceleration produces a different loading upon the supporting terrain resulting in variations in the traction developed by the ground-contacting elements of the vehicle. Not only is traction necessary for developing speed, but it is essential to lateral stability, steering control, and braking. Note, however, that these factors—speed, traction, lateral stability, steering control and braking—are the essential elements of vehicle mobility. One can conclude, therefore, that the primary purpose, or functional objective, of the suspension system is to improve the overall mobility of the vehicle.

Several corollary functions of the suspension system exist, some of which are unique to military vehicles. These are discussed briefly in the paragraphs that follow.

1-2 SUPPORT VEHICLE BODY OR HULL

A somewhat obvious function of the suspension system is that of supporting the vehicle body or hull at some selected height above the ground. This feature of having some space beneath the vehicle body has both desirable and undesirable aspects, though usually not at the same time. It is desirable in that it permits the vehicle to straddle certain obstructions in its path; such as boulders, logs, deep ruts, and holes that otherwise might be obstacles. It permits the vehicle to negotiate deep mud, snow, and soft sand without becoming bellied, and to ford shallow bodies of water without the risk of water damage to the power plant or cargo. Furthermore, it permits the incorporation of an elastic support system between the vehicle body and the ground to attenuate the effects of ground irregularities.

Among the undesirable aspects of raising the vehicle body off the ground are such factors as: increased overall vehicle height; increase in height of vehicle center of gravity; introduction of problems of lateral strength and stiffness; increased overturning effects of equipment operating on vehicle, weapon recoil, and blasts; and increased vehicle weight and cost. Increasing overall vehicle height makes it more vulnerable to enemy observation and fire, and increases the cargo space required to transport the vehicle. Increasing the height of the vehicle's center of gravity makes the vehicle less stable, particularly on side slopes and when cornering.

The operation of vehicle-mounted equipment that has an over-turning effect on the vehicle—such as cranes, power shovels, draglines, or the firing of weapons—is inhibited by an increase in vehicle height due to the increased moment arm of the overturning couple.

The added weight resulting from an increase in overall height requires increased power for comparable vehicle performance and affects the ground pressure of the vehicle which, in turn, has a direct bearing upon vehicle mobility. The effect of increased weight upon fuel consumption and upon the suitability of the vehicle for airborne operations is obvious. Thus, even though a vehicle designer may desire the advantages that become available when the vehicle body is raised above the ground, they do require a price in the form of design compromise.

1-3 PROVIDE LATERAL STABILITY

Since the majority of vehicles, both wheeled and tracked, have a suspension system that holds the vehicle body above the ground by means of an elastic support system, problems arise in keeping the vehicle reasonably level. When rounding a curve, the centrifugal force acting on the vehicle together with the cornering force developed by the ground-contacting elements (tires, tracks, feet, etc.) act as a couple which tends to overturn the vehicle outward about the outer wheels (or track). This causes the center of gravity of the vehicle body to shift toward the outside of the curve, increasing the vertical load upon the elastic support system on that side of the vehicle and decreasing the vertical loads on the support system on the inside of the curve by a like amount. Since the deflection of an elastic system is proportional to the forces acting upon it, this unbalance of vertical forces between the two sides of the vehicle will tend to tip the vehicle body toward the outside of the curve.

A similar lateral shifting of the center of gravity of the vehicle body takes place when operating on a side slope, when operating crane-type equipment off the side of the vehicle, or when firing heavy weapons broadside from the vehicle. Under the severe conditions imposed by the military environment in which the military vehicles must operate, this lateral instability is not merely an unpleasant inconvenience. Structural failure of suspension components will result from the large lateral forces that are encountered if sufficient lateral rigidity and strength are not designed into the system. Furthermore, a very real danger exists of the vehicle overturning under the conditions mentioned.

In order to minimize, prevent, or counteract the effects of this lateral unbalance of forces, various means are incorporated into the design of suspension systems. These take the form of stabilizing linkages; variable-rate spring systems; pneumatic, hydraulic, and hydropneumatic compensating devices; semi-active and active suspension systems; and suspension lockouts. These, and others, are discussed in this handbook.

1-4 PROVIDE LONGITUDINAL STABILITY

The foregoing discussion of the function of the suspension system to provide for lateral stability of the vehicle also applies to the provision of longitudinal stability. Instability of the vehicle body or hull in the longitudinal direction is caused by such factors as operations on longitudinal slopes, vehicle acceleration and braking, crane-type equipment operating over front or rear of vehicle, bulldozing operations, towing, firing of heavy weapons, etc. Under these conditions, the forces acting upon the vehicle body or hull result in a couple tending to rotate the body about its lateral axis. Although the danger of the vehicle actually overturning about its axis is not as great as in the previous case, due to the greater dimension along the longitudinal axis, it is not impossible and has happened under severe conditions. Longitudinal instability is minimized by the same methods mentioned in the previous paragraph for reducing lateral instability.

1-5 PROVIDE SELECTIVE DISTRIBUTION OF WEIGHT ON GROUND

Obviously, the weight of the vehicle must be transferred to the ground if the ground is to support the vehicle. The way in which this weight is distributed over the ground, however, has an important bearing upon such factors as sinkage, rolling resistance, drawbar pull, and ride characteristics of the vehicle. A uniform weight distribution is not necessarily the most desirable, as one might assume; hence, it is a function of the suspension system to distribute the vehicle weight in accordance with a predetermined design plan.

Furthermore, as the vehicle traverses irregular terrain, it is subjected to large, erratic vertical accelerations, particularly at high speeds. These accelerations produce radical variations of terrain loading beneath the ground-contacting elements of the vehicle, the magnitude of which is the product of the acceleration at any instant and the mass of a particular portion of the vehicle. This leads to a loss of traction due to insufficient ground pressure when the terrain loading is decreased, and due to soil failure when the terrain loading becomes excessive. Soil failure may also increase the rolling resistance at the same time that tractive effort is decreased. Thus, the overall mobility of the vehicle suffers from excessive variations in terrain loading.

The suspension system is designed to isolate the mass of the vehicle body from the vertical accelerations experienced by the ground-contacting elements and the so-called "unsprung mass." The degree to which this is accomplished determines the amount by which the fluctuations of the ground loading is reduced.

1-6 ADJUST FOR TERRAIN IRREGULARITIES

Since one of the functions of a good suspension system is to permit the unsprung mass to follow closely the irregular contour of the ground with minimum disturbance to the vehicle body, it follows that it must have the capability of varying its vertical height to adjust for terrain irregularities. The response of this system must be as rapid as possible to prevent the development of vertical acceleration forces that would disturb the equilibrium of the vehicle body and affect the load being exerted on the ground. Furthermore, the vertical adjustment capabilities must be as large as is practical within the limiting physical dimensions of the vehicle envelope as dictated by the utility intended for that vehicle.

1-7 PROVIDE CREW AND PASSENGER COMFORT

The provision for crew and passenger comfort is another corollary function of the suspension system. Since it is not possible to completely isolate the crew and passenger compartments from the vertical accelerations experienced by the unsprung mass, the elastic suspension system attenuates the acceleration forces it transmits. Crew and passenger comfort depend not only upon the amplitude of the vibrations but also upon the frequency. Certain frequencies bring on discomfort and nausea (motion sickness) even though the amplitude of the vibrations are acceptable. Therefore, the suspension system is also required to alter the vibrations to acceptable frequency levels.

Noise, too, produces discomfort, fatigue, nervousness, and even illness. Tracked vehicles are particularly notorious for the amount of noise they generate as the successive track blocks strike the ground. The suspension system plays an important part in limiting the transmission of this noise to the vehicle hull.

1-8 PROTECT VEHICLE COMPONENTS AND CARGO FROM SHOCK AND VIBRATION DAMAGE

Ground-induced shocks and vibrations affect every component of the vehicle as well as its cargo, passengers, and crew and are probably the most troublesome problem area in the entire field of vehicle design. Not only must the suspension components be strong and sufficiently rugged to withstand the pounding to which they are subjected, but they must be capable of attenuating the shocks to a level that is acceptable to the rest of the vehicle, its cargo, and its occupants. This permits the vehicle body to be designed of a lighter construction; isolating shock sensitive components, equipment, or cargo becomes much less of a problem; the reliability of vehicle components; and the crew and passengers can enjoy greater comfort. These improved features permit increased vehicle speed, resulting in improved mobility—the ultimate aim of the military vehicle designer.

1-9 PROVIDE TRACTION CONTACT WITH GROUND

A further function of the suspension system is to provide the necessary traction contact with the ground The tractive effort of a smooth wheel or track is limited by the coefficient of friction between the surfaces in contact. The addition of aggressive treads to wheels, or grousers to tracks, will increase the grip of these elements upon the ground and permit the development of more traction. When operating in soft sand, snow, or mud, good traction contact is extremely important. Under these conditions, specially designed ground-contacting elements are often necessary in order to develop the required propulsive effort, or traction. A similar situation exists in the case of amphibious vehicles swimming in deep water and using their normal wheels or tracks for water propulsion. In these cases, the tractive elements act somewhat like paddles to propel the vehicle through the water.

1-10 TRANSMIT DRIVING AND BRAKING TORQUES

Although the suspension system is primarily the impact and vibration absorbing mechanism of the vehicle, its design is influenced to a greater extent by the vehicle's power transmission requirements. The reactions to driving and braking torques must be transmitted through the suspension system since this is the only connection between the vehicle and the ground. Thus, the elastic elements of the suspension system must be made sufficiently rigid to resist these torques, or other means must be introduced to accomplish this end. These are usually in the form of torque tubes, torque arms, or mechanical linkage systems.

1-11 PROVIDE OBSTACLE-CROSSING CAPABILITIES

The ability to cross obstacles is a requirement especially important to military vehicles. Obstacles may be in the form of vertical obstruction such as boulders, logs, embankments, or low walls; or in the form of ground depressions such as holes, ruts, ditches, or trenches. In crossing obstacles of this type, there is a tendency to lose traction due to several factors. When climbing a vertical obstacle, there is a redistribution of forces acting on the ground brought about by the tilting of the vehicle. This causes a reduction of the normal force being applied by the wheels or track elements to the obstacle itself and an increase of the normal force being applied by the other wheels or track elements. The former condition will obviously result in less traction because of the reduced normal force. The latter condition may permit an increased tractive effort if the soil conditions are favorable, or it may result in a reduction or loss of traction if the soil fails due to the increased load upon it. In the latter case, the vehicle will only succeed in digging its tracks or wheels into the ground until the vehicle becomes totally immobilized.

When bridging a depression or ditch, a similar situation occurs. In this case, the wheels or track portions that become suspended over the ditch lose all traction because they are not even making contact. The remaining ground-contacting elements experience an increased load which, again, may either improve their traction or degrade it, as in the previous case. If the increased load causes the supporting soil to fail, the vehicle will again become immobilized.

A well designed suspension system will adjust for terrain irregularities and maintain an optimum load distribution on the ground to minimize overloading the soil. Aggressive tread on tires, or grousers on tracks, help maintain traction under difficult load conditions. Thus, it becomes readily apparent that the suspension system plays a prominent role in the ability of a vehicle to negotiate obstacles.

1-12 PROVIDE MEANS FOR CHANGING COURSE

Although directional control of a vehicle is done by means of the steering system, the actual maneuvering of the masses is accomplished by forces exerted through the suspension system to the ground. Lateral forces are introduced when the vehicle changes course, particularly at high speeds, which introduces problems of lateral stability. A properly designed suspension system will maintain lateral stability under all operating conditions and will permit the vehicle to change course at reasonably high speeds. A loss of lateral stability on a curve will force a vehicle to slow down, therefore reducing its mobility.

Various facilities are incorporated into the suspension to provide a means of changing course. In wheeled vehicles, directional changes are usually accomplished by pivoting the front wheels on their supports and pointing the planes in which they are rotating in the direction the vehicle is required to travel. Rear steered vehicles accomplish the direction change by directing the planes of the rear wheels in a direction opposite to that desired for the vehicle. Both front and rear steer are sometimes used in vehicles having an extremely long wheelbase. Camber steering is accomplished by tilting the front wheels. Wheeled vehicles can also be steered by varying the relative speeds of the wheels on each side of the vehicle.

Directional control of tracked vehicles is usually effected by varying the relative speeds of the two tracks. Mechanically bowing the tracks in the direction of steer has also been used but is not a popular method because only large radius turns are possible and the method introduces many complications and problems.

1-13 PROVIDE STABLE GUN PLATFORM

The mounting of weapons, particularly heavy weapons, upon a vehicle chassis places additional requirements upon the suspension system. Aside from the obvious requirements imposed by the heavy recoil of the weapon, there is a problem of maintaining a stable weapon platform under all conditions of elevation and traverse of the weapon Provisions are sometimes made to lock-out the action of the suspension system when the weapon is fired. This not only assures a stable gun platform while firing, but it protects the elastic components of the suspension system from the recoil shocks and adds the unsprung mass to the sprung mass to aid in maintaining a firm firing position on the ground.

In general, the accuracy of fire from vehicle-mounted weapons delivered while the vehicle is in motion leaves much to be desired—particularly if the vehicle is moving over rough terrain. Various weapon stabilizing systems have been introduced from time to time with limited effect. In theory, the ideal suspension system would maintain a stable gun position despite the vehicle speed or terrain irregularities. In practice, the theoretical ideal is not always attainable, but any degree of success in attaining that goal is an improvement.

SECTION III BASIC DISCUSSIONS

1-14 SUSPENSIONS, GENERAL

The suspension system of a vehicle is defined as that complex of mechanical, structural, pneumatic, hydraulic, and electrical components that provides or is associated with the provision of flexible support between the ground and frame, or ground and hull, of the vehicle. This includes, in addition to the elastic or resilient components, the load carrying members, traction members, and such members as are required to control the geometrical relationship of the elastically constrained parts. Although some of the components transmit power or provide reaction torque, the suspension system is not considered part of the power train; however, its design is greatly influenced by the vehicle's power transmission and steering requirements. Dissimilar methods of locomotion and types of ground contact, resulting from the wide range of military requirements, produce major design variations in suspension components. For this reason, it is necessary to expand the general definition of a suspension system to cover every type of vehicle—wheeled, tracked, and unconventional.

1-15 SUSPENSIONS, WHEELED VEHICLES

Typical suspension systems for wheeled vehicles are illustrated in Figures 9-2, 9-3, 9-5 and 9-8. The main components that comprise these systems are: (a) springs, (b) shock absorbers, (c) axle assemblies, (d) bogie assemblies, (e) wheels, (f) tires, (g) torque arms and tubes, (h) control arms, (i) stabilizers (antiroll bars), and (j) bump stops. Detailed discussions of these components are given in Chapter 9. For the purpose of this introductory definition, the basic functions of these components are given as follows:

(a) *Springs.* Provide the elastic support for the vehicle body.

(b) *Shock absorbers.* Regulate the dynamic characteristics of the sprung and unsprung mass by introducing damping into the system.

(c) *Axle assemblies.* Provide cross support for the vehicle body and a spindle upon which the wheels revolve and through which the vehicle weight is transferred to the wheels. Some axle assemblies have the additional function of transmitting propulsive power to the wheels, some have a role in the steering of the vehicle, and some do both, i.e., transmit power and provide for the steering.

(d) *Bogie assemblies.* Distribute the vertical forces equally to tandem axles, maintain uniform ground pressure under the tandem wheels of the assembly despite ground irregularities, and reduce vertical displacements and shock to the vehicle body through the geometrical arrangement of the components.

(e) *Wheels.* Minimize the resistance of the vehicle to translatory motion in the desired direction while providing vertical support for the vehicle body and lateral stability. Some wheels have secondary functions of transmitting driving and braking torques, and providing steering actions.

(f) *Tires.* Provide traction contact with the ground for propulsion, braking, and steering. Some tires have an additional function of reducing road shocks, vibrations, and noise transmitted to the suspension system.

(g) *Torque arms and tubes.* Provide reaction torque to counter the driving torque of the power train and transmit the driving thrust to the vehicle frame or body.

(h) *Control arms.* Maintain the desired geometric relationship between the various elastically constrained suspension components.

(i) *Stabilizers.* Reduce the amount of angular movement of the vehicle body in the vertical plane.

(j) *Bump stops.* Limit the maximum vertical excursion of the elastically supported mass, both upwards and downwards, from its normal free-standing position.

Not all of the foregoing major components are necessarily found in every wheeled vehicle suspension system. Furthermore, the properties of some components may be modified or expanded to provide some function normally provided by another component. Nevertheless, the components listed, when present in a vehicle, comprise the vehicle's suspension system.

1-16 SUSPENSIONS, TRACKED VEHICLES

Suspension systems for tracked vehicles are basically quite similar to those found in wheeled vehicles, although this similarity may not be obvious to a casual observer. The main difference is the presence of the track assemblies which are absent in the wheeled vehicle suspension. If one considers the track assembly merely as a portable roadway upon which the vehicle rolls, the similarity to a wheeled vehicle will become more apparent (Figures 10-1, 10-2, 10-3 and 10-4). It is not surprising, therefore, to find the suspension system comprised of similar components. The major items are: springs, shock absorbers, road wheels, tires, road wheel arms, bogie assemblies, track assemblies, sprockets, support rollers, bump stops, and idler wheel assemblies. These are discussed in detail in Chapter 10. The basic functions of the components common to both wheeled and tracked vehicles are given in the previous section. The functions of those components more or less unique to tracked vehicles are:

(a) *Road wheel arms.* Support the road wheel spindles, upon which the road wheels rotate, and transmit the vertical loads experienced by the road wheels to the torsion bars or other spring systems.

(b) *Track assemblies.* Distribute the vehicle weight over a larger ground area to reduce the ground pressure and improve traction.

(c) *Sprockets.* Function as part of the power train since their main function is to drive the track to propel the vehicle. They are always considered with the suspension elements, however, because their design is heavily dependent upon the design of the track.

(d) *Support rollers.* Support and guide the return run of the track assembly.

(e) *Idler wheel assemblies.* Return the driven track to the driving sprocket. Compensating idlers compensate for changes in the circumferential length of track envelope, resulting from the envelope constantly changing its shape as it follows the contour of irregular terrain.

There is a type of suspension system that is used in a particular class of track-laying equipment that is quite different from that found in contemporary combat and tactical vehicles. It is found in such equipment as heavy duty crawler cranes and shovels, agricultural and industrial-type crawler tractors, and bulldozers. Basically, this system consists of a series of small idler wheels, mounted to a longitudinal beam beneath the vehicle to provide a supporting base. The track, passing under the rollers and around the beam longitudinally, can make relatively small adjustments to the contour of the ground due to a very limited deflection capability of the spring system.

The above type of suspension system is quite adequate and practical for the equipment to which it is applied. Heavy duty crawler cranes and power shovels require only a large, slow-moving base or platform, and they operate on almost level ground with very little contour variation. Width and length of base (for purposes of stability) and low ground pressure are the chief considerations in the design of suspensions for this equipment. In the design of agricultural and industrial type tractors, adhesion, traction, and low ground pressure are the main considerations—since the purpose of the vehicle is to provide maximum drawbar pull. Vehicle speed requirements are relatively low and the terrain requirements are not particularly severe. Requirements placed upon bulldozers used in construction operations approach the severity encountered in combat and tactical operations but, even here, the requirements of speed, extremes of terrain and weather, and power and weight considerations are not as severe.

Since the design problems associated with this type of suspension system are far less complex than those encountered in the design of suspension systems for high performance combat and tactical vehicles, the design of the beam-type of tracked suspension system is not treated in this handbook.

1-17 SUSPENSIONS, SLED-TYPE VEHICLES

Suspension systems for sled-type vehicles consist functionally of the same general components as make up other suspension systems; namely, ground-contacting elements to distribute the vehicle weight selectively over the ground, supporting elements to hold the vehicle body at some desired distance above the ground, a system of adjustable links to allow the ground-contacting elements to follow the irregular contour of the ground with minimal disturbance to the vehicle body, and provisions to control the course of the vehicle. One difference between a sled-type vehicle and a wheeled or tracked vehicle is that the runners cannot be used to propel the vehicle by developing tractive effort with the ground, as is the case with wheels or tracks. Sled-type vehicles are usually towed by a prime mover or are propelled by an auxiliary track, traction wheel, air propeller, or thrust motor. The runners are, therefore, designed for minimum friction with the ground and minimum resistance to forward motion. In this respect, the sled-type vehicle even differs from an unpowered trailer because the wheels of a trailer are required to develop sufficient traction to overcome their rolling resistance to preclude sliding over the ground.

Since sled-type vehicles are primarily designed to operate on snow, ice, and marshy ground, the development of sufficient propulsion is more of a problem than when operating on firm soil. For this reason, sled-type vehicles generally operate at relatively slow speeds, making the suspension requirements much less severe. Consequently, the elastic elements (springs and shock absorbers) are often omitted. Heavy duty cargo sleds and sleds designed for high speeds, however, do include these elements. The methods of calculating the dynamic requirements of the suspension system are much the same as for conventional systems discussed in Chapter 8. The design of skis, sleighs, and toboggans is covered in Chapter 6, Section III.

1-18 SUSPENSIONS, UNCONVENTIONAL VEHICLES

The category of unconventional vehicles embraces many types designed to meet special applications or to operate in unusual environments. Their suspension systems fall into four general categories; namely, unusual wheel types, unusual track types, walking and leaping types, and screw types. Ground-effect vehicles, which float a few inches above the ground on a cushion of air, are not discussed in this book. The main objective of these unusual suspensions is to increase vehicle mobility by enabling the ground-contacting elements to develop greater tractive effort and less resistance to motion than is possible with conventional components. The principles governing terrain-vehicle relationships are in the domain of land locomotion mechanics; therefore, more detailed discussions of the unusual suspension types are given in Chapter 6.

1-19 COMMON SUSPENSION TERMINOLOGY

1-19.1 SPRUNG MASS AND UNSPRUNG MASS

The portion of a vehicle that is supported by the primary elastic elements (springs, torsion bars, etc.) of a suspension system is referred to as the *sprung mass*. The portion of the vehicle that is not so supported constitutes the *unsprung mass*. Thus, the sprung mass is generally comprised of such items as the vehicle body or hull, frame, power plant, transmission, transfer assembly, fuel, coolants, lubricants, armament, ammunition, fire control equipment, cargo, crew, passengers, and all other components that are attached to these elements. The unsprung mass is usually comprised of the suspension components plus certain portions of the power train. Proportionate parts of the springs, shock absorbers, swinging linkages, and other components attached to the frame are included in both the sprung mass and in the unsprung mass. In a tracked vehicle, that portion of the track which is not in contact with the ground is considered part of the sprung mass, and the portion that is in contact with the ground is part of the unsprung mass.

When considering the mobility of a vehicle, it is generally conceded to be more advantageous to have a small unsprnng mass. The lighter unsprung mass is capable of faster response to terrain irregularities, resulting in more uniform contact with the terrain and providing improved traction and steering characteristics. With the more uniform terrain contact comes a decreased variation in the loading of the supporting soil, resulting in improved floatation and less rolling resistance. Stresses induced into the suspension system by impacting rough terrain at high speeds are also reduced with a lighter unsprung mass. This is particularly important to the life of such components as the wheels and tires, track components, and road-wheels.

The magnitude of the unsprung mass has an inverse effect upon the frequency of wheel danee, a condition discussed in a later section. Wheel dance tends to induce secondary disturbances into the main portion of the vehicle—especially if the terrain irregularities recur at the natural frequency of the wheel dance. A decreased unsprung mass increases the frequency of the wheel dance vibration. This increased frequency may be desirable if it is increased above the frequency of the ground disturbances in the operating speed range of the vehicle. The greatest objection, then, to the higher frequency is an increase in harshness of the secondary disturbances of that frequency which may be induced into the main portion of the vehicle.

1-19.2 SHOCK VERSUS VIBRATION

The term *vibration* is used in this book to describe an oscillation in a mechanical system. A mechanical *oscillation* is a continuing variation of the position of a mechanical element about a given reference position—alternating from one side of the reference to the other. The complete displacement history of a periodically oscillating element—from any starting point, through all of the successive positions on both sides of the reference axis, and back to the starting point—is a *cycle*. A *period* is the time required to complete one cycle. The reciprocal of the period, or the number of cycles occurring during a unit of time, is the *frequency* and is usually expressed as cycles per second. The displacement in one direction from the reference axis is the *amplitude*. These basic terms are shown graphically in Figure 1-1. A *vibration* is defined by stating the frequency (or frequencies in a complex vibration) and the amplitude.

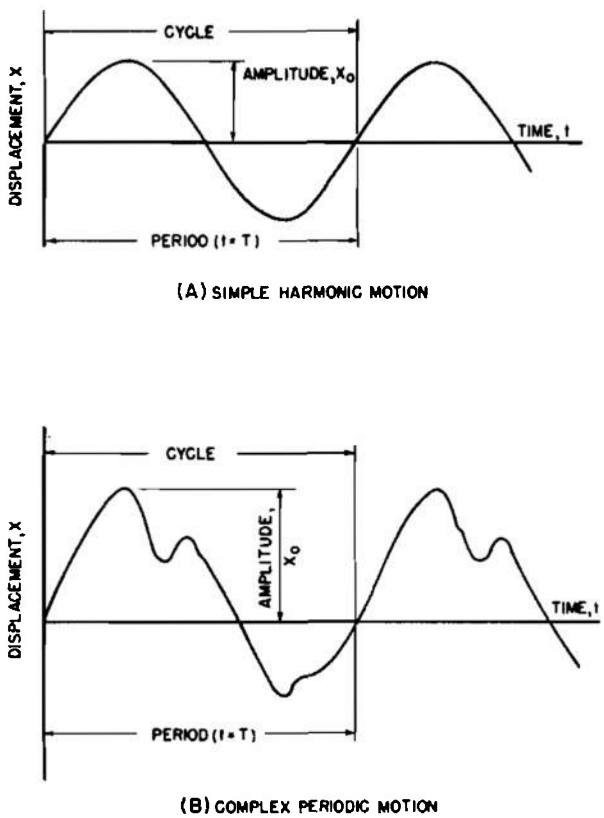


Figure 1-1. Elements of Periodic Motion

Vibrations fall into two basic categories, namely, *periodic vibrations* (also called deterministic) and *random vibrations*. The foregoing definitions apply specifically to periodic vibrations. By definition, a *periodic vibration* is a motion that repeats itself in all of its particulars after a certain interval of time (Refs. 3-6). Since the instantaneous amplitude x of the displacement varies with time t it can be plotted to show the displacement-time history graphically (Figure 1-1). Furthermore, the displacement-time history can also be expressed mathematically if x, the instantaneous amplitude, is expressed as a function of time t:

$$x = f(t) \tag{1-1}$$

The simplest kind of periodic motion is the *sinusoidal* or *simple harmonic motion*. It exists when the relationship between x and t satisfies the equation

$$x = x_o \sin \omega t \tag{1-2}$$

where x_o represents the maximum value of the displacement (amplitude). The factor ω expressed in radians per second, is known as the *circular frequency* and comes from the classic representation of vibratory motions by means of rotating vectors (Refs. 7-10). A complete cycle of the vibration takes place when ωt has passed through 360°, or 2π radians. Thus, when $\omega t = 2\pi$, the time interval t is equal to the period T, or

$$T = \frac{2\pi}{\omega} \text{ sec} \tag{1-3}$$

Since the frequency f is the reciprocal of the period T:

$$f = \frac{\omega}{2\pi} \text{ cycles per second} \tag{1-4}$$

The first and second derivatives of the displacement equation, (Eq. 1-2), with respect to time give the equations for the velocity and acceleration, respectively. Thus

$$\text{Velocity} = \frac{dx}{dt} = \dot{x} = x_o\omega \cos \omega t \tag{1-5}$$

$$\dot{x}_{(max)} = x_o\omega \tag{1-6}$$

$$\text{Acceleration} = \frac{d^2x}{dt^2} = \ddot{x} = -x_o\omega^2 \sin \omega t \tag{1-7}$$

$$\ddot{x}_{(max)} = x_o\omega^2 \tag{1-8}$$

Complex periodic vibrations include all periodic oscillations that are not sinusoidal (Figure 1-1(B)).

Mathematical theory has shown (Refs. 11, 12) that any periodic curve $f(t)$, having a frequency ω, can be considered a summation of sinusoidal curves of frequencies ω, 2ω, 3ω, 4ω, etc., provided that $f(t)$ repeats itself after each time interval $T = 2\pi/\omega$. The summation of sinusoidal curves results in the classic series of sine terms known as a Fourier series, and the methods for solving Fourier series are well known. Thus, the amplitudes and phase relationships of the various sine curves that comprise the complex vibration can be determined analytically when the term $f(t)$ is known. Many references are available that give detailed discussions of the procedures involved. Some of these are listed in the references at the end of this chapter.

Random vibrations, the second general category, do not lend themselves to rigid mathematical analysis as do periodic vibrations. Random vibrations are oscillations whose instantaneous magnitude can be specified only in terms of the probability that the amplitude will be within a given range of values during a given time interval. Thus, instead of applying exact mathematics to obtain definite answers, statistical analysis and the mathematics of probability are applied. These procedures are well established and ample literature is available discussing them (Refs. 13-17).

Shock is a somewhat loosely defined aspect of vibration, wherein the excitation is nonperiodic; i.e., in the form of a pulse, a step, or a transient vibration. Furthermore, the word "shock" implies a degree of suddenness and severity. A mechanical shock is characterized by significant changes in stress, position, acceleration, velocity, or displacement occurring in a relatively short period of time. The time factor is considered short when the duration of the excitation is less than the longest natural resonant period (the fundamental natural period) of the member or equipment under consideration. Some definitions (Ref. 18) limit the duration of the excitation to "less than one half of the fundamental natural period." The shock will, therefore, excite the member so that it will vibrate at its natural frequency.

In general, the determination of the damage potential of a shock, and the technique of designing structures to resist shock, involves approximate calculations of the responses of structures to shocks. Responses of stress and strain are significant from the strength-of-materials point of view; relative displacement responses are significant from clearance considerations; and accelerations, velocity, and relative displacement response are significant in evaluating the damage potentials of different shock motions.

A meaningful description of shock requires more than a statement of its magnitude. The duration, and a time related history of the rise and decay of the acceleration, velocity, displacement, or forces the shock (the waveform) must be either expressly stated or tacitly implied. Without these, an expression of the maximum magnitude of a shock is meaningless. Shocks are generally classified as *velocity*, *simple impulse*, and *complex shocks*.

Velocity shock (Refs. 19, 20) is a simplified concept that considers the shock as a sudden velocity change (Figure 1-2(A)). It is assumed to be an impulse of such an infinitesimally short duration that the acceleration magnitudes and waveforms are immaterial. The application of this concept represents a large class of shock motions for which this approximation is sufficiently accurate for engineering purposes. Shocks that involve the dropping of equipment onto hard, rigid surfaces, or impacts of relatively light objects with heavy, rigid bodies, can be considered as velocity shocks. Mathematical procedures for evaluating this type of shock loading are given in Refs. 19-23.

Simple impulse shock (also called *pulse-type shock*) is a shock during which the waveform, or time history of the acceleration, velocity, displacement, or force, cannot be neglected but can be approximated by a waveform of simple shape—one that is easy to generate and tractable to use. Figure 1-2(B) shows a shock type known as an acceleration step in which the acceleration is assumed to rise simultaneously to a constant value. The resulting velocity and displacement-time histories are shown. Figure 1-2(C) shows a shock during which the acceleration time-history is a half-sine pulse. It is the type of acceleration experienced by a rigid mass when it is dropped upon a linear spring of negligible mass. The half-sine

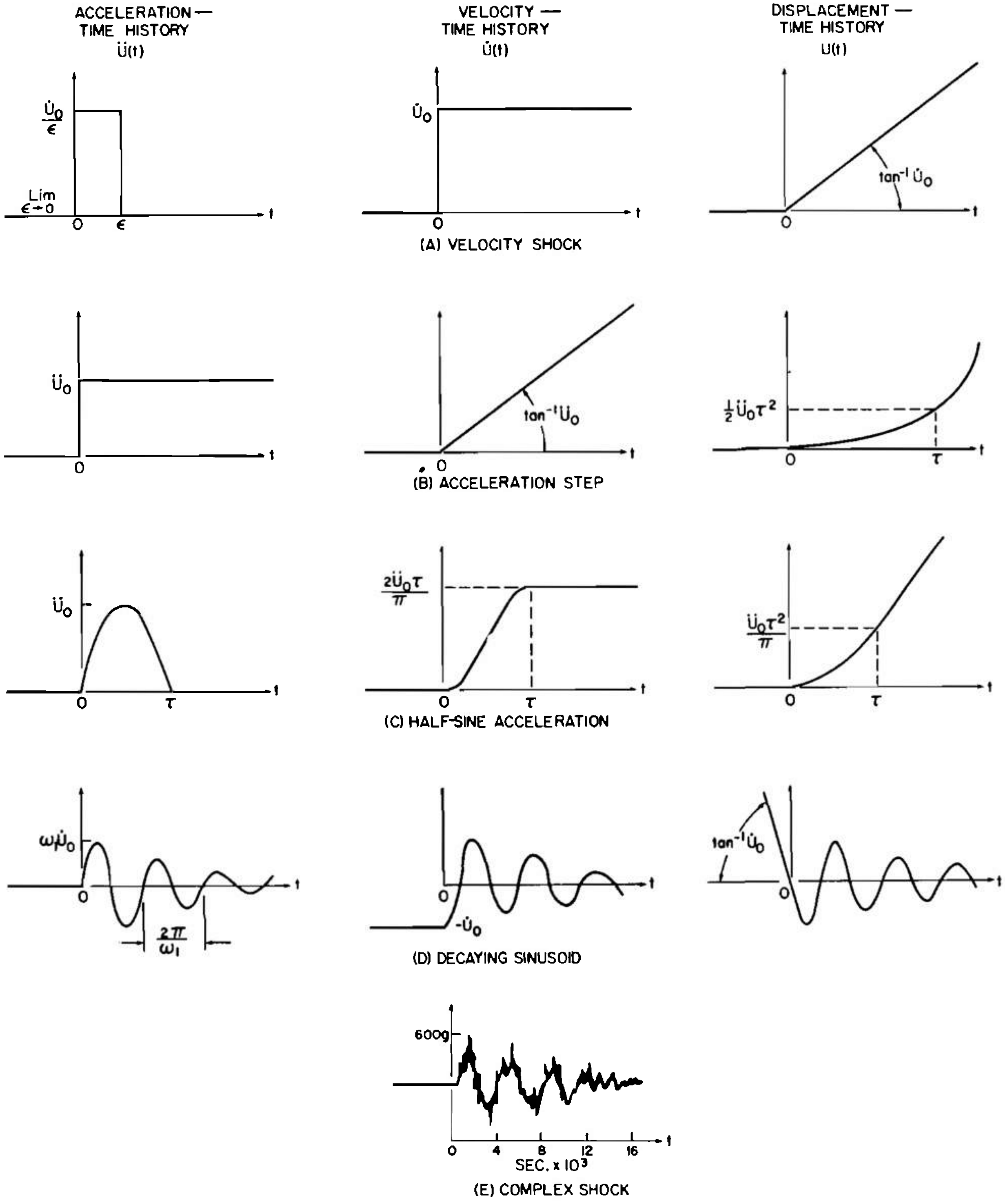


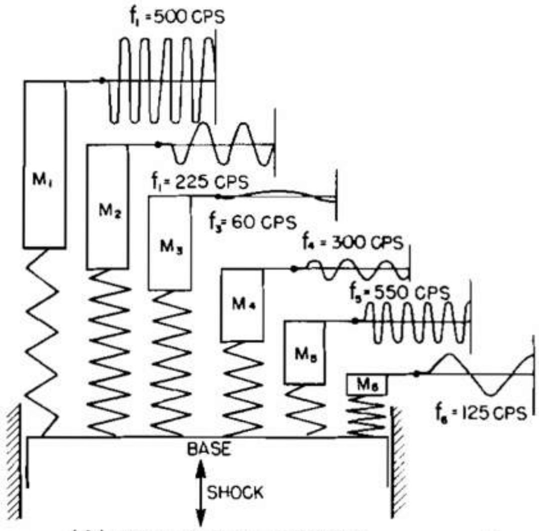
Figure 1-2. Examples of Shock Motions

pulse is very popular in shock analysis, not because it is the proper simulation of many situations, but because it is easy to generate and can be easily expressed mathematically. Figure 1-2(D) illustrates the acceleration, velocity, and displacement-time histories of a decaying sinusoidal shock. Here the impulse is assumed to have imitated a sinusoidal vibration at the resonant or natural frequency in the presence of a damping factor. The damping causes the vibration to decay at some constant rate. The mathematical procedures applicable to design problems involving impulse-type shock are given in many standard texts on vibrations (Refs. 24-26).

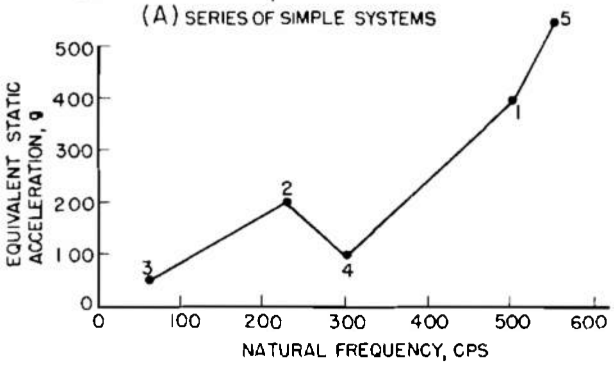
Most shock motions encountered in practical equipment, particularly large military equipment and structures, are extremely complex (Figure 1-2(E)). They react back upon the excitations and make them complex, even though the excitations might have been simple without this reaction. These motions are commonly known as *complex shocks*. They cannot be expressed in simple mathematical or graphical form and are, therefore, expressed in terms of their shock spectra.

A *shock spectrum* is defined as the maximum responses of a series of simple systems (Figure 1-3) to the shock motion, and is expressed as a function of the natural frequencies of the simple systems. Unless stated otherwise, the simple systems are considered to be undamped. The responses are taken as relative displacements of the simple mass elements with respect to the base. Velocity and acceleration spectra are defined as the displacement responses multiplied by $2\pi f$ and $2(\pi f)^2$, i.e., ω and ω^2, respectively. Complex shocks are treated in Refs. 27-30. Shock spectra do not define what the shock motions are but rather describe what the shock motions do—in effect, they describe their damage potentials. The spectra are used to compare the intensities of different shocks and to determine whether the output of a shock machine is equivalent to a desired range of field conditions.

Figure 1-3(B) shows the shock spectrum of the hypothetical series of single-degree-of-freedom systems shown in Figure 1-3(A). The ordinate used (equivalent static acceleration in g's) can be determined from the following relationships:



(A) SERIES OF SIMPLE SYSTEMS



(B) SHOCK SPECTRUM FOR COMPLEX SHOCK

Figure 1-3. Shock Spectrum Representation

$$\tfrac{1}{2} M V^2 = \tfrac{1}{2} k x_c^2 \tag{1-9}$$

$$n = \frac{a}{g} = \frac{F}{Mg} = \frac{V}{g}\sqrt{\frac{k}{M}} = \frac{V}{g} \quad \omega \tag{1-10}$$

where

F = force exerted on M, lb
M = mass, lb-sec²/in.
V = velocity change of the mass, in./sec
n = multiplier used to express a specific acceleration in terms of g, dimensionless
a = acceleration, in./sec²
g = acceleration due to gravity, in./sec²
k = spring rate, lb/in. (see next paragraph)
x_c = maximum spring deflection, in.
ω = circular frequency ($= 2\pi f$), rad/sec

1-19.3 NATURAL FREQUENCY

The natural frequency of a spring-mass system is the frequency at which the system will vibrate

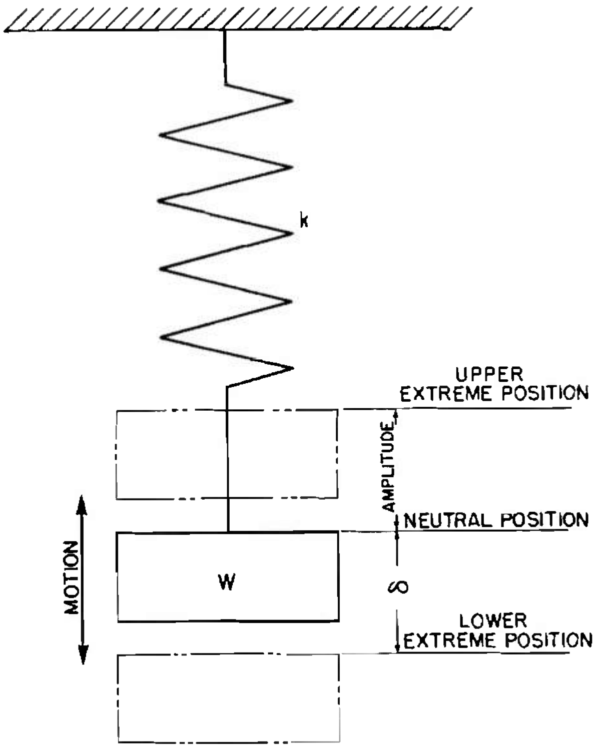


Figure 1-4. Simple Spring-Mass System

due to internal forces inherent in the system and without the benefit of fluctuating external forces of any kind. As an example, consider the simple spring-mass system shown in Figure 1-4, consisting of a mass of weight W, suspended by means of a spring whose spring rate is k (the force necessary to stretch or compress the spring one unit of length). Initially, the mass is in equilibrium under the action of two equal and opposite forces; namely, the weight W, acting downward and the spring force $k\delta$ acting upward. This is the *neutral position* (also called *equilibrium position* or *central position*) from which displacements of the mass are usually measured. The factor δ is the spring deflection under static conditions produced by the weight W.

Suppose, now, that the mass is forced downward some additional distance x and then suddenly released. At the instant of release, the spring force will be larger than the weight by an amount kx, causing the mass to start moving upward. While the mass is below the neutral position, the upward spring force is greater than the weight, and the mass will accelerate upward; although, the acceleration will progressively decrease as the neutral position is approached. When the mass arrives at the neutral position, the forces will again be balanced and the acceleration will be zero, but the mass will be moving upward with maximum velocity and will, therefore, continue to move upward. As it moves farther from the neutral position, however, the spring force will progressively decrease, leaving a progressively larger downward force to decelerate the mass. When the velocity finally becomes zero, the mass will have reached the upper *extreme position*. At this time, the downward force will be at its maximum causing the mass to start downward. The downward velocity will increase until the neutral position is again reached; whereupon, the velocity will decrease and become zero when the mass reaches the lower extreme position. Since this is the same position from whence it had started, the same upward force imbalance will exist; and the mass will continue to oscillate about the neutral position. The maximum excursion of the vibrating mass, measured from the neutral position, is the *amplitude* of the vibration.

The motion just described is known as a *free vibration*, because it is self-propagating; i.e., once started, it is perpetuated by the interaction of forces inherent to the system and is not dependent upon periodic external forces. The frequency of this free vibration is the *natural frequency* of the system. The natural frequency of a system is independent of the amplitude, increases as the square root of the spring rate k, and decreases as the square root of the mass. Or, to express this relationship mathematically:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{M}} \qquad (1\text{-}11)$$

Theoretically, the simple system just described should continue to oscillate indefinitely, once the oscillation is started. In all practical systems, however, there exists a certain amount of friction that always acts in opposition to the motion and causes the amplitude to decrease slightly with each excursion until the mass finally comes to rest at the neutral position. This friction force, which is often very complex, is called *damping*. The amount of damping present in practical systems of the type just described is so slight that it is usually neglected, and the system is considered to be undamped.

If the mass of Figure 1-4 were made to oscillate up and down by the application of a periodically fluctuating force applied to the spring support, there would occur an initial period of *transient vibration* during which the vibration characteristics would be very irregular. The damping properties of the system, however, would soon smooth out the irregularities, and a steady-state vibration would remain whose amplitude and frequency would be different from those of free vibration. This new steady-state condition is called *forced vibration*. The frequency of the forced vibration will be the same as the frequency of the fluctuating force; the amplitude, however, will depend on the magnitude of the fluctuating force and on the ratio of its frequency to the natural frequency of the system. When this frequency ratio becomes equal to unity, a condition known as *resonance* occurs. During resonance, amplitudes build up rapidly to dangerous values; hence, the purpose of many vibration calculations is the prevention of resonance.

1-19.4 DAMPING

Damping is defined as the process of effecting a continuing decrease in the amplitude of an oscillating component and is generally accomplished through some type of friction that dissipates the energy of the system with time or distance. The preceding section describes how an oscillation, once started, tends to be self-propagating. The initial force kx, acting through the initial displacement x, represents the amount of energy kx^2 that was introduced into the system and is supporting the oscillation. This energy must be dissipated, usually in the form of heat, before the system will stop oscillating.

Several methods are used for dissipating energy and, hence, damping a vibrating mechanical system. Among the more common are: solid friction or hysteresis damping, environmental damping, viscous damping, coulomb damping, inertia damping, and electromechanical damping.

Solid friction, internal friction, or hysteresis damping occurs in all mechanical vibrating systems that have restoring forces derived from elastic elements such as springs although it is not always large enough to be a deciding factor in limiting amplitude. It seems to arise from slight relative displacements between adjacent crystals or minute elements of the spring material and is often referred to as *internal friction*. It is manifested in a peculiarity of all stress-strain diagrams when they are cycled indefinitely between two limits. The stress has a slightly higher value for a given strain when the load is increasing than when it is decreasing—even though the stress is kept well within the elastic limit of the material (Figure 1-5). The area inside the hysteresis loop indicates the amount of energy that is dissipated (converted into heat) during one cycle by a unit volume of the material. It is independent of the frequency and dependent only upon the nature of the material and

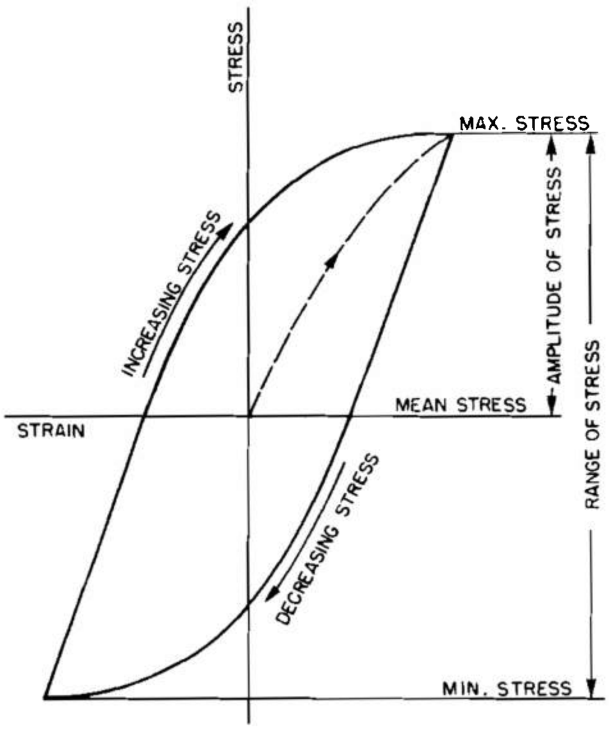


Figure 1-5. Typical Hysteresis Loop

the *range of stress*, as shown in Figure 1-5. It is usually expressed as a function of half of the stress range, which is often called the stress amplitude.

For mild steel, hysteresis damping is approximately proportioned to the 2.3 power of the stress amplitude when the stress amplitude is below 8,000 psi (Ref. 31). For higher values of stress amplitude, hysteresis damping increases much more rapidly with increasing amplitude. Hard steel has less hysteresis damping, in the same stress range, than mild steel; and cast iron has more damping than mild steel. It is difficult to estimate the amount of hysteresis, or solid friction, that will be present in a system being designed unless experimental data are available.

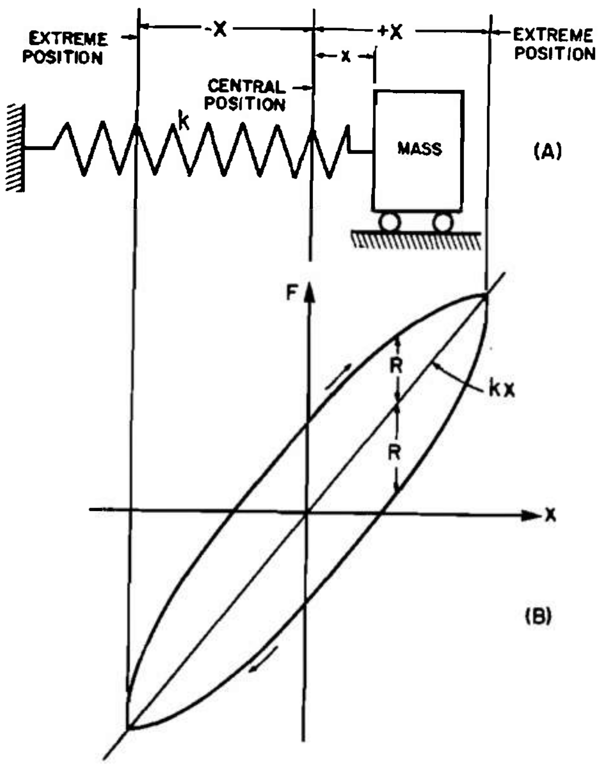


Figure 1-6. Spring-Mass System and Hysteresis Loop

The action of hysteresis, or solid friction, is illustrated in Figure 1-6. Part (A) shows a simple spring mass system set up to oscillate in a horizontal direction. If the spring in the system were alternately extended and compressed between the two extreme positions and a trace were made plotting the spring force F against the displacement x, a characteristic hysteresis loop such as shown in Figure 1-6(B) would result. The restoring force kx, is the straight line which bisects the hysteresis loop. It is evident that at any deflection x, the spring force differs from the restoring force kx, by an amount R which is the resisting force of internal friction. When the masss moves to the right ($\dot{x} > 0$), the spring force will be $kx + R$; but when the mass moves to the left ($\dot{x} < 0$), the spring force will be $kx - R$. This shows that there will always be a resisting force R opposing the motion in addition to the restoring force.

Environmental damping is the term sometimes applied to the dissipation of energy from a system vibrating in air or a fluid. As the mass vibrates it displaces the surrounding air or fluid against the opposition of the fluid friction. In the majority of engineering applications, the force developed by environmental damping is approximately proportional to the square of velocity of the fluid. The exact value of the velocity exponent depends upon the shape of the vibrating body and upon the ratio of the frictional drag to the wavemaking drag.

Viscous damping, also called *velocity damping,* is an energy-absorbing technique in which the damping force resisting the motion is a linear function of the velocity. In practice it is found where there is relative motion between two well-lubricated surfaces and where a viscous fluid is forced through a relatively long passage of small cross sectional area. From the standpoint of theoretical analysis, this is the easiest type of damping to deal with because the mathematics involved is quite simple (Refs. 32-34). Two variations of viscous damping are *degenerate viscous damping* and *hydraulic damping.* Degenerate viscous damping is characterized by a damping force that is proportional to a fractional power of the velocity, and hydraulic damping is characterized by a damping force that is proportional to the square of the velocity. Or to put it another way, if v is the velocity and c_1, c_2, and c_3 are proportionality constants, the damping force F will be

$F = c_1 V$ for viscous damping

$F = c_2 V^m$ (where $m < 1$) for degenerate viscous damping

$F = c_3 V^2$ for hydraulic damping. (Ref. 44)

Coulomb damping, also called *dry friction damping,* is the type of energy dissipation that comes from the rubbing of dry surfaces with each other. The damping force is assumed to be independent of the velocity and acceleration of the oscillating mass, to be a function only of the materials involved and of the normal force acting on them. Friction damping is not considered suitable for the principal damping requirements of an automotive vehicle because the energy dissipated with each oscillation is directly proportional to the amplitude, whereas the vibratory energy is proportional to the square of the amplitude. If a friction damping force suitable for energy dissipation of a medium amplitude vibration were used, it would be inadequate for large amplitude vibrations and excessive for small amplitudes.

Inertia damping is a type for which the damping force is directly proportional to the acceleration of the vibrating mass. Its amplitude decay characteristics are similar to those obtained with viscous damping, i.e., they result in a logarithmic curve which makes inertia damping suitable for large amplitude vibrations. At very low amplitudes, inertia damping needs to be supplemented by suitable coulomb (friction) damping.

In comparison with viscous damping, inertia damping is considered more responsive. The damping force of inertia damping is not only proportional to the accelerating force, but it acts in direct opposition to it. The damping force with viscous damping lags behind the accelerating force since it is proportional to the velocity which results from the acceleration.

Electromechanical and *electromagnetic damping* are methods of energy dissipation which make use of the interaction between electrical or electromagnetic phenomena and physical compouents of the vibrating system. Damping forces that are associated with magnetic hysteresis and eddy currents are examples of this type of damping.

The subject of damping is discussed further in Chapter 10, Section VIII—particularly as it is applied to vehicle suspension systems.

1-19.5 WHEEL DANCE

Wheel dance is a term that is applied to a vertical vibration of the unsprung mass that occurs at the natural frequency of the spring-mass system and is the principal source of secondary vibrations. The wheels of a wheeled vehicle and the road wheels of a tracked vehicle are cushioned against the impacts of the terrain, or of the track shoes, by suitable rubber tires. Each wheel is, therefore, suspended between two elastic elements that have parallel action; namely, the rubber tire and the vehicle spring. The combined spring rate of a system of this type is equal to the sum of the individual spring rates; or, if k_T and k_S represent the spring rates of the tire and spring, respectively, the combined spring rate, k_c, will be $(k_T + k_S)$; and the natural frequency of the system in cps (neglecting damping) will be:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k_c}{M}} = \frac{1}{2\pi}\sqrt{\frac{k_T + k_S}{M}} \quad (1\text{-}12)$$

where M is the unsprung mass. Since k_T, the spring rate of the tire, is usually much greater than k_S, it is the elastic factor to which wheel dance is generally attributed.

Decreasing the unsprung mass increases the frequency of the wheel dance vibration. This may be desirable if it raises the frequency above the frequency of the terrain impacts, or track shoe impacts, that are being experienced at the operating speeds of the vehicle. The secondary disturbance transmitted to the sprung mass, however, will be more harsh with the higher frequency. If this is objectionable, a compromise must be established.

1-19.6 PRIMARY VERSUS SECONDARY VIBRATIONS

The principal *primary vibrations* pertaining to automotive suspension systems are those that result directly from the movement of the vehicle over irregular terrain. As the ground-contacting elements (wheels, tracks, etc.) of the unsprung mass follow the irregular contour of the terrain surface, they experience large and erratic vertical accelerations. The fluctuating force that results from these accelerations excites the total spring-mass system. The resilient elements (springs and shock absorbers) that support the sprung mass of the vehicle on the running gear (unsprung mass) function to cushion the sprung mass from the shocks and vibrations resulting from these vertical accelerations. The vibratory motions of the sprung and unsprung masses caused by the terrain irregularities are the principal primary vibrations of concern to the automotive designer.

Additional vibrations that come under the classification of primary vibrations are others that originate from sources external to the vehicle. Among these are shocks and vibrations resulting from vehicle acceleration and braking, ballistic shock, wind pulsations, firing of vehicle-mounted weapons, and from fluctuating forces developed by vehicle-mounted equipment, such as cranes, shovels, bulldozers, etc.

Secondary vibrations, in general, are those that originate from internal sources and are usually of a higher frequency than are primary vibrations. The principal secondary vibration of concern to the vehicle designer is that developed by wheel dance, which is discussed in the preceding paragraph. Similar secondary vibrations may have their origins in the impacting of sprocket teeth against track elements when there is a discrepancy in the pitches of the sprocket and the track, in the impacting of the track blocks against the ground, and in the shimmy or wobble of the steerable wheels of a vehicle about their king pins.

Secondary vibrations are transmitted to the spring mass by the shock absorbers because these are usually selected to give the desired degree of damping at the relatively low frequencies of the primary vibrations. At the higher frequencies of the secondary vibrations the shock absorbers are much more effective, resulting in overdamping and a harsh ride.

1-19.7 DEGREES OF FREEDOM

The term *degrees of freedom* refers to the number of ways a mass is capable of moving with respect to a set of coordinate axes and with respect to other masses in a system comprised of more than one mass. Systems such as shown in Figures 1-4 and 1-6, comprised of but one mass and constrained so as to be capable of motion along only one axis, are said to have *one degree of freedom.* If more than one coordinate is necessary to completely specify the configuration of a vibrating system, either because there are several masses or the mass is capable of more than one motion, the system is said to have more than one degree of freedom. *The number of degrees of freedom in a system is equal to the minimum number of coordinates necessary to specify the configuration of the vibrating system at any time.*

A rigid body that is restrained to move in two directions, or to rotate about two axes, has two degrees of freedom. Several simple systems having two degrees of freedom are shown in Figure 1-7.

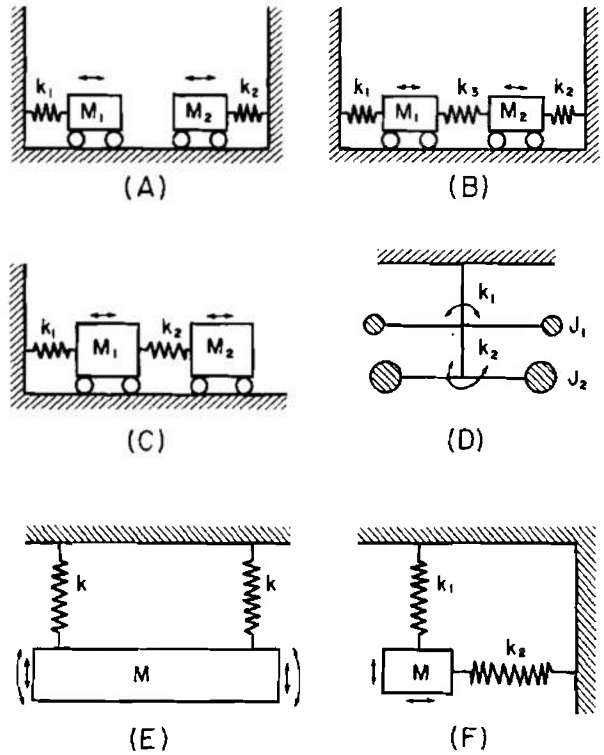


Figure 1-7. Examples of Two-Degree-of-Freedom Systems

Figures 1-7(A) to (C) show two masses with various combinations of elastic restraint. The total number of springs in a system has no influence on the number of degrees of freedom that a system has. This is wholly dependent upon the freedom of motion of the masses. Figure 1-7(D) shows a simple torsional system with two degrees of freedom; Figure 1-7(E) shows a rigid body that is free to translate vertically and to rotate about an axis perpendicular to the plane of the paper; Figure 1-7(F) shows a mass that is free to move both horizontally and vertically.

A rigid body that is perfectly free in space (except for elastic restraints) has six degrees of freedom—three in translation and three in rota-

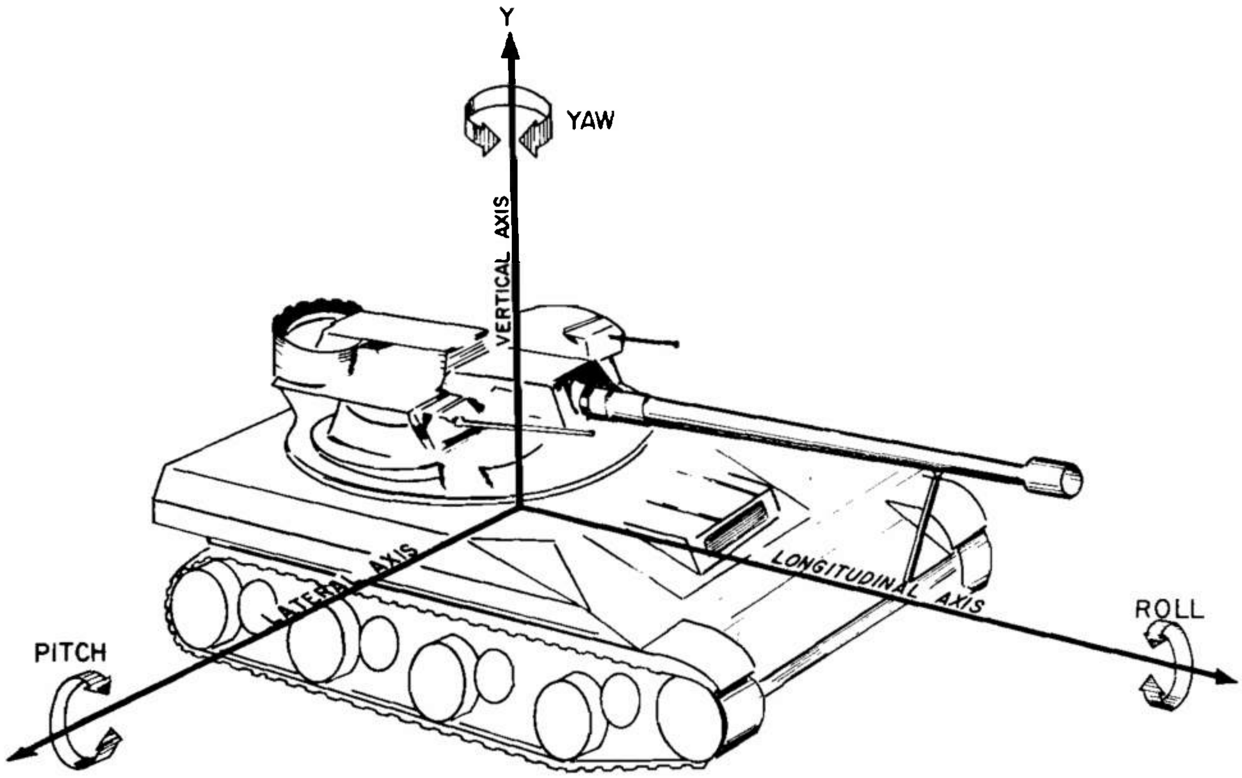


Figure 1-8. Six Degrees of Freedom

tion. It is free to translate longitudinally, laterally, and vertically; and in addition, it is free to rotate about each of these axes, as shown in Figure 1-8. Rotation (angular motion) about these axes is known as *roll, pitch,* and *yaw,* respectively.

A four-wheeled automotive vehicle can be idealized as shown in Figure 1-9. The spring mass is considered a rigid structure, and the unsprung mass is considered as two separate masses—one representing the front axle assembly, including the two front wheels, tires, brakes, and associated linkages, and the other representing the rear axle assembly, wheels, tires, etc. The suspension springs are represented by simple linear springs, the shock absorbers by dashpots, and the tires by linear springs of equivalent spring rate. Tire damping is neglected in this illustration. All of the masses, sprung and unsprung, can experience translational motions along the x-, y-, and z-axes and rotational motions α, β, and γ about these axes. Therefore, 18 coordinates are required to describe the motion of this system completely; or in other words, the system has 18 degrees of freedom.

Systems of many degrees of freedom are much more complicated to deal with analytically than systems of one degree of freedom because such systems have more than one natural frequency. However, it is often only the lowest natural frequency that has practical importance, and an approximate solution can usually be found for this by applying Rayleigh's method (Refs. 35-39).

Another procedure is to take advantage of justifiable simplifying assumptions to reduce the complex system to one that is more easily handled. For example, consider the four-wheeled vehicle idealized in Figure 1-9. It is common in practice to apply the following restrictions to the motions of the masses:

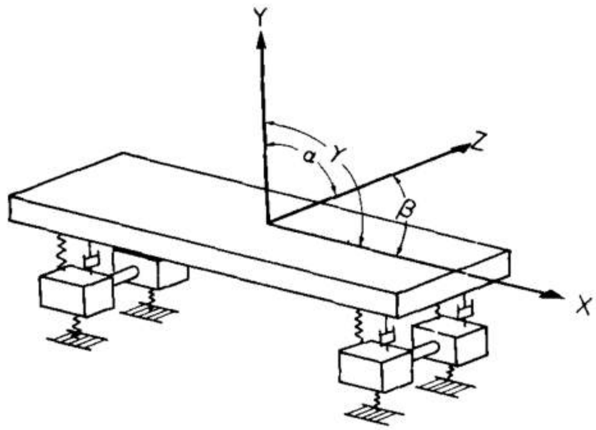


Figure 1-9. Simplified Dynamic Model of an Automotive Vehicle

(a) Relative motion between the sprung and unsprung masses along the x- and z-axes, i.e., the restriction of longitudinal and lateral motions of the masses with respect to each other.

(b) Angular motion of the sprung mass in the α and β coordinates, i.e., roll and yaw.

(c) Angular motion of the unsprung mass in the β and γ coordinates, i.e., yaw and pitch.

This leaves the following motions to be considered when analyzing the dynamics of a wheeled vehicle experiencing shocks and vibrations:

(a) Translation of the sprung mass (body) in the y, or vertical, coordinate with the motion of the unsprung mass (axles) approximately zero in the y coordinate. This vertical oscillation is commonly referred to as *bounce*.

(b) Angular motion of the sprung mass (body) in the γ, or pitch, coordinate with the motion of the unsprung mass (axles) assumed approximately zero in the y coordinate.

(c) Oscillation of the unsprung mass (axles) in the y, or vertical, coordinate while the sprung mass (body) is assumed to remain in a horizontal position. This is the oscillation commonly referred to as *wheel dance* and described earlier in this chapter.

(d) Angular motion of the unsprung mass (axles) in the α, or roll, coordinate while the sprung mass (body) is assumed to remain in a horizontal position.

1-19.8 JERK

Jerk is the term applied to the rate of onset of acceleration and is a significant factor in evaluating human tolerance to acceleration. It is a vector quantity that specifies the time rate of change of acceleration; thus, it is the third derivative of displacement with respect to time.

1-19.9 STABILITY

The term *stability* as used in discussions of the dynamics of automotive vehicles is difficult to define due to its many different applications. Basically, when used in a mechanical sense, stability refers to certain properties of mechanisms that cause them, when disturbed from conditions of static or dynamic equilibrium, to develop forces or moments to restore their original equilibrium condition. Thus, in a vibratory system, damping is a property associated with stability. After a disturbance to such a system with excessive damping present, the system will recover its equilibrium conditions slowly. Response will be sluggish, and the system is said to be overly stable. With insufficient damping, the system will respond quickly but will overcorrect and continue to oscillate for a long time. Successive disturbances may even aggravate the situation. Such a system is obviously unstable. The ideal system is one that achieves maximum speed of response with no overcorrection. This is the condition that exists with critical damping.

In the field of vehicle dynamics there are several factors that tend to create undesirable and even dangerous conditions in the vehicle. These conditions of instability can be minimized and even eliminated through the stabilizing properties of a properly designed system. Among the factors to be considered are: the vertical and angular oscillations of the sprung mass (bounce, pitch, and roll), the high frequency vertical oscillations of the unsprung mass (wheel dance), the tipping tendencies of the vehicle, and various conditions of directional instability.

The vertical and angular oscillations of the sprung mass (bounce, pitch, and roll)—if of the proper frequency and amplitude—can be not only annoying to the passengers but also can cause sickness and physical injury, and structural damage to the vehicle. These effects can be minimized through the careful selection of suspension characteristics, proper damping, the selective location of personnel within the vehicle, and the objective arrangement of vehicle geometry.

Wheel dance, or the high frequency vertical oscillations of the unsprung mass discussed in paragraph 1-19.5, results in three types of instability. It has been mentioned previously that wheel dance was the principal cause of secondary vibrations in a vehicle. If not sufficiently damped, especially if the excitations that cause the wheel dance recur at the frequency of the wheel dance, the shocks transmitted to the main part of the vehicle and its contents will become so severe that the speed of the vehicle will have to be reduced to prevent damage to the vehicle or its cargo. Furthermore, the violent up and down action of the wheels subjects the supporting soil to impact loading. This may impair its load carrying characteristics, resulting in increased rolling resistance and decreased tractive effort. And finally, violent wheel dance reduces steering control. In the case of a wheeled vehicle where wheel dance is occurring in the steerable wheels, steering control may be lost entirely.

Stability against tipping—in both the lateral and longitudinal directions—is an essential characteristic of all vehicles. Apart from the feeling of insecurity that an extremely canted vehicle gives its passengers and the danger of an actual overturn, lateral tipping often results in traction and steering problems produced by the unequal ground loading on the two sides of the vehicle. This is particularly true for tracked vehicles. Similarly, longitudinal tipping results in a reduction or total loss of steering control—if the tipping causes a load decrease on the steerable wheels—and in traction difficulties arising from the redistribution of ground loading.

Tipping tendencies are produced by various moments in the vertical plane that act on the vehicle. These tipping moments are functions of the location of the vehicle center of gravity, the presence of loads that act outside the wheelbase (as in the case of mobile cranes, shovels, etc.), acceleration forces, including those resulting from moving loads or moving equipment mounted on the vehicle (heavy guns, crane booms, etc.), and the slope of the ground supporting the vehicle. The stabilizing factors that act in opposition to the tipping moments are: the wheelbase dimensions, vehicle's moment of inertia, and the elastic properties of the suspension system. A soft suspension system will permit greater angular displacement of the sprung mass, resulting in an aggravated situation. A stiffer suspension, on the other hand, will afford greater opposition to the tipping moments, thereby reducing the angular displacement and improving the stability of the vehicle.

Directional stability refers to the dynamic properties of a vehicle to maintain a given course despite the action of disturbing influences, or to establish a new state of equilibrium if the disturbance remains constant. The chief disturbing influences are side forces due to crosswinds, lateral slopes, centrifugal effects of curvilinear motion, uneven distribution of braking forces, asymmetrical weight distribution, movements of the steered wheels due to a variety of causes, and transient forces resulting from ground irregularities. Stabilizing influences are such factors as the cornering forces developed by the vehicle, the kinematics of the steering linkage, the kinematics of the suspension linkage, the degree of opposition to adverse steering effects afforded by the suspension system, and the location of the center of gravity of the vehicle relative to the action line of the disturbance. Analytical discussion relating these factors can be found in many readily available references that treat the mechanics of vehicles (Refs. 40-43).

The immediate reaction of a directionally stable vehicle to lateral disturbances may be a minor yawing from side to side about the mean line of a new path, but these oscillations will quickly diminish and the vehicle motion will settle down to a steady state condition. A directionally unstable vehicle, under the influence of a disturbing force, will move in a new path that will diverge more and more from the original course; or, if oscillations occur, they will increase in amplitude to dangerous proportions. The driver of a directionally stable vehicle finds it easy to control and keep on its intended course, whereas the driver of an unstable vehicle finds control difficult if not impossible.

1-19.10 HANDLING AND RIDE

The terms *handling* and *ride* are often associated with vehicle stability. *Handling* refers to a general overall quality of a vehicle comprised of a total of subjective impressions of the driver regarding the vehicle's behavior and response to directional commands. It is a subjective evaluation of the vehicle's dynamic stability; and being subjective, it may vary with different drivers.

Similarly, *ride* is a subjective evaluation of the roll, pitch, and bounce characteristics of a vehicle as they affect the comfort of the operator and passengers. It, too, is usually dependent upon the driver's opinion. However, since the vibratory motions that determine ride qualities can he measured and recorded, they could be evaluated against human tolerance limits to these motions, and ride qualities could be evaluated quantitatively. This is not the practice, however, and ride continues to be subjectively evaluated.

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CHAPTER 2

BASIC DESIGN PARAMETERS*

A vehicle designer engaged in designing a new vehicle has two general types of requirements to consider: those related to the desired functional characteristics of the new vehicle and those related to factors over which he has limited or no control. The latter includes such factors as size and weight limitations, physiological and psychological considerations, environmental considerations, transportability requirements, standardized performance characteristics, and maintenance and production considerations. These fundamental design parameters are based upon civil laws, international agreements, and various military directives. A knowledge of these basic design requirements is important to the suspension design engineer as well as to the vehicle designer. A detailed enumeration of these is usually given in the military characteristics prepared for each vehicle to be developed. Some of these—particularly those that affect the suspension design engineer—are given in this chapter to serve as design guides.

SECTION I PHYSICAL LIMITS

2-1 GENERAL DISCUSSION

The primary restrictions placed upon a vehicle design are those dealing with size, weight, and the distribution of the weight upon roads and bridges. Maximum permissible size and weight are usually specified in the military characteristics prepared for each developmental vehicle. These limitations are based upon official agreements and directives, but the military characteristics often limit the physical characteristics further than the maximums specified in general directives. This is particularly true in the case of special purpose vehicles where the specific nature of the vehicle's mission may require additional restrictions. In the general case, however, the physical limits specified are governed by considerations of vehicle transportability.

The policy of the Department of Defense with regard to the transportability of materiel is given in AR 705-8, *Department of Defense Engineering for Transportability Program* (Ref. 2). It directs that transportability should be a major consideration when assigning priorities to the characteristics to be considered in the design of any new item of materiel or equipment. All materiel and equipment developed for use by the military departments must be of such gross weight and outside dimensions as will permit ready handling and movement by available transportation facilities. In general, these are: length 32 ft, width 8 ft, height 8 ft, weight 11,200 lb. Designs that exceed any of these conditions or which require special or unique arrangements of schedules, right-of-ways, clearances, or other operating conditions will be permitted in exceptional cases only after first obtaining approval from the appropriate Transportability Agency. The procedure for requesting this approval is given in AR 705-8.

The following Departmental agencies have been designated as "Transportability Agencies" to implement the *Department of Defense Engineering for Transportability Program*, AR 705-8.

(a) Department of the Army: DCSLOG, Director of Transportation, U.S. Army, Washington, D. C. 20315.

(b) Department of the Navy: Chief, Bureau of Supplies and Accounts (H122), Washington, D. C. 20360.

(c) Department of the Air Force: Headquarters,

* Written by Rudolph J. Zastera of the IIT Research Institute, Chicago, Illinois.

Air Force Systems Command (SCMMT), Washington, D. C.

(d) U. S. Marine Corps: Commandant, Marine Corps (COS), Washington, D. C. 20380.

The dimensions and weight of an item may be adjusted to suit the capabilities of specific modes of transportation when it is known that the item will require no other modes of transportation under peacetime or mobilization conditions. However, due consideration must be given by designers to the following:

(a) Military equipment and materiel is often subjected to movement by several modes of transportation. The factors that govern the choice of mode, or modes, are availability and capability of facilities, destination time requirements, operating conditions, and cost.

(b) Combat and rough terrain equipment is subject to the normal transportation conditions, from manufacturer to the off-the-road destinations, that apply to other equipment.

(c) Transportation systems in overseas areas are generally more restrictive than those found within the continental United States.

(d) The first step toward rapid mobility of military forces is the proper design of equipment to meet the required operational characteristics while adhering to the fundamentals of transportability.

Size and weight restrictions imposed by the various modes of transportation available are specified in the following regulatory documents:

(a) The Highway Weight and Size Limitations, established by the Federal Aid Highway Act of 1956 and a majority of the state laws of continental United States, and the physical limitations of highways in foreign countries to accommodate the potential volume and type of traffic anticipated.

(b) The Outline Diagram of Approved Limited Clearances of the Association of American Railroads referred to in Car Service Rule 14, Section 2 (e), with eight limitations of individual carriers, shown in the current issue of the Railway Line Clearance publication for individual railroads of the United States, Canada, Mexico, and Cuba.

(c) Diagram of the Berne International Rail Interchange Agreement with weight limitations applicable to the railroads of individual countries for all items that may require transportation by rail in foreign countries.

(d) Loading and stowage limitations of ocean vessels, related factors, and Army and Navy procedures therefore.

(e) Regulations of the Department of the Treasury (U.S. Coast Guard), the Army Corps of Engineers, and the Navy covering water transportation.

(f) Regulations and instructions of the Army, Navy, Air Force, Marine Corps, Federal Aviation Agency, and Civil Aeronautics Board for loading cargo and combat aircraft.

As a matter of convenience to the designer, some of the more pertinent data has been extracted and is given in the following paragraphs. More detailed information is given in Refs. 1, 2, and 69.

2-2 PHYSICAL LIMITS—WHEELED VEHICLES

Maximum allowable physical dimensions and weights for pneumatic tired highway and off-the-road types of vehicles intended for unrestricted highway operations are summarized in the paragraphs which follow. Procedures for obtaining permits for oversize, overweight, or other special movements on public highways are covered in AR 55-162. However, vehicles that exceed any one or more of the transportability requirements specified in AR 705-8, and summarized in the preceding section, must be approved by the appropriate Transportability Agency.

2-2.1 WIDTH

Maximum overall width of a wheeled vehicle shall not exceed 96 in. No part of the vehicle, fixtures, or equipment that is attached or placed permanently upon the vehicle shall protrude beyond the outer face of the tires by more than 9 in. on either side of the vehicle.

TABLE 2-1
SIZE AND WEIGHT RESTRICTIONS BY STATES ON TRUCK-TRACTOR-SEMITRAILER COMBINATIONS

	SIZE LIMITS								WEIGHT LIMITS													
			Length (ft)											Vehicle and Combination (1000 lb)							
STATE	Width (in.)	Height (ft)	Straight Truck, Bus, or Trailer	Semitrailer	Tractor & Semi-trailer Comb.	Other Combinations	Number of Trailers Allowed (Semitrailer = ½)	Minimum Tandem Axle Spacing (in.)	Per Inch of Tire Width (lb)	Per Wheel (1000 lb)	Per Single Axle (1000 lb)	Per Tandem Axle, 48-in. Spacing (1000 lb)	Vehicle & Combination Gross Weight Basis	2-Axle Single Unit	3-Axle Single Unit	2-Axle Tractor 1-Axle Semitrailer	2-Axle Tractor 2-Axle Semitrailer	3-Axle Tractor 2-Axle Semitrailer	3-Axle Truck 2-Axle Trailer	Trac., Semi & Trail. 5-Axle Comb.	Maximum Gross Weight
Ala.	96	13¼	40	NS	55	NP	½	40+	NS	NS	18	NS	A	36	54	48	66	73.2	NP	NP	73.2
Alaska	96	13½	40	40	65	65	2	40+	500	NS	18	A32	SA	29	43	47	61	75	76.8	76.8	76.8
Ariz.	96	13½	40	NS	65	65	1½	40+	NS	NS	18	A32	A	36	50	48	62	76	76.8	76.8	76.8
Ark.	96	13½	40	NS	55	55	NS	40	NS	NS	18	32	W	30	44	48	62	73.2	73.2	73.2	73.2
Cal.	96	13½	35	40	60	65	NS	NS	NS	9.5	18	A32	A	36	50	48	62	76	76.8	76.8	76.8
Colo.	96	13½	35	NS	60	60	2	40+	NS	9	18	NS	SF	30	46	48	60	76	76	76	NS
Conn.	102	12½	55	40	55	55	1 or ½	NS	NS	NS	22.4	36	S	32	53.8	53.8	67.4	73	73	NP	73
Del.	96	13¼	40	40	55	60	NS	48	700	NS	20	A36	SA	30	46	48	66	73.2	68	70	73.2
D. C.	96	12¼	40	NS	50	50	1 or ½	40+	NS	NS	22	A38	A	44	60	56	70	70	70	NP	70
Fla.	96	13½	40	40	55	55	1 or ½	40+	NS	NS	20	NS	A	40	60	52	66.6	66.6	66.6	NP	66.6
Ga.	96	13½	55	55	55	55	NS	40+	NS	10.1	20.3	NS	W	40.6	61	52.6	73	73.2	73.2	73.2	73.2
Hawaii	108	13	40	NS	55	65	1½	42	NS	12	24	F32	F	48	56	60	68	72	80	80	NS
Idaho	96	14	40	40	60	65	1½	NS	800	9	18	A32	A	36	50	48	62	76	76.8	76.8	76.8
Ill.	96	13½	42	42	55	60	1½	40+	NS	NS	18	32	SA	36	50	48	62	73.2	73.2	73.2	73.2
Ind.	96	13½	36	NS	55	65	2	40+	800	NS	18	32	W	36	50	48	62	72	72	72	72
Iowa	96	13¼	35	NS	55	60	1½	40	NS	NS	18	A32	A	36	50	32	62	72.6	72.6	72.6	73.2
Kan.	96	13½	42½	42½	55	65	NS	40+	NS	9	18	32	A	36	50	48	62	73.2	73.2	73.2	73.2
Ky.	96	13½	35	NS	55	NP	½	42	600	NS	18	32	W	36	50	48	62	73.2	NP	NP	73.2
La.	96	13½	35	NS	55	60	1 or ½	40+	450	NS	18	32	W	36	41	45	59	73	77	NP	NS
Maine	102	13¼	55	NS	55	55	1 or ½	48	600	NS	22	36	SA	32	51.8	51.8	66.3	73.2	73.2	NP	73.2
Md.	96	13¼	55	55	55	55	NS	48	NS	NS	22.4	40	SA	44.8	55	55	65	73.2	73.2	73.2	73.2
Mass.	96	NS	35	NS	50	NS	1 or ½	NS	NS	NS	22.4	36	A	44.8	55	56.8	70.4	73	51	NP	73
Mich.	96	13½	35	40	55	55	1½	42+	700	NS	18	26	W	36	44	48	62	73.2	80	84	NS
Minn.	96	13½	40	40	50	50	1 or ½	40+	NS	9	18	32	A	36	50	48	62	73.2	73.2	NP	73.2
Miss.	96	13½	35	NS	55	55	1 or ½	40+	NS	9	18	A32	A	36	50	48	62	73.2	73.2	NP	73.2
Mo.	96	13½	40	NS	55	65	NS	40+	600	NS	18	32	A	36	50	48	62	73.2	73.2	73.2	73.2
Mont.	96	13¼	35	NS	60	60	1 or ½	40+	NS	NS	18	A32	A	36	50	48	62	76	76	NP	76.8
Nebr.	96	13½	40	40	60	65	2	40+	NS	9	18	A32	A	36	50	48	62	71.1	71.1	71.1	71.1
Nev.	96	NS	NS	NS	NS	NS	NS	NS	NS	NS	18	A32	A	36	50	48	62	76	76.8	76.8	76.8
N. H.	96	13½	35	NS	55	55	NS	NS	600	NS	22.4	36	SA	33.4	55	52.8	66.4	73.2	66.4	66.4	73.2
N. J.	96	13½	35	NS	55	55	2	40	800	NS	22.4	32	W	44.8	54.4	56.8	66.4	73.2	73.2	73.2	73.2
N. M.	96	13½	40	NS	65	65	1½	40+	600	11	21.6	A34.3	A	43.2	55.9	55.2	67.9	80.6	86.4	86.4	86.4
N. Y.	96	13	35	NS	55	55	1 or ½	46	800	11.2	22.4	36	F	44.8	58.4	56.8	70.4	71	71	NP	71
N. C.	96	13½	35	NS	55	55	1 or ½	48+	600	9	18	NS	S	30	47.5	47.5	64	70	70	NP	70
N. D.	96	13¼	40	NS	60	60	1½	40+	550	9	18	32	F	36	50	46	62	64	64	64	64
Ohio	96	13½	35	40	55	60	NS	48+	650	NS	19	24	F	38	51	50	63	76	78	78	78
Okla.	96	13½	40	NS	55	65	1½	40+	650	NS	18	A32	A	36	50	48	62	73.2	73.2	73.2	73.2
Ore.	96	13½	35	35	50	65	1½	40	550	9	18	32	A	36	50	48	62	73.2	76	76	76
Pa.	96	13¼	35	40	55	55	1 or ½	36	800	NS	22.4	36	S	33	47	50	60	71.1	71.1	NP	71.1
R. I.	102	13½	40	NS	55	55	1 or ½	40+	NS	NS	22.4	36	S	36	44	53.8	67.4	73.2	80	NP	NS
S. C.	96	13½	40	NS	55	55	1 or ½	40	NS	10	20	36	S	32	46	50	65	73.2	73.2	NP	73.2
S. D.	96	13½	35	NS	60	60	1½	40	600	9	18	A32	A	36	50	48	62	73.2	73.2	73.2	73.2
Tenn.	96	13½	35	NS	50	50	1 or ½	40	NS	NS	18	32	W	36	50	48	62	73.2	47.5	NP	73.2
Texas	96	13½	40	40	55	65	NS	40	650	9	18	32	A	36	50	48	62	72	72	72	72
Utah	96	14	45	45	60	60	NS	40+	NS	9.5	18	A33	A	36	51	48	63	78	79.9	79.9	79.9
Vt.	96	13½	55	NS	55	55	1 or ½	48	600	NS	22.4	36	A	44.8	58.4	56.8	70.4	73.2	73.2	NP	73.2
Va.	96	13½	35	NS	50	50	1 or ½	40+	650	NS	18	32	A	36	50	48	62	70	70	NP	70
Wash.	96	13½	35	40	60	65	1½	42	550	NS	18	A32	SA	28	36	46	60	73.2	76	NP	76
W. Va.	96	13½	40	NS	55	55	1 or ½	40+	NS	NS	18	A32	A	36	50	48	62	70	70	NP	70
Wis.	96	13½	35	NS	55	55	1 or ½	42+	NS	11	19.5	A32	A	39	51.5	51	63.5	73	73	NP	73
Wyo.	96	13½	40	NS	65	65	2	40+	NS	9	18	A32	A	36	50	48	62	73.9	73.9	73.9	73.9

VEHICLE AND COMBINATION weight limits are based on 48-in. tandem spacing (except North Carolina and Ohio—over 48 in.), overall axle spacing 5 ft. less than maximum overall length, and a 12,000-lb front axle loading on combinations—unless otherwise limited. Except as noted, trailer includes a semitrailer with converter dolly.

NOT INCLUDED are tolerances and special provisions for overweight or oversize vehicles or loads, farm vehicles, specific commodities, special types of vehicles or drive-away-towaway operation.

TANDEM AXLES spaced less than minimum shown are subject to single axle weight limits. A "+" sign means "more than."

TIRES inflated 100 psi or more are considered "high pressure" and under 100 psi "low pressure."

LEGEND

A—States where vehicles and combination gross weights are controlled by a table of axle spacing up to a specified maximum. Where tandem axle weights are similarly limited, tandem limit shown is for 48-in. spacing.

F—States where vehicles and combination gross weights are controlled by a "bridge" formula. Where tandem axle weights are similarly limited, tandem limit shown is for 48-in. spacing.

S—States where vehicle and combination gross weights are controlled by maximum limits for specific types.

W—States where vehicles and combination gross weights are controlled by axle weight limits shown up to, in most states, a specified maximum.

NP—Not permitted.

NS—Not specified.

2-2.2 HEIGHT

The maximum overall height of wheeled vehicles designed for travel over highways or roads in the continental United States shall not exceed 150 in. (12 ft 6 in.). The maximum permissible height for wheeled vehicles designed for overseas highway operations shall not exceed 132 in. (11 ft 0 in.).

2-2.3 LENGTH

The maximum overall length of a wheeled vehicle comprised of a single, nonarticulated unit shall not exceed 35 ft. The maximum overall length of a wheeled vehicle comprised of an articulated double unit, such as a truck-tractor coupled to a semitrailer or a truck-tractor coupled to a full trailer, shall not exceed 50 ft. These dimensions exceed the statutory limitations for unrestricted highway movement in certain states in the continental United States. In these states, a special permit is required. Procedures to be followed in obtaining these permits are given in Ref. 3. Table 2-1 shows that the size and weight limitations imposed by the various states on truck-tractor-semitrailer combinations as of August 30, 1959.

2-2.4 AXLE LOADING

An axle load is defined as the total load transmitted to the road by all wheels whose centers are included between two parallel transverse vertical planes 40 in. apart, extending across the full width of the vehicle. The maximum axle load (subject to gross weight limitations) is 16,000 lb for axles located from 3½ to 7½ ft from the nearest adjacent axle. This applies to both the continental United States and overseas areas. For axles located more than 7½ ft from the nearest adjacent axle, the maximum permissible load is 18,000 lb for vehicles designed for highway operation in the continental United States, and 16,000 lb for vehicles designed for overseas operations.

2-2.5 GROSS WEIGHT

The maximum permissible gross weight of a wheeled vehicle designed for highway operations (subject to axle load limitations) shall not exceed 36,000 lb, for vehicles having a distance of 10 ft or less between the extreme front and rear axles, and shall increase by 850 lb for each additional foot of extreme axle spacing in excess of 10 ft to a maximum gross weight of 60,000 lb. This maximum gross weight exceeds the statutory limits of some states within the continental United States (see Table 2-1). For operations in these states, a special permit is required (see Ref. 3).

2-3 PHYSICAL LIMITS—TRACKED VEHICLES

2-3.1 GENERAL LIMITS

Limiting dimensions for tracked vehicles in accordance with their gross weights are given in

TABLE 2-2
LIMITING DIMENSIONS AND WEIGHTS OF TRACKED VEHICLES FOR MOVEMENT ON HIGHWAYS AND BRIDGES

Maximum Gross Weight, lb	Width		Maximum Height, in.	Minimum Ground Contact	
	Maximum, in.	Minimum, in.		Length,* in.	Total Width, in.
8,000	96	None	132	32	20
16,000	96	78	132	55	24
24,000	96	80	132	73	27
32,000	96	84	132	87	30
40,000	120	96	132	98	33
48,000	120	100	132	107	36
60,000	120	100	132	132	37
80,000	120	112	132	144	45

*The maximum ground-contact length for any vehicle is 180 in.

Table 2-2. These limits are necessary for unrestricted operations on highways, in both the continental United States and in overseas areas, and are chiefly governed by bridge widths. The values given for width of ground contact are the total widths of all ground-contacting elements; e.g., for a conventional vehicle having two tracks, the width of ground contact equals twice the width of one track.

It should be noted that the values given in Table 2-2 apply to tracked vehicles that are to be capable of unrestricted movement on highways and bridges. When vehicles that are being designed will exceed these specifications, they must be approved by the appropriate departmental Transportability Agency (paragraph 2-1 and Ref. 3).

2-3.2 HEIGHT AND GROUND CLEARANCE

The *maximum height* of 132 in. given in Table 2-2 is based upon clearance requirements encountered in operations outside the continental United States. Because of the requirement to minimize the vehicle silhouette height in order to reduce vulnerability to enemy action, this limit is seldom reached.

The *minimum height* of a tracked vehicle is greatly affected by the ground clearance specified in the military characteristics. A generous ground clearance reduces the danger of the vehicle bellying in soft ground or when straddling an obstacle, provides space for the installation of escape hatches in the belly, and provides clearance for the high flanges found on the treads of military floating bridges. The minimum ground clearance of tracked vehicles has been about 17 inches for a long time. In recent designs, however, attempts are being made to increase this as much as possible without penalizing the overall height unduly. The latest M60 medium tank has a ground clearance of 20⅝ inches, and a similar tank with an experimental suspension system can vary its ground clearance from 6 to 25 inches at the will of the driver.

2-3.3 GROSS AND DISTRIBUTED WEIGHTS

The maximum permissible gross weight of a tank is 160,000 lb (80 tons). This limitation is based upon the capacity of U. S. highway bridges of the heaviest classification (Ref. 4), and tanks of this weight are not capable of unrestricted movement on highways and bridges.

Minimum ground contact dimensions are given in Table 2-2. These are of importance as they affect the distribution of the gross weight. Two considerations affect the distribution of the gross weight of a tracked vehicle: one, the average ground pressure, determined as the quotient of the gross weight and the total ground-contact area of the tracks; the other is the load distributed per linear foot of the length of ground contact. Ground pressures are regulated primarily to control flotation in cross-country operations and to develop traction, while loading per foot of track length is regulated to control the effect of the vehicle's weight upon roads and bridges.

Ground pressures of heavy tracked vehicles are limited to 12.5 psi. Although pressures of 6 to 8 psi are considered more desirable, they are difficult to obtain in the design of larger vehicles. Certain light, amphibious and special purpose tracked vehicles that require maximum flotation obtain ground pressures as low as 2½ psi.

The distributed load per linear foot of ground contact is obtained by dividing the gross vehicle weight, GVW in pounds, by the length of the ground contact, L_t in feet, as seen in a side elevation. The maximum permissible distributed load is determined from the following (Ref. 4):

For values of GVW less than 60,000 lb:

$$\frac{GVW}{L_t} \leq 3000 + 0.06\,(GVW - 8000) \qquad (2\text{-}1)$$

For values of GWV greater than 60,000 lb:

$$\frac{GVW}{L_t} \leq \frac{20{,}000 \times GVW}{160{,}000 + GVW} \qquad (2\text{-}2)$$

2-3.4 VEHICLE LENGTH

The length of the track at the region of ground contact is governed by the equations of weight distribution, the ground contact pressure, and by certain requirements for efficient steering. Requirements imposed by specified angles of approach and departure have an influence upon overall vehicle length. Equations 2-1 and 2-2 for maximum permissible distributed weight can be solved for L_t to determine the minimum permis-

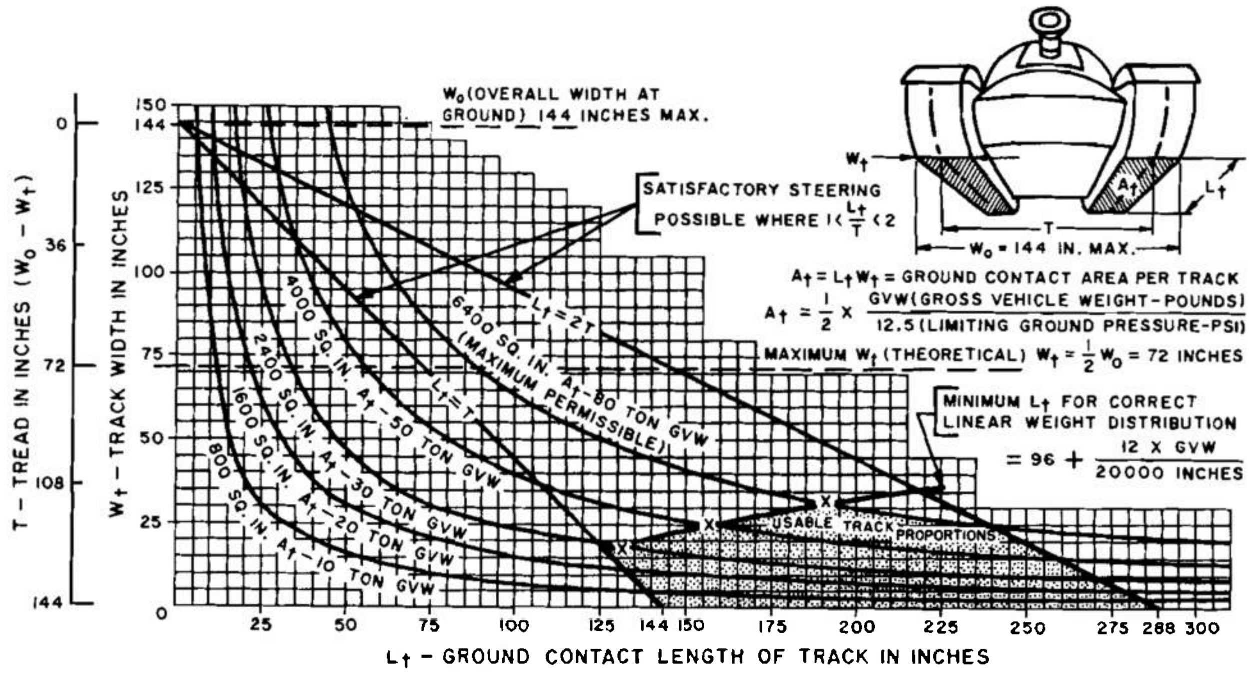

Figure 2-1. Correlation of Physical Limits of a Tracked Vehicle (Ref. 4)

sible track length in ground contact. Thus, when GVW is less than 60,000 lb:

$$L_t \text{ (inches)} \geq \frac{12\,GVW}{3000 + 0.06\,(GVW - 8000)} \quad (2\text{-}3)$$

when GVW is greater than 60,000 lb:

$$L_t \text{ (inches)} \geq 96 + \frac{12\,GVW}{20{,}000} \quad (2\text{-}4)$$

The steering characteristics of a tracked vehicle are affected by the ratio of the track length in contact with ground L_t to the tread L_t/T, where T is the width between the track centers. When this ratio becomes less than unity, i.e., L_t is less than T, steering becomes relatively unstable. When this ratio approaches a value of 2, i.e., L_t approaches a value of $2T$, steering imposes excessive power demands. Therefore, for satisfactory steering, the L_t/T ratio should be between 1.0 and 1.7. In actual practice, the values usually used are between 1.125 and 1.69 (Ref. 4).

2-3.5 CORRELATION OF PHYSICAL LIMITS

When dimensional limits are applied simultaneously, a relatively narrow field of choice is available to the designer. This is illustrated in Figure 2-1 which shows the track proportions that may be used lying in a relatively small zone. In the figure, the ground-contact length of the track L_t is plotted against the track width W_t and against the tread T. A theoretical maximum overall track width of 144 inches is used as a limit in order to establish $T = 0$ on the graph. Hyperbolas representing constant track areas are plotted on the graph using the equation $A_t = L_t W_t$ (where A_t is the ground-contact area) for selected ground-contact areas of 800, 1600, 2400, 4000, and 6400 sq inches. Based upon a limiting ground pressure of 12.5 psi, each hyperbola also represents a definite gross vehicle weight. The highest gross vehicle weight shown is 80 tons, the maximum permitted by regulations. Points of minimum track length are calculated, using Equations 2-3 and 2-4, and located on the hyperbolas. These points are connected to form a limiting minimum track length curve. Finally, the limiting steering curves ($L_t = T$ *and* $L_t = 2T$) are determined and drawn.

The usable track proportions are confined to the hatched area in the graph bounded by the limiting dimension curves. The chief significance of the usable proportions thus obtained lies in their effect on hull width. Decreasing the track width of a given vehicle gains only a small amount of space within the vehicle because the track width changes relatively little regardless of length, within the usable range of proportions.

One of the most critical dimensions on a combat tank is the turret ring diameter. This must be large enough to allow space for such major items as gun recoil, ammunition handling, personnel, and fire control equipment, yet its size is restricted by the overall vehicle width limits, the need for adequate track width, and the need for armor protection. Here the designer is faced with a necessity to compromise, but his freedom to compromise is somewhat limited.

Turret rings currently being used on production vehicles are 85 inches inside diameter. Figure 2-1 shows that for a 50-ton vehicle, the designer can select a track width of from 15 to 25 inches. For most efficient steering he would probably restrict his choice to track width between 22 and 25 inches, letting considerations of ground-contact length determine the final choice. On the assumption that the 22 inches track width satisfied all requirements, the inside diameter of the turret ring plus the width of two tracks account for 129 inches of vehicle width. Additional width is needed for armor, track shrouding, width of turret bearing, and clearances. Thus, requirements to reduce the overall vehicle width present serious problems to the designer.

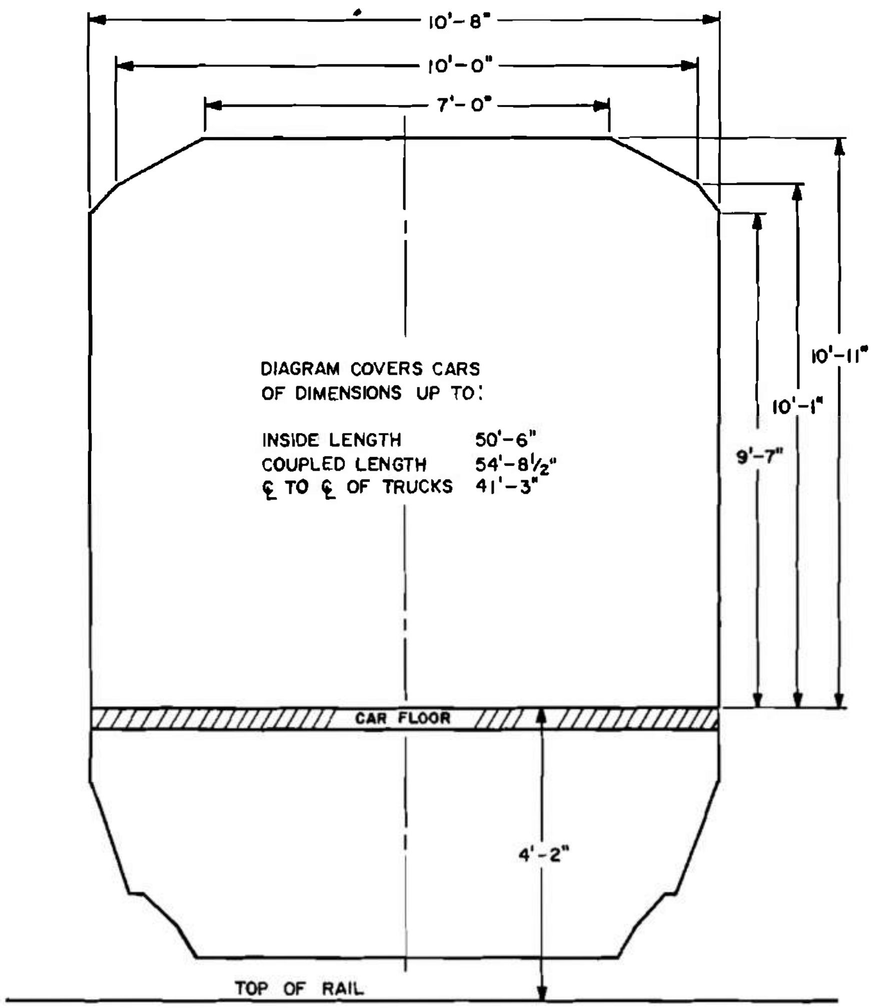

Figure 2-2. Outline Diagram of Approved Limited Clearances of the Association of American Railroads

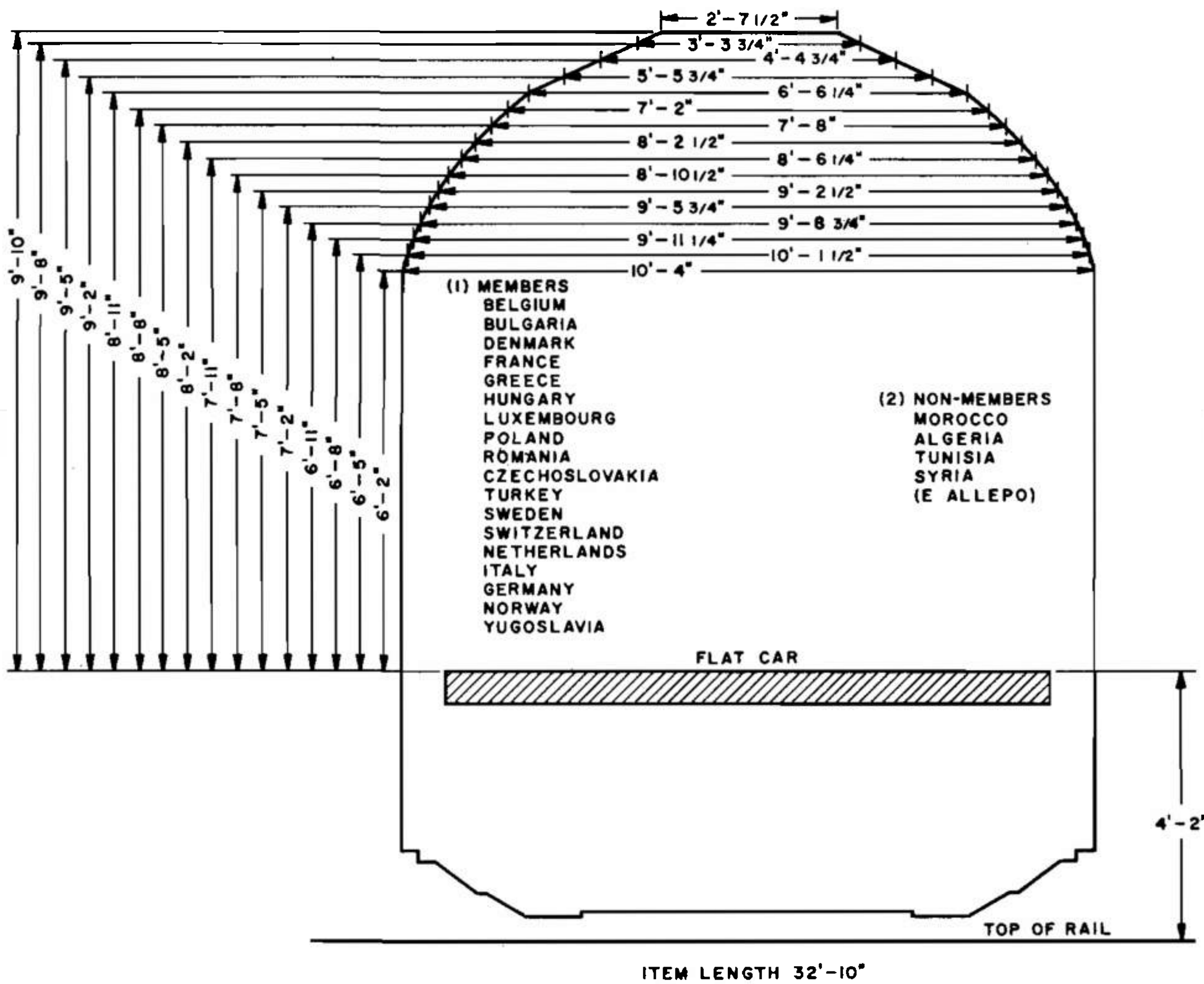

Figure 2-3. Berne International Clearance Diagram

2-4 PHYSICAL LIMITS IMPOSED BY RAIL TRANSPORTABILITY REQUIREMENTS

In order to meet the requirements for transportability by railroad, railway lading clearances must be considered. Railway lading clearances are determined by bridges, tunnels, platforms, telephone and electric poles, and miscellaneous wayside structures. Data on all such clearances have been compiled and are presented by the railroads in the form of dimensioned outline diagrams. Railroad car loads whose outlines exceed these limiting outlines must be cleared by the superintendent of the railway lines prior to their acceptance. Figure 2-2 shows the outline diagram of the approved limited clearances published by the Association of American Railroads and applies to all standard gauge unrestricted main lines in the continental United States. Figure 2-3 shows a comparable outline diagram that prescribes the minimum railway clearances required on railroads in overseas areas. It is the result of an international meeting held at Berne, Switzerland and is, therefore, often referred to as the Berne International Tunnel Diagram.

These diagrams indicate the maximum allowable cross section of a vehicle as loaded upon a railway car for shipping. For a vehicle of larger section, the maximum allowable cross section is the size to which the section must be reducible. An important factor in reducibility is the facility with which the vehicle can be reassembled in terms of time, tools, and skills required. It is current practice to use overhanging tracks on many types of tracked vehicles. By removing tracks, sprockets and tool boxes, vehicle widths can be reduced approximately 20 inches—or to the width across the outer faces of the road wheels. Further reducibility requires the disassembly of suspension elements, and is not considered feasible. A more comprehensive coverage of this subject is given in Refs. 1, 2, and 69.

SECTION II HUMAN FACTORS

2-5 GENERAL DISCUSSION

One of the many functions of the suspension system is to provide a reasonable degree of ride comfort for the vehicle's passengers and crew. The high degree of refinement in ride quality found in commercial passenger vehicles is not required of military vehicles, but a knowledge of comfort criteria is important to the suspension system designer. When operating cross-country over rough terrain or on irregular road surfaces, the operator and crew are subjected to severe jostling, vibrations and impacts. Aside from physical discomfort, they find it difficult to perform their assigned tasks. The driver has difficulty in controlling the vehicle due to the mechanical stresses being experienced by his body, his vision becomes impaired, and physical discomfort causes him to reduce vehicle speed to a more acceptable level. Thus, maximum mobility, the ultimate goal of the military vehicle designer, is compromised by the physiological tolerance levels of the human occupant.

The mechanical stresses to which the human body is exposed in a vehicle are produced by the accelerations and vibrations experienced by the vehicle. Exposure to these stresses might be continuous over a period of time or might occur only in certain situations. Furthermore, it might affect only certain organs of the body, or it might produce general impairment and damage to body structures and processes. A great deal of research has been done in this field, particularly as it applies to advanced types of manned aerospace vehicles, and a great deal of literature exists on this subject (Refs. 5-16, 70). Unfortunately, the differing techniques and objectives of the various experimenters make their results difficult to compare.

Early work in acceleration testing lacked sophistication and standardization. A high incidence of physical discomfort and injury was reported at relatively low values of acceleration, but was actually due to inadequate body support. Data recorded in experimentation with human (and animal) subjects are often open to question because of the instrumentation procedures used. Accelerometers, for example, are extremely sensitive to their immediate environment, making rigid mounting and attachment essential to obtaining accurate data. In many cases, an instrument will record the response of its mount rather than the gross accelerations of the vehicle. In other cases, resonant vibrations are set up which give erroneous data. Accelerations measured directly on the human (or animal) body should be highly suspect because of the difficulty of achieving rigid mounting for the sensors.

Considerable experimental data are based upon animal testing. The animals ordinarily chosen bear a degree of similarity in body structure to humans. Chimpanzees, bears, goats, and hogs have been used. Many factors exist, however, which preclude the direct application of information obtained in this manner to the establishment of human tolerance limits. Since the subjects must be considered as uncooperative, a true evaluation of the restraint-support system used is difficult to obtain. Furthermore, despite the similarities between animal and human subjects, anatomical and physiological differences do exist; therefore, due caution must be exercised in using information obtained from animal experiments as applicable to man.

2-6 ACCELERATION EFFECTS

2-6.1 ACCELERATION TERMINOLOGY

Acceleration is a vector quantity defined as the time rate of change of velocity and is, therefore, the first derivative of velocity and the second derivative of displacement with respect to time. In human factors considerations, it is usually compared to the acceleration produced by the normal force of gravity (32.2 ft per sec per sec), and is expressed as an acceleration of so many g's ($n\ g$). Since the force required to accelerate an object is the product of the mass and the desired acceleration ($F = MA$), expressing the acceleration in terms of g results in a simple comparison of the force magnitude with respect to the object's weight W.

$$M = \frac{W}{g}$$

$$F = Ma = \left(\frac{W}{g}\right)(n\ g) \qquad (2\text{-}5)$$

$$F = n\ W$$

Thus, an acceleration of 25 g, for example, requires a force equal to 25 times the weight of the body being accelerated. Furthermore, and most important in human factors considerations, the response of the human body to accelerations is determined by the reactive forces developed within the body as a result of the body's inertia and the inertia of its constituents. These reactive forces are equal in magnitude, but opposite in direction, to the acceleration force. Thus, in the 25 g example just cited, the body will react as if its weight, and the weight of its components, were increased 25 fold.

A certain amount of caution must be exercised when discussing accelerations, acceleration forces, and reactive forces to prevent confusion and misunderstanding. Many systems have been used to describe the direction of acceleration vectors. These differ mainly in the directional references upon which they are based. Some use vehicular displacement for reference while others use the human body as reference. The latter is perhaps most useful to human factors considerations. It describes acceleration directions with respect to directions parallel and perpendicular to the longitudinal axis of the body. Since any number of vertical planes can be passed through the longitudinal axis, only two are pertinent. One passes through the shoulders to define the *lateral plane*, and the other through the chest and back to define the *transverse plane*. Accelerations in the longitudinal plane are referred to as *headward* or *footward* (sometimes *tailward*); accelerations in the lateral plane are referred to as *right lateral* or *left lateral*; and accelerations in the transverse plane are referred to as *forward* or *backward* (also *sternumward* or *spineward*). This terminology applies regardless of body orientations.

Recently, the need for simplification and assurance of positive identification of the physiological axis of g loading has lead to the development and general adoption of a system of vernacular expressions used by test pilots and design engineers. The system has as its basis the inertial response of the ocular bulb (eyeball) to applied accelerations, and has been recommended for general international use by the Acceleration Committee of the Aerospace Medical Panel, Advisory Group for Aeronautical Research (Ref. 17). This system associates a physical sensation with the acceleration directions, thus promoting a better understanding of the associated loadings. With this system, headward and footward accelerations in the longitudinal plane are *eyeballs-down* and *eyeballs-up* accelerations, right lateral and left lateral accelerations in the lateral plane are *eyeballs-left* and *eyeballs-right* accelerations, and forward and backward accelerations in the transverse plane are *eyeballs-in* and *eyeballs-out* accelerations, respectively.

The foregoing discussion of acceleration directions is primarily related to *linear acceleration*, i.e., an acceleration in which the position of the longitudinal axis, and thus the whole body, moves in a straight line. Two other types of acceleration exist; namely, *radial* or *centrifugal* and *angular* acceleration.

Radial or *centrifugal* acceleration is produced when a body is rotated about an axis passing through it perpendicular to its longitudinal axis so that end-over-end tumbling occurs.

Angular acceleration is experienced during rotary and curvilinear motion when the angular velocity about some central point is varying with respect to time. In human factors studies, this term is usually applied to rotations of the body about the longitudinal axis (spinning), or to circular movements wherein the longitudinal axis is rotated about some externally located center of rotation. In the latter circumstance, the acceleration force is directed toward the center of rotation, resulting in physical responses similar to those experienced during linear accelerations. In addition, however, certain visual, auditory, and postural illusions often occur along with physiological disturbances. These are discussed later in this section.

In most practical systems, particularly those involving a human riding across rough terrain in a ground vehicle, two or all three types of acceleration are usually present. One type usually predominates to such an extent, however, as to permit neglecting the other two. It is convenient, therefore, to study the response of the body to one type of acceleration at a time.

2-6.2 LINEAR ACCELERATION

Linear acceleration is usually divided into three categories according to their duration and the physiological effect that accelerations of different duration produce. These categories are

(a) *Abrupt.* Accelerations of less than 2 seconds duration.

(b) *Brief.* Accelerations of from 2 to 10 seconds duration.

(c) *Prolonged.* Accelerations of more than 10 seconds duration.

Accelerations in the brief category are of only minor interest to the automotive designer, and those in the prolonged category hold only an academic interest. These two categories are, therefore, treated only briefly in this discussion. In general, duration and magnitude of an acceleration force that can be tolerated are inversely related. The shorter the duration, the higher the tolerance level. Table 2-3 shows examples of various common acceleration situations giving their duration and magnitude.

2-6.2.1 Abrupt Accelerations

This type of acceleration is experienced in falls, automobile or aircraft crashes, parachute openings and landings, seat ejections for escape from high-speed aircraft, during bottoming or "bump-out" of an automotive suspension system, and many other situations. During the brief time period involved, tissues react by structural damage or failure; therefore, interest in the body's response to these acceleration forces is chiefly centered on mechanical stress limits. The injuries that occur most often are bruises, tissue crushing, bone fracture, rupture of soft tissues and organs, and concussions.

When soft tissues are displaced considerably by appropriate forces, rupture of membranes or organ capsules may take place resulting in so-called internal injuries. Such injuries, in practice, are usually produced by forces of longer duration and are usually dangerous. The rate of onset of the acceleration is also an important factor in determining human tolerance limits. An acceleration force applied over an infinitesimally short period of time will produce a deformation strain, and corresponding stress, twice as great as would be produced by a force of equal magnitude applied statically.

The limits of human tolerance to abrupt accelerations involve the following factors:

(a) The degree of body support and restraint.

(b) The orientation of the body with respect to the direction of the force.

(c) The rate of onset and duration of the force.

(d) The magnitude of the force.

The limits of human tolerance to abrupt accelerations are shown in Figures 2-4 through 2-7 (Ref. 8). The limits indicated are based upon situations involving healthy, young subjects that were expecting the application of the forces involved and were provided with maximum body support or restraint.

Biologically, the limiting factor in human tolerance to headward, or eyeballs-down, acceleration in the normal seated position is fracture of the spine in the upper lumbar region just above the concavity commonly known as the "small of the back." With optimum body alignment, up to 35 *g*

TABLE 2-3
APPROXIMATE DURATION AND MAGNITUDE
OF SOME COMMON ACCELERATIONS (Ref. 10)

Vehicle	Type of Operation	Acceleration Magnitude, g	Acceleration Duration, sec
Elevator	Average in "fast service" Comfort limit Emergency deceleration	0.1–0.2 0.3 2.5	1–5
Automobiles	Comfortable stop Very undersirable Maximum obtainable Crash (potentially survivable)	0.25 0.45 0.7 20–100	5–8 3–5 3 <0.1
Train	Normal acceleration and deceleration Emergency stop from 70 mph	0.1–0.2 0.4	5 2.5
Aircraft	Ordinary take-off Catapult take-off Crash landing (potentially survivable) Seat ejection	0.5 2.5–6 20–100 10–15	>10 1.5 0.25
Man	Parachute opening at 40,000 ft Parachute opening at 6,000 ft Parachute landing Fall into fireman's net Approximate survival limit with well distributed forces (fall into deep snowbank)	33 8.5 3–4 20 200	0.2–0.5 0.5 0.1–0.2 0.1 0.015–0.03

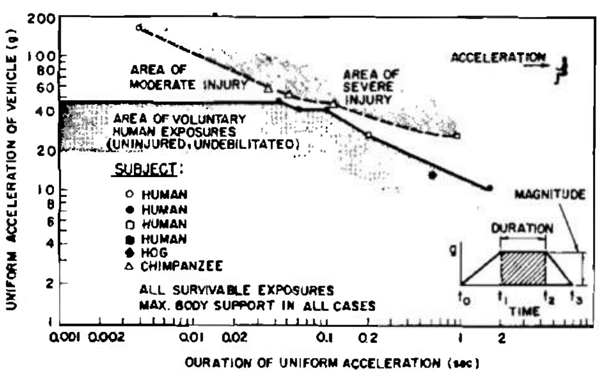

Figure 2-4. Tolerance to Eyeballs-Out Acceleration (Ref. 8)

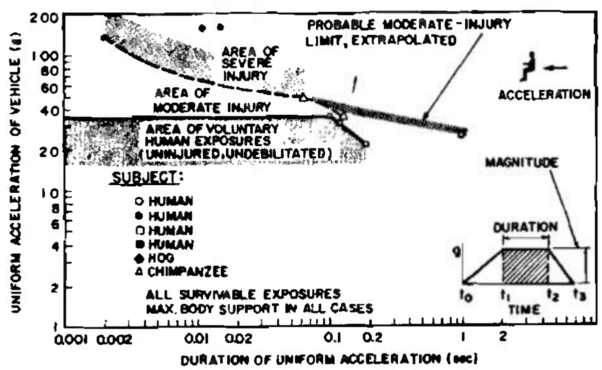

Figure 2-5. Tolerance to Eyeballs-In Acceleration (Ref. 8)

can be tolerated at a rate of onset of less than 500 *g* per second; but, with the back bent forward to the limit of motion, this limit diminishes to less than 15 *g*.

Acceleration in the tailward, or eyeballs-up direction have not been tested to tolerance limits with human subjects. However, 10 to 12 *g* is tolerable for periods of 0.003 to 0.3 seconds at rates of onset between 10 and 82 *g* per second.

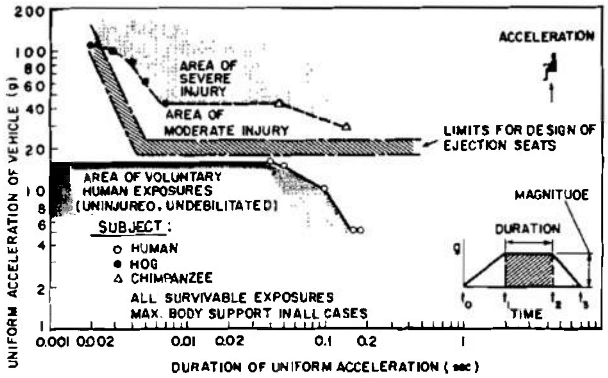

Figure 2-6. Tolerance to Eyeballs-Down Acceleration (Ref. 8)

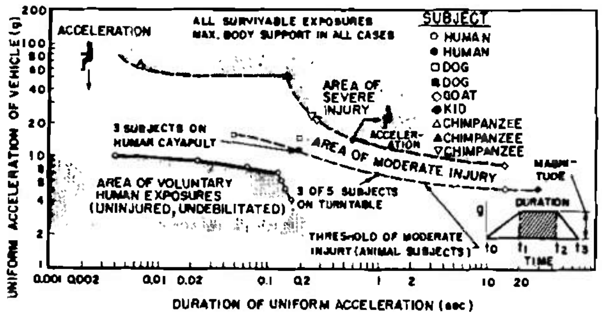

Figure 2-7. Tolerance to Eyeballs-Up Acceleration (Ref. 8)

When exposed to abrupt transverse (eyeballs-in-or-out) accelerations, the subject reacts somewhat like a boxer does to a blow. At a threshold of 30 *g* with an onset rate of 1,000 *g* per second, the subject experiences a brief period of shock manifested by lowered blood pressure and weakness. At 40 *g* and an onset rate of 1,500 *g* per second, he loses consciousness.

The structure of the neck is rather well adapted to support the head. Injury to the neck due to abrupt transverse acceleration seems to occur only upon backward flexion and extension of the neck when the body is accelerated from back to front without the benefit of a head support. This gives rise to the so-called "whip-lash" injury so common in rear-end automobile collisions.

2-6.2.2 Brief Accelerations

Brief accelerations are categorized as those whose duration lasts from 2 to 10 seconds. At a duration of about 2 seconds, so-called hydraulic effects begin to take place in the body. These consist of the displacement of fluids in the body cells, changes in the hydrostatic equilibrium between small blood vessels and their surrounding body fluids, changes in permeability and structural failure of blood vessels, and the mechanical displace-eral orientation of the body with respect to the ment or deformation of body structures. The gen-direction of acceleration determines the physiological effect of the hydraulic displacements.

With headward (eyeballs-down) acceleration, body fluids and organs are displaced downward, i.e., away from the head. Oxygen-carrying blood is unable to reach the brain and pools in the lower portions of the body. Vision fails before consciousness, because arterial blood must overcome a normal presure of about 28mm Hg in the eyeballs, whereas the opposing pressure inside the skull is negligible. Time limits for tolerance of these conditions are set by the occurrence of visual blackout and unconsciousness. These are the hypoxic effects shown in Figure 2-8. Total cessation of blood flow to the brain causes unconsciousness in about 7 seconds and irreparable brain damage in 2 to 3 minutes.

During tailward (eyeballs-up) acceleration, body fluids and organs are displaced upward, i.e., toward the head. Tears fail to drain from the eyes causing an impairment of vision. The increased

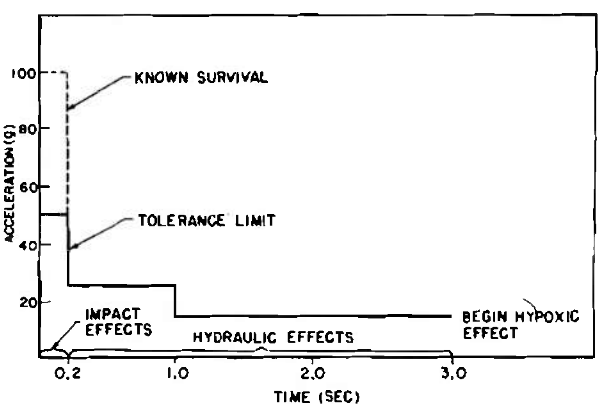

Figure 2-8. Hypoxic Effects During Eyeballs-Down Acceleration

blood and pressure on the brain and eyes cause confusion, pain, and hemorrhage. About 3 to 5 g of tailward acceleration can be tolerated for periods up to 30 seconds.

Tolerance limits for transverse accelerations in the seated position are set by hemorrhage into the eyes, retinal detachment, chest pain, and difficulty in breathing. The following acceleration forces are tolerable: 3 g for 900 seconds, 10 g for 120 seconds, 15 g for 5 to 50 seconds, and 25 g for 1 second.

2-6.2.3 Prolonged Accelerations

Accelerations whose duration lasts for more than 10 seconds fall into this category. After the body experiences severe acceleration forces for about 5 seconds, it begins to respond physiologically with compensatory reflexes. These are chiefly in the form of increased heart rate and force, to help circulate the blood that would otherwise pool peripherally, and by constricting blood vessels to reduce peripheral flow and to reduce the escape of blood through vessel walls. These compensatory reflexes become fully active in 15 to 20 seconds and may persist for many minutes or even hours if the acceleration forces are not too great. If the acceleration forces are excessive, the compensatory reflexes will fail, marking the subject's time limit for tolerating that particular level of acceleration. Situations involving prolonged acceleration are of vital interest to the designer of aerospace vehicles, but they are of only academic interest to the designer of contemporary automotive land vehicles.

2-6.3 RADIAL ACCELERATIONS (TUMBLING)

End-over-end tumbling produces centrifugal forces that are directed headward and footward from the axis of rotation. These forces are directly proportional to the square of the angular velocity and may reach magnitudes sufficient to cause extensive physiological damage. The most serious effect of tumbling is the pooling of blood in the head and legs, and an increase in the hydrostatic pressure in these regions. Although this effect can cause the rupturing of small blood vessels and mechanical damage to loosely suspended organs in these areas, the most serious effect is the reduction of blood supply to the heart.

Experimental studies aimed at establishing human tolerance limits for tumbling are forced to terminate before the subject experiences blackout because of the extreme pain involved. Tolerance data are therefore extrapolated from these studies and from studies using animals. Figures 2-9 and 2-10 summarize the data available. It should be noted that at the tumbling rates and durations that are likely to be encountered in military land vehicles physical and physiological stresses are not a problem. They are of great concern to the designer of aerospace vehicles and aerospace escape systems.

2-6.4 ANGULAR ACCELERATIONS

In many respects the vestibular apparatus of the inner ear is similar to part of an inertial guidance system. The vestibular, ocular, audio, and proprioceptive systems of the human body normally interact to provide sensory data relating to the perceptual, psychomotor, and physiological conditions existing at any time. These data are somehow compared with mental records of voluntarily commanded movements of the eyes, head, and body and with stored intelligence concerning the situation. With these various sources of information available, the body has fairly accurate information concerning its state of movement, position, and visual environment. When one or more of these sources of sensory data is missing or experiencing unusual sensory stimulation, the resultant information is often disorienting. Combinations of linear and angular accelerations that are different from those normally experienced often produce abnormal reactions and grossly inappropriate compensatory behavior.

Oculogyral illusion is associated with the effects of angular acceleration on the vestibular apparatus of the ears and on the motion of the eyeballs. With the onset of rotation, an object that is fixed with respect to the subject will appear to move in the direction of rotation. As the rotation attains a uniform angular velocity, the apparent motion of the object will slow down until it comes to rest. If rotation is suddenly stopped, the object

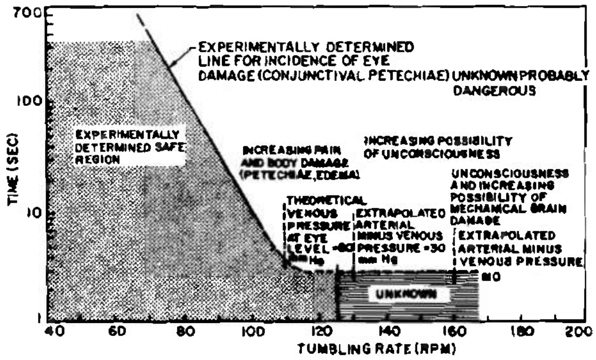

Figure 2-9. Tolerance to Tumbling—Axis at Hip Level (Ref. 5)

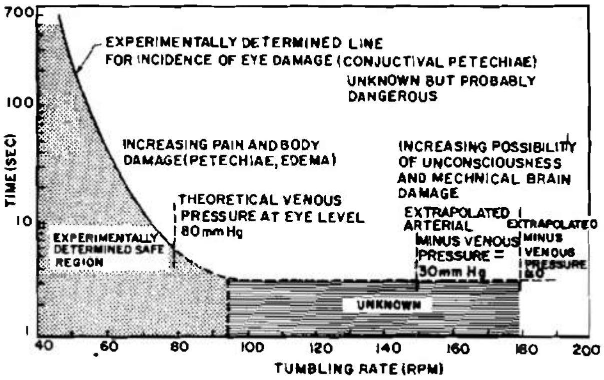

Figure 2-10. Tolerance to Tumbling—Axis at Heart Level (Ref. 5)

will appear to move rapidly in a direction opposite to the previous rotation.

In the dark, or when blindfolded, this illusion is transferred to the body. At the onset of rotation, the subject is usually aware that motion has started and correctly interprets his direction of rotation. With continued rotation at constant speed, however, his feeling of rotating will diminish until he comes to believe that he has stopped rotating. Following this state, and while the subject is still being rotated at a constant speed, he will occasionally experience an illusory feeling of rotating in the opposite direction. When rotation is stopped, a series of illusions follow that are all in the opposite direction to the actual previous rotation.

Also associated with the vestibular apparatus of the inner ear is audiogyral illusion which occurs during angular deceleration. After rotation has stopped, the sources of sound appear displaced from their true sources in the direction of the previous rotation. For example: after rotation to the right, a sound eminating from a source directly in front of the subject appears to eminate from a source to the right of his direct front. Average displacements of 17 degrees have been recorded. This effect is opposite in direction to that of oculogyral illusion in which, following rotation to the right, the illusory visual displacement would be toward the left.

When rotational, centrifugal, and gravitational effects are combined, oculogravic illusions occur. One form takes place when a subject is seated facing the center of rotation. As rotation begins, he is subjected to a normal gravitational force, directed downward, and a transverse force directed radially outward, caused by his rotation. As his rotational speed increases, he has the sensation of being tilted backwards; and, when the transverse force reaches about 1.5 g, he experiences an illusion of lying on his back in a horizontally tilted chair fixed to a vertical support and the walls of the room rotating in vertical planes past him. When his rotational speed decreases and stops, the opposite sensations are experienced. This illusion develops over a period of from 5 to 50 seconds while rotation continues at constant speed. When the subject is seated facing away from the center of rotation, a similar illusion occurs, but in the opposite direction.

The illusions just described are brought about by angular accelerations of relatively long duration (several seconds). It is not readily conceivable how these conditions might exist in land vehicles particularly in contemporary types. The information is given, however, to augment general knowledge and it may prove to be an important consideration when applied to some future concepts.

2-7 VIBRATION EFFECTS

The consideration of vibration effects upon the human being are chiefly concerned with their influence upon his physical comfort, health, sensory and mental acuity, and proficiency in performing assigned tasks. Of interest are such periodic mechanical forces which displace or damage body organs, rupture tissue, or produce perceptible feelings of pain, annoyance, or fatigue. In general, these are high-amplitude, low-frequency vibrations;

although vibrations in the audiofrequency range can cause damage to hearing.

Vibration studies conducted on the M41A1, 76mm gun, military tank and on the M59 Armored Infantry Vehicle (tracked) operating at various speeds up to 25 mph over various military test courses (10 mph over the cross-country test course) showed a broad spectrum of vibrations present at various parts of the vehicles. Frequencies ranged from 0 to 1200 cps—although dominant frequencies were below 500 cps—and zero to peak accelerations as high as 13 *g* were recorded (Ref. 18). The frequencies were divided into four frequency bands for consideration; namely, 0 to 99, 100 to 199, 200 to 500, and over 501 cycles per second. Frequencies in all bands were often recorded at the same location demonstrating the complexity of the vibrations experienced by the vehicle. These vibrations are transmitted to the occupants, cargo, and equipment within the vehicle.

The effect of vibrations on the human body depends upon the physical parameters of the impinging energy, its direction of application relative to the axes of the body, and the mechanical impedance and absorption coefficients of body tissue, organs, and of the body as a whole. Furthermore, since the applied frequencies will react with the natural frequency of the body and of its parts to produce resonances, the resonant frequencies of the body and its parts assume special importance.

2-7.1 VIBRATION CHARACTERISTICS OF THE HUMAN BODY (REFS. 5, 6, 9)

Figure 2-11 shows the mechanical impedance of a man standing and sitting on a platform that is vibrated in a vertical direction. The figure illustrates the relationship between the impedance and the frequency of the vibration. At frequencies of 2 cps and less, the body acts as a unit mass, and the impedance varies linearly with frequency. Above 2 cps, the first resonance occurs for the sitting man, between 4 and 6 cps; while for the standing man, resonance peaks occur at about 5 and 12 cps. The figure also illustrates the effect on mechanical impedance of muscular tension and of the use of a semirigid envelop around the abdomen. It is clearly demonstrated that relaxing

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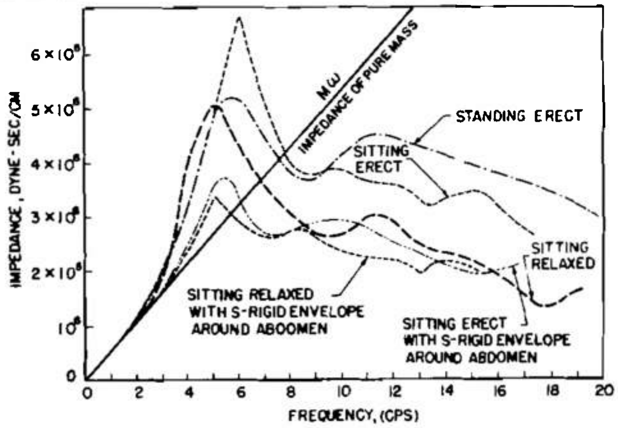

Figure 2-11. Mechanical Impedance of Standing and Sitting Human Subject (Ref. 6)

the musculature brings about a decrease in impedance.

The amplitude and directions of motion of various parts of the body in a vertically vibrating environment are determined not only by the impedances shown in Figure 2-11 but also by the location of the center of gravity of the upper torso and of the head relative to the axis of the vibration. Since the torso center of gravity is forward of the spine, nonvertical motions of the torso will take place. Similarly, a forward-backward oscillation of the head will take place as a result of the displacement of the center of gravity of the head.

Examples of relative amplitudes of different parts of the body when subjected to vertical vibrations are shown in Figure 2-12, for a standing subject, and in Figure 2-13, for a seated subject. The curves show an amplification of motion in the region of resonance and a decrease at higher frequencies. Impedance and transmissibility factors vary considerably with different body types, posture, and degree of support afforded by the seat and back rest, or by the state of the ankle and knee joints for the standing subject; but the resonant frequencies remain relatively constant. Above approximately 10 cps, vibration displacement amplitudes of the body are smaller than the amplitude of the exciting table, and they decrease with increasing frequency. At 100 cps the attenuation of the head is about 40 db.

Figures 2-12 and 2-13 show that the head exhibits a mechanical resonance between 20 and 30

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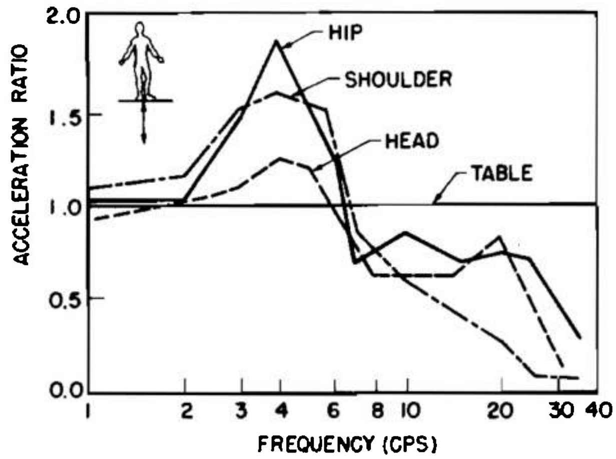

Figure 2-12. Transmissibility of Vertical Vibration of Standing Human Subject (Ref. 6)

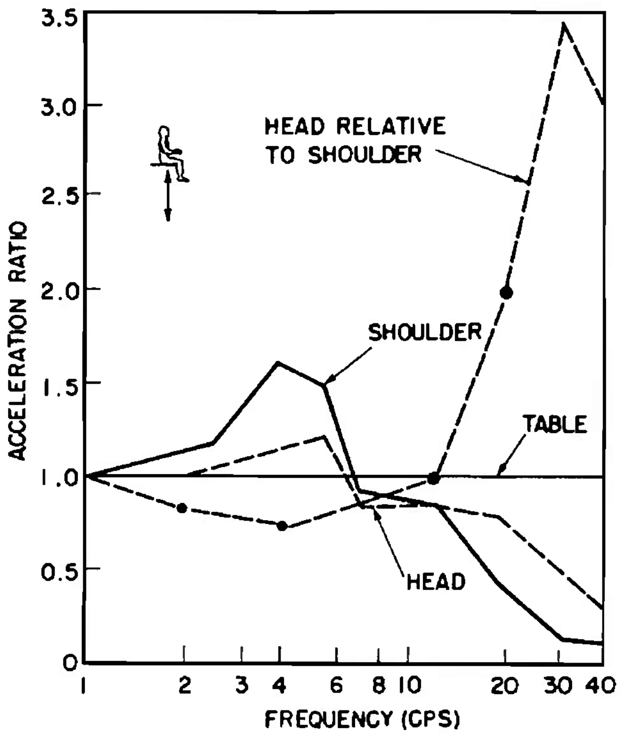

Figure 2-13. Transmissibility of Vertical Vibration of Seated Human Subject (Ref. 6)

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cps. When subjected to vibrations in this frequency range, the head displacement amplitude can exceed the shoulder amplitude by a factor of three. This resonance is an important consideration in connection with the deterioration of visual acuity under the influence of vibration.

Another important factor in human tolerance to vibration is the thorax-abdomen system. The abdominal viscera have a high mobility due to the very low stiffness of the diaphragm and the air volume of the lungs and chest wall behind it. Under the influence of longitudinal and transverse vibrations of the torso, the abdominal mass vibrates in and out of the thoracic cage. This periodic displacement of the abdominal viscera has a sharp resonance between 3 and 3.5 cps.

Transverse vibrations—vibrations in a horizontal plane acting in a front-to-back direction on a human subject in a normal upright position—produce quite a different physical response from that produced by vertical vibrations. In vertical vibration, the acceleration forces act in line with the gravitational forces acting on the body masses; whereas in transverse vibration, the acceleration forces act at right angles to this line. Therefore, the distribution of the body masses along this line is of utmost importance to the characteristics of the response. Furtherfore, the design of the body's supporting structure—the skeleton and spine—is much more effective for vertical loading than it is for transverse loading. This results in a greater difference in response between sitting and standing positions for transverse vibrations than for vertical vibrations.

The transmissibility of transverse horizontal vibrations of the standing and seated human subject is shown in Figure 2-14. The displacement amplitudes of the hip, shoulder, and head of the standing subject are about 20 to 30 percent of the amplitude of the exciting table at a frequency of 1 cps and decrease as the frequency increases. The shoulder and head attain maximum amplitude at about 2 and 3 cps, respectively. The seated subject exhibits amplifications of the hip and head vibration reaching resonance at about 1.5 and 2 cps, respectively. All critical resonances occur between 1 and 3 cps.

In addition to transverse motions, transverse vibrations also excite motions in the longitudinal direction. This is particularly noticeable in the movement of the head. Under transverse vibration the head performs a nodding motion due to the structure of the upper vertebrae and the location of the center of gravity of the head. Above

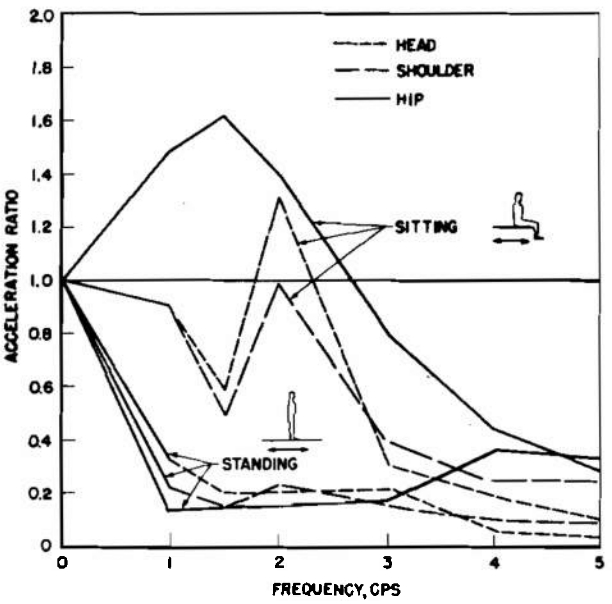

Figure 2-14. Transmissibility of Transverse Horizontal Vibration of Seated and Standing Human Subject (Ref. 6)

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about 5 cps, the head motion for the sitting and standing subject is predominantly vertical.

2-7.2 MAJOR PHYSIOLOGICAL EFFECTS OF VIBRATION

The application of periodic mechanical forces to the human body produces motions and stresses that may have any of several types of effects; namely, mechanical interference, mechanical damage, biological effects, subjective responses, and thermal and chemical effects. Thermal and chemical effects are not considered relevant to the designer of vehicle suspension systems and are, therefore, omitted from the following discussion. The other effects are discussed in the paragraphs which follow.

2-7.2.1 Mechanical Interference

Certain types of displacement, velocity, and acceleration, if of sufficient magnitude, can be disturbing to sensory and neuromuscular activities; such as the reading of instruments (visual acuity), making accurate adjustments of manually operated controls, or maintaining precisely a desired body position. Speech communication becomes difficult. Quantitative data on the types of motion that cause mechanical interference are rather scarce. Tolerance limits are affected by the frequency and amplitude of the vibration, and by the degree of precision required in performing a particular task or activity. A disturbance such as an undesirable lowering of visual acuity, for example, can be controlled by either changing the frequency of the vibration, reducing the amplitude, or decreasing the acuity required for a specified task.

A study (Ref. 19) to clarify the relative importance of changes in frequency compared to changes in amplitude in determining the severity of whole body vibration in the vertical longitudinal plane yielded some interesting results. Tests were conducted to determine the ability of human subjects to perform difficult, two-dimensional tracking tasks while being vibrated with varying degrees of severity. The study showed significant changes in performance as either frequency, amplitude, or both were varied. However, the performance changes that resulted from changes in the amplitude of the vibration overshadowed the changes in performance resulting from changes in vibration frequency to such an extent as to indicate that amplitude is the most significant variable in defining the severity of vibration as they affect human subjects. Furthermore, a threshold amplitude (about 0.06 inch) was shown to exist, below which tracking proficiency is not adversly affected regardless of the frequencies employed (in a range of 2 to 15 cps). This is a rather noteworthy observation, since it is common practice to relate vibration severity to the levels of acceleration that are produced. Acceleration, however, is proportional to the product of amplitude times the frequency squared ($\ddot{y}_o = y_o\omega^2$) and, thus, reflects frequency changes to a much greater degree than it does changes of amplitude. In view of these test results, mathematical expressions intended to characterize the severity of vibrations to humans should recognize amplitude as the predominant contributing factor.

2-7.2.2 Mechanical Damage

Physical damage is produced when the accelerative forces are high enough. Considerable experimental data are available from tests performed on animals, but relatively little is available from tests on man because of the dangers involved to the test subject. Experimental data from animal experiments must be evaluated very carefully before applying any results to humans. Mice, rats, and cats have been killed by exposure to vibrations (Refs. 20, 21, 22). The frequencies of the lethal vibrations coincide with the resonant frequencies of visceral organs.

Mice have been killed at accelerations of 10 to 20 g at vibration frequencies between 15 to 25 cps after only a few minutes exposure. Rats and cats were killed within 5 to 30 minutes at accelerations above 10 g. Post-mortem examination of these animals showed lung damage, heart damage, and brain injury. The heart and lung damage was probably caused by an actual beating of these organs against each other and against the walls of the chest cavity. The brain injury, which was in the form of superficial hemorrhage, was probably caused by relative motion of the brain within the skull, by mechanical action involving the blood vessels or sinuses directly, or by secondary mechanical effects. Tearing of intra-abdominal membranes occurred rarely.

Vibration tests performed on human subjects indicate that a sharp pain occurs in the chest at about 3 g at 3 cps. Traces of blood occasionally have been found in the feces after a 15 minute exposure to 6 g acceleration at a frequency of 20 to 25 cps, indicating mechanical damage to the intestine or rectum.

It is only reasonable to expect some of the effects observed on animals to occur in humans; although, acceleration-frequency curves for these effects have not been established. Due to the relatively greater visceral masses of the human, the minima of such curves would be expected at lower frequencies than in animals and should correspond to resonant ranges of the visceral organs.

Chronic injuries are also possible and are produced by long term exposure to relatively low level vibrations that produce no acute effects. Under actual conditions, chronic injuries are found after long exposure to repeated blows or to random jolts. When such blows or jolts are applied to the human body at relatively short intervals, the relationship of the interval to tissue response time becomes important. Exposure to such an environment is frequently encountered in buffeting aircraft, small, high speed water craft, and in land vehicles operating over rough terrain. Acute injuries are rather rare in these situations, but complaints of discomfort and chronic minor injuries are quite common. Truck and tractor drivers often complain of sacroiliac strain. Minor kidney injuries are occasionally suspected and traces of blood sometimes appear in the urine. The duration of the exposure and the degree of support given to the body are important factors.

2-7.2.3 Biological Effects

The cyclic stresses and oscillatory motions produced in a human body subjected to mechanical vibrations are believed capable of stimulating various receptor organs in the skin and elsewhere, or of stimulating parts of the nervous system directly. These stimuli initiate nervous system and hormonal activities which bring about marked modifying actions to metabolic processes that relate to the assimilation of food, muscular activity, mental activity, etc. These changes are difficult to measure and correlate; nevertheless, there is considerable indirect evidence to substantiate these beliefs. The results of these biological responses—when exposure to mechanical vibration is of sufficient duration and the vibration is sufficiently severe—are such phenomena as fatigue, changes in capacity for work, ability to maintain attentiveness, etc. Acute stimulation may excite brain centers to produce such emotional reactions as fear or unpleasantness and lead to automatic, or deliberate, compensatory or protective behavior.

2-7.3 VIBRATION TOLERANCE LEVELS

Levels of human tolerance to mechanical vibration involve acceleration, rate of onset or acceleration change (jerk), frequency, amplitude, duration, mode of application, body position, degree of support, and physical condition of the subject all acting in combination to produce a variety of physiological (and psychosomatic) effects. Such a large number of parameters, coupled with the difficulties of instrumenting a human being, make it exceedingly difficult to characterize experimental results. The methods generally used are based upon subjective responses of the subjects. These include such factors as perception of vibra-

TABLE 2-4
TOLERANCE CRITERIA FOR SHORT EXPOSURE TO VERTICAL VIBRATIONS AT VARIOUS FREQUENCIES (Ref. 13)

Frequency, cps	Symptom Cited as Reason for Terminating Test						
	Abdominal Pain	Chest Pain	Testicular Pain	Head Symptoms	Dyspnea	Anxiety	General Discomfort
1					8		3
2					8		4
3	2	2			5	1	5
4	2	2		2	3	2	5
5		4				1	6
6	3	4		1			4
7	2	5	1	1			1
8	1	4		1		2	3
9	2	4			1		5
10	1	1	3	2		1	
15							8

NOTE: Numbers in body of table indicate the number of human subjects that reported experiencing the particular symptom listed.

tion, feelings of discomfort, apprehension, and pain. Table 2-4 shows the results of a test (Ref. 13) involving 10 human (male) subjects exposed to vibrations in the frequency range of 1 to 15 cps for periods of less than 5 minutes. The subjects were exposed to a specified acceleration amplitude until it became intolerable. They were then asked for their reactions and for their specific reason for terminating the test run. The table shows the major reasons that were given and the frequencies at which they occurred.

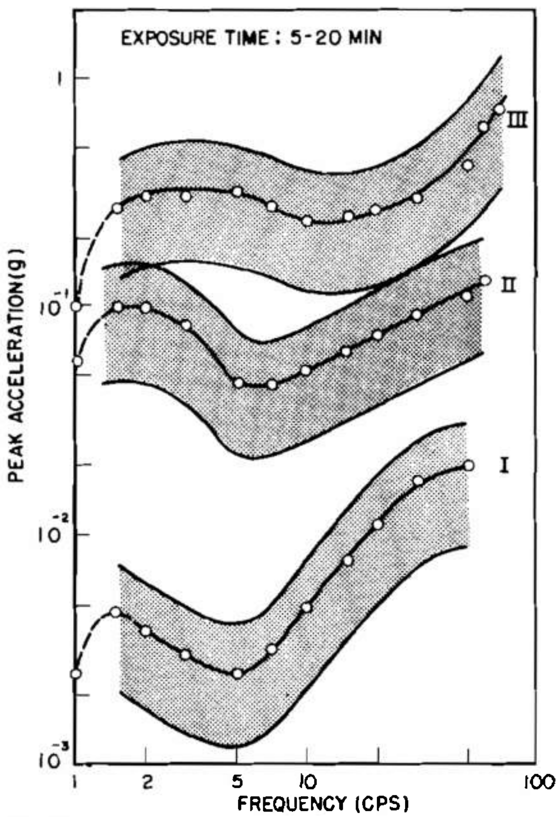
Generally, there are three simple criteria used for classifying subjective responses to vibrations; namely, the threshold of perception, the threshold of unpleasantness, and the limit of tolerance. The last two criteria are difficult to identify precisely, but reasonably reproducible results have been obtained. A compilation of these results, based upon exposures of from 5 to 20 minutes, is shown in Figure 2-15.

Janeway (Refs. 11 and 12) conducted an extensive survey of vertical vibration limits for passenger comfort in which he correlated the experimental data obtained by a large number of investigators. The result of this work has received considerable popularity in the automotive field. He related human tolerance of simple sinusoidal vibrations in the vertical plane to one of three vibration parameters in each of three portions of the frequency range from 1 to 60 cps. In the low frequency range of 1 to 6 cps, he found the jerk, or rate of acceleration change, to be the chief criterion of discomfort; in the intermediate frequency range of 6 to 20 cps, he found it to be the acceleration; while in the upper range of 20 to 60 cps. he related it to the maximum velocity. Table 2-5 gives the maximum recommended values for these parameters and the corresponding vibration amplitudes. Figure 2-16 shows these same relationships graphically.

2-8 NOISE EFFECTS

Noise is usually defined as those audible vibrations, or sounds, that are unpleasant or undesirable by virtue of their loudness, pitch, or quality. From a purely physical viewpoint, sound is a longitudinal wave phenomenon consisting of compressional disturbances produced and propagated in solid and fluid media and, like all wave motions, exhibits a wavelength, frequency, and an amplitude or intensity. Sound waves whose frequencies lie between about 16 and 20,000 cps are perceived as audible sound by the average, young human observer. Above and below this frequency range, the sound waves are inaudible but obey the same laws as do audible sound waves.

Within the audible sound range, the human ear is responsive to a wide range of intensities. It is



LEGEND:

CURVE I IS THRESHOLD OF PERCEPTION
CURVE II IS THRESHOLD OF UNPLEASANTNESS
CURVE III IS LIMIT OF TOLERANCE
DURATION OF EXPOSURE WAS 5 TO 20 MINUTES
SHADED AREAS REPRESENT ONE STANDARD DEVIATION ABOUT THE MEAN

Figure 2-15. Vibration Tolerance Criteria (Ref. 9)

most sensitive in the range of 500 to 5,000 cps and, in this range, the ratio of sound intensity which the ear can tolerate to that which it can just detect is approximately 10^{12}. The minimum sound intensity that the human ear can detect varies widely at different frequencies. A sound intensity which can just be perceived at 3,000 cps, for example, must be increased by a factor of 10^8 to be perceived at 50 cps.

Sound intensity is usually expressed in decibels by comparing it with a standard reference intensity and is the sound power that is transmitted through a unit area of wave front. A decibel is a logarithmic scale unit for expressing the relative magnitude of two sound (or electrical) powers. No instrument is available for the direct measurement of sound intensity; therefore, the magnitude of a sound field is usually specified by its sound pressure level, in decibels. A decibel is defined as 20 times the logarithm to the base 10 of the ratio of the actual sound pressure to the reference sound pressure. It is always necessary to state the reference pressure. A sound pressure of 0.0002 microbar is the most commonly used reference pressure for airborne sound. It is the stimulus found to be barely audible by observers selected for their very acute sense of hearing (about 1 percent of the population).

The *quality* of sound is a subjective attribute of an audible vibration by means of which equally loud sounds can be distinguished as different in kind. For musical tones, differences in quality are caused by differences in the energy distribution among the harmonics (overtones) of the fundamental frequency. For the unpitched sounds which constitute typical noise, differences in quality represent differences in the energy distribution in various parts of the acoustical spectrum. The quality of a noise affects both the sensation of loudness that it produces and its psychological annoyance. Shrill, high-pitched, and irregular sounds are usually judged less pleasant than low-pitched and regular sounds.

Loudness of a sound, or noise, is another subjective attribute of audible vibrations and one that cannot be simply correlated with physically measurable attributes of sound waves. In order to provide a quantitative basis for considering the subjective attribute, loudness, the term *loudness level* is used, and its unit of measure is the *phon*. The number of phons of loudness of a given sound is equal to the pressure level (in decibels above a standard reference) of a pure 1000 cps tone that is judged by a typical observer to have the same loudness as the sound in question. This loudness-level scale is applicable to complex sounds as well as to pure tones since it has been shown that observers can make consistent judgments of loudness equality even when the sounds being compared vary widely in frequency and quality.

Figure 2-17 shows the audible frequency and intensity range for human subjects. The equal loudness lines show the pressure levels required for a given pure tone to sound as loud as a corre-

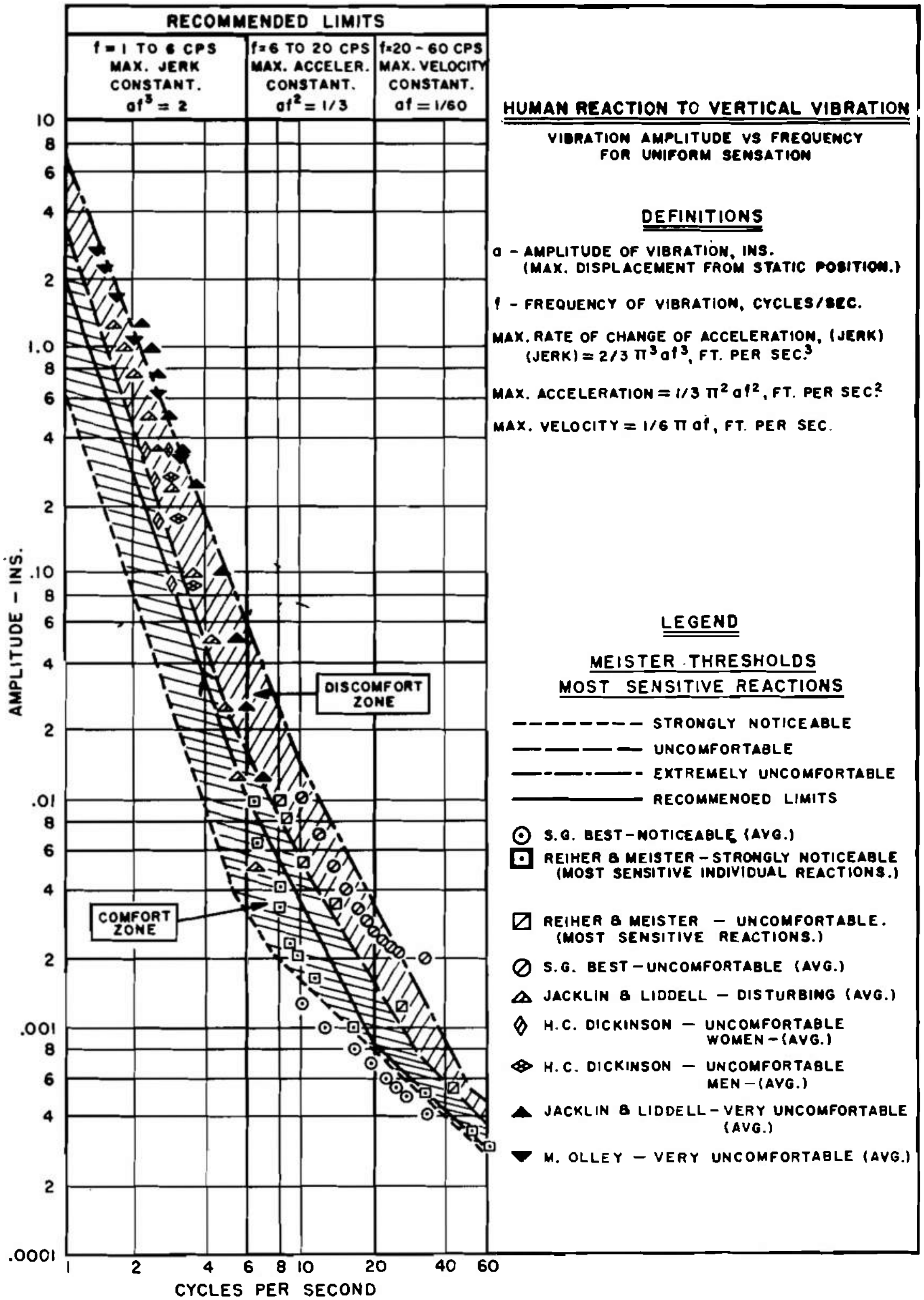


Figure 2-16. Human Reaction to Vertical Vibrations (Ref. 12)

TABLE 2-5
RECOMMENDED CRITERIA OF HUMAN DISCOMFORT FOR SIMPLE SINUSOIDAL BODY VIBRATION IN THE VERTICAL PLANE (Ref. 11)

Frequency, cps	Discomfort Criteria	Tolerable Amplitude, inches
1–6	Jerk, Max. = 40 ft per sec^3	$\frac{2}{f^3}$
6–20	Acceleration, Max. = 1.1 ft per sec^2	$\frac{1}{3f^2}$
20–60	Velocity, Max. = 0.105 inches per sec	$\frac{1}{60f}$

Legend: f is the frequency in cycles per second.

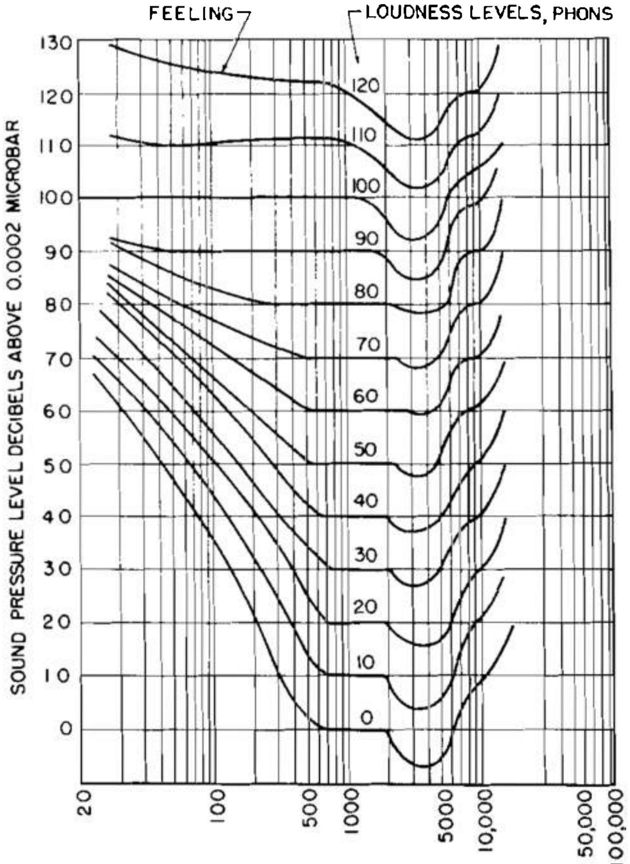


Figure 2-17. Audible Frequency and Intensity Range for Human Subjects

sponding 1000 cps reference tone. The extreme lower line represents the lower threshold of audibility for an observer with acute hearing; the upper extreme represents the threshold at which sound perception merges into a feeling sensation within the ear. Typical sound levels of some common sounds and environments are given in Table 2-6.

Of principal interest from a human factors viewpoint are the maximum levels of noise that are allowable and the effects produced when these limits are exceeded. The specific effects of noise on man are (Ref. 23):

(a) Interference with communication.
(b) Loss of hearing, both temporary and permanent.
(c) Discomfort, pain, and, finally, damage to tissue.
(d) Immediate or short-term effects such as fatigue, loss of sleep, psychosomatic or neuropsychiatric symptoms, etc.

Table 2-7 summarizes the results of some significant tests on humans that should be of interest to vehicle designers.

Temporary hearing losses resulting from exposure to noise increase with increases in noise level, with the duration of the exposure, and with decreases in the band width within which the energy is concentrated. The effect is a loss of auditory acuity, especially in the 1,000 to 6,000 cps range, and a reduction in the apparent sound level of the noise. Temporary hearing losses are produced rapidly and reach their maximum within about 7 minutes for exposure to pure tones. Maximum loss from wide-band noise takes longer and depends upon whether or not it is a steady-state noise. For a steady-state noise in a tracked ve-

TABLE 2-6
TYPICAL SOUND LEVELS OF SOME COMMON SOUNDS AND ACOUSTICAL ENVIRONMENTS

Sound Pressure		Description	
decibels	dynes/cm²		
0 10	0.0002 0.001	Very faint	Threshold of audibility Soundproof room
20 30	0.002 0.01	Faint	Whisper Rustle of leaves Quiet conversation Quiet auditorium
40 50	0.02 0.1	Moderate	Quiet home Quiet radio Average conversation Average office
60 70	0.2 1.0	Loud	Noisy home Average factory Average radio Average street noise, noisy typewriter
80 90	2 10	Very loud	Noisy office Police whistle, loud speech Heavy street traffic Noisy factory
100 110	20 100	Deafening	Very loud street noise Boiler factory Elevated trains Nearby riveter
120 130	200 1,000	Very deafening	Loud thunder and artillery fire Jet engines Threshold of feeling Ear discomfort
140 150 160	2,000 10,000 20,000	Intolerable	Jet engines

hicle or in an industrial setting containing octave-band pressure levels of 90 to 100 decibels, an average loss of auditory acuity of 15 decibels for tones about 1,000 cps can be expected following an exposure of about 4 hours. Exposure to a non-steady and intermittent noise of the same intensity is much less severe.

Table 2-8 shows noise level criteria that have been established by the U.S. Air Force as the maximum tolerable noise levels that can be permitted at a human operator's position. The criteria are divided into three categories according to operational requirements. The first category, "Damage Risk (less than one hour)," is used in situations where the equipment will normally operate for periods of less than one hour per day and where no speech communication is required. The second category, "Damage Risk (over one hour)," is used in situations where the equipment will normally operate for periods of 1 to 8 hours per day and where no speech communication is required. The noise levels given in these two categories represent the maximums to which unprotected personnel can be exposed for the indicated periods without risk of hearing damage. The third category, "Communication," is used in situations

TABLE 2-7
PHYSIOLOGICAL EFFECT OF NOISE
AT VARIOUS FREQUENCIES AND INTENSITIES (Ref. 23)

Frequency, cps	Intensity, decibels	Effects
100 to 12,000	130	Interference with voice communications, permanent cumulative hearing loss.
1,000	135	Inner ear effects, nausea, vomiting, nystagmus, shifting of visual field.
100 to 12,000	140	Auditory pain, permanent cumulative hearing loss even with best protective devices.
100 to 12,000	150	Massive stimulation of many senses, nausea, vomiting, intense ear discomfort even with best protective devices.
100 to 12,000	150 to 160	Severe breakdown of psychomotor performance.
20,000	160	Unusual fatigue, unbearable pain in palms of hands, body heating

TABLE 2-8
NOISE LEVEL CRITERIA AT OPERATOR'S POSITION (Ref. 24)

Frequency Band, cps	Damage Risk (Less than one hr), decibels*	Damage Risk (Over one hr), decibels*	Communication, decibels*
37–75	115	106	106
75–150	105	96	96
150–300	97	88	88
300–600	94	85	81
600–1200	93	84	75
1200–2400	92	83	72
2400–4800	91	82	71
4800–9600	90	81	69

*Decibel reference 0.0002 dynes per square centimeter.

where minimum speech communication is required. The noise levels given in this category will permit adequate voice communication only at distances of less than 3 feet and require that the talker shout.

SECTION III ENVIRONMENTAL FACTORS

2-9 GENERAL

Environmental factors that affect the suspension system and the mobility of a vehicle fall into three broad categories; namely, terrain, climatic conditions, and combat conditions. Terrain includes such considerations as slopes and defiles, ground conditions, vegetation, obstacles, and streams; climatic conditions include the ambient temperature, humidity, wind, and precipitation; and combat conditions include shocks produced by weapon recoil, ballistic impacts, high energy blast, shocks encountered during air-drop operations, and temperatures above the ambient produced by thermonuclear weapons and flames. Any one of these considerations can be the determinant in the mobility of a vehicle at a particular time; but more commonly, mobility is determined by the combination of two or more elements. Although these various environmental elements are discussed separately here, one should bear in mind that the effects of one element often depends, at least partially, upon the effects of other elements; and it is the integration of effects which determines whether a vehicle will move in a particular environment with ease, with difficulty, or not at all.

2-10 TERRAIN

The most obvious characteristics of terrain are its slopes and defiles, soils (including accumulations of snow and ice), vegetation, streams, and cultural features. Climate is sometimes considered as a terrain factor in that its influence has a direct bearing upon the other factors; such as causing the soil to be dry, moist, wet, frozen, or snow covered, or by effecting the condition of streams. These terrain factors and their influence upon vehicular suspension systems are taken up separately.

2-10.1 SURFACE CONFIGURATIONS

2-10.1.1 Slopes

By definition a slope is the inclined surface of a hill, mountain, ridge, or any other part of the earth's surface. It includes not only the inclination of the macrorelief features (hills and mountains), but also the microrelief features, such as small ditches and gullies, mounds, low escarpments, and small sink holes. Some of these microrelief features might be appropriately classified as roughness factors rather than slope factors, but they are included with slopes because their obstacle value is derived from the steepness of their sides.

Slopes are generally expressed as percent of slope rather than in angular degrees. The percent of slope is the amount of vertical rise in elevation per 100 feet of horizontal distance. Thus a slope of 100 percent would be the hypothenuse of a right triangle 100 feet long on the base and 100 feet high on the side. A 45° angle to the horizontal, therefore, is equal to a 100 percent slope.

A diagram of a 100 percent, or 45°, slope does not appear to be particularly steep, but analysis should not be deceived by this. Slopes in nature greater than 80 percent are rather uncommon, unless they are rock escarpments or slopes of loess or volcanic ash. Slopes of 60 percent are about the steepest that soldiers can ascend by walking straight up the incline. Above this limit they must adopt a zigzag course; and above 100 percent (45°), progress is extremely difficult, although not wholly impossible (Ref. 25).

Military vehicles, both combat and tactical, are required to climb slopes of 60 percent. This requirement applies to wheeled as well as tracked vehicles. However, under most practical military operations, this limit is considered too high for vehicles to negotiate. The slope climbing ability obviously depends upon the vehicle having sufficient power (Ref. 26) and less obviously upon the soil strength and grouser action of the wheels or tracks. In weak soils, the ground-contacting elements (wheels or tracks) cannot develop sufficient traction, due to soil failure, to overcome the grade resistance. On extremely hard, smooth surfaces, such as rock slopes or frozen ground, the grousers cannot penetrate and the maximum tractive effort is limited by the coefficient of friction of the surfaces in contact.

Furthermore, the maximum tractive effort that can be developed by either wheels or tracks, through either grouser action or through surface friction, is dependent upon the component of the vehicle weight that acts normal to the ground. Figure 2-18 shows a vehicle on a longitudinal slope. The normal force component F_n is a function of the slope ($F_n = W \cos \Theta$, where W is the vehicle weight) and decreases in magnitude as the slope increases. Additionally, the slope or grade resistance F_g increases as the slope increases. Thus, the vehicle is inherently capable of developing less traction on a slope than on level ground but requires more traction on a slope to overcome the grade resistance.

Tractive effort is further influenced by the nonuniform load distribution on the wheels (or road wheels) that occurs on a slope. This nonuniform distribution of load is related to a number of factors; such as the location of the vehicle's center of gravity, a downhill shifting of the center of gravity due to an elastic suspension system, the type of soil in which the vehicle is operating, dimensions of the wheels or tracks, configuration of the soil surface, the presence of acceleration forces,

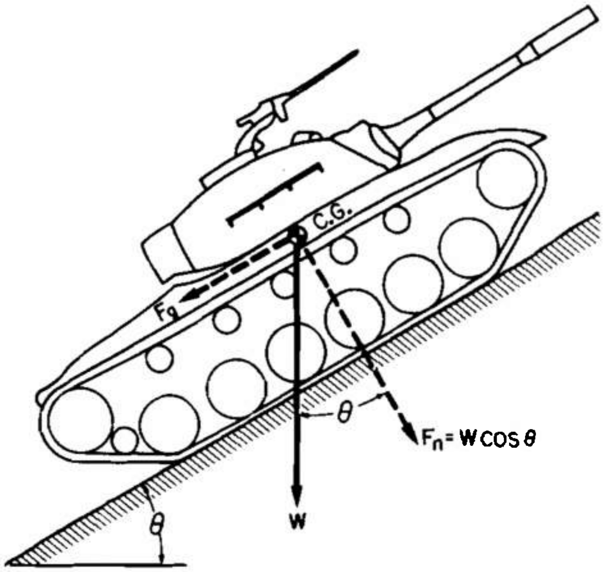


Figure 2-18. Effect of Slope on Force Component Normal to Ground

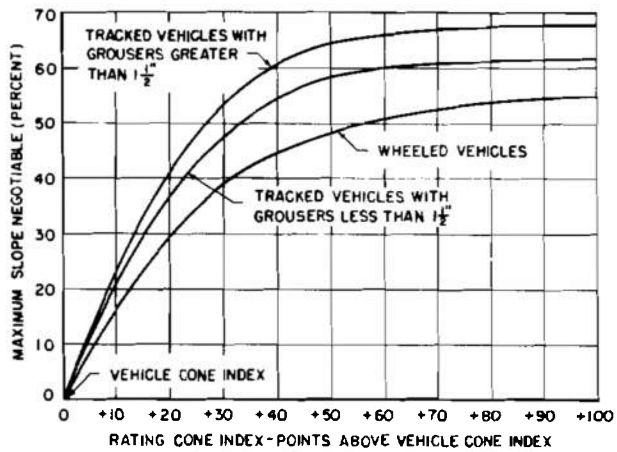


Figure 2-19. Relationship Between Slope Climbing Ability and Soil Strength (Ref. 25)

and the drawbar height, in cases of a towed load. The net result of these factors is an increased loading of the downhill wheels and a corresponding unloading of the uphill wheels. In a wheeled vehicle, this unloading of the front wheels makes steering more difficult and may result in a total loss of steering control. In a vehicle with multi-axle propulsion, this unloading of the front wheels may reach proportions where the driving torque surpasses the traction capacity of the axle, making that axle entirely ineffective in driving the vehicle. The increased loading on downhill wheels or track components increases the loading on the supporting soil, which results in an increase of rolling resistance and may lead to soil failure.

In downhill operations, much the same kind of reduction of maximum tractive effort occurs. Traction is now needed, however, to develop braking effort. The weight component acting parallel to the slope (F_g in Figure 2-18) becomes an acceleration force tending to increase the vehicle's downhill speed. Traction capabilities, however, have decreased due to the factors mentioned previously. In the case of a wheeled vehicle on a particularly steep downgrade, the unloading of the rear wheels may result in such a loss of traction as to make the vehicle dangerously unstable.

Figure 2-19 shows the relationship between the maximum slopes that could be climbed by wheeled and tracked vehicles and the soil strength. The tests were performed on poorly drained, fine-grained soils and sand with fines. The vertical scale represents the slope in percent, and the horizontal scale represents the soil strength required above that needed for movement over flat ground. Discussion of the cone index, the units of soil strength, is given in Chapter 7. It should be noted, that at slopes of about 30 to 40 percent for wheeled vehicles and 45 to 50 percent for tracked vehicles, the curves begin to flatten. Beyond these points, very little is gained in slope climbing capacity, even though the soil strength is increased substantially.

There is nothing that a suspension designer can do to improve the strength of the soil supporting his vehicle, but he can do something to improve the vehicle's slope climbing ability by proper suspension design. Obviously, the wheel and track design must strive for maximum traction at the ground-contacting surfaces. This requires the use of aggressive grousers on tracks and tires, as uniform a distribution of load on the ground as is possible, and the use of materials that have high friction coefficients on smooth, hard surfaces.

Grousers must also be effective in a lateral direction to give stability on side slopes, on curves, and during broadside firing of the major weapons mounted on the vehicle. Since a certain amount of lateral slip is required to negotiate a turn, particularly in the case of tracked vehicles, the effectiveness of the grouser elements in resisting lateral forces becomes a handicap in steering the vehicle and requires the expenditure of considerable power (Refs. 27-31). Thus, the designer must tread a careful path of compromise to achieve adequate lateral stability with minimum power loss to steering.

2-10.1.2 Hard Surface Roads

The chief characteristics of hard surface roads are: reasonably smooth surfaces that are free of vertical obstacles or sudden depressions, gentle longitudinal undulations, gentle slopes (generally under 10 percent), and excellent load-carrying characteristics which provide minimum rolling resistance. These conditions permit operations at greatly increased speeds and allow increased payloads on the vehicles.

As the vehicle speeds increase, however, certain undesirable effects are encountered. In the case of tracked vehicles particularly, the increased speed and violence with which the track blocks strike the hard pavement increases the frequency of secondary vibrations that are transmitted to the vehicle body and to its contents. The noise level within the vehicle is greatly increased and may reach intolerable levels (Ref. 32). The rubber blocks applied to the faces of certain track designs become overheated more readily in high speed operation on hard surfaced roads. This leads to a blowing out of the rubber blocks and to internal separation of the rubber—both of which are highly undesirable because they decrease the usable life of the blocks.

In addition to generating heat and noise, the continuous impacting of the track blocks against the hard surfaced road constitutes a power loss. This loss increases with speed and with the hardness of the road surface.

The maximum tractive effort that can be developed on a hard surface road is limited by the coefficient of friction between the road and the ground-contacting surface of the track. Thus, the objectives of track block designs for hard road operations are the attainment of high tractive forces through the use of materials with high coefficients of friction, designs that tend to make the track blocks adhere to the road, low hysteresis, and good heat dissipation characteristics.

In addition to the effects of hard surfaced roads upon tracks, the suspension designer must also be concerned with the effect of tracks upon the road. A tracked vehicle uses skewing maneuvers to change or correct its course; but, since the traction surface is designed to resist lateral motion, this skewing tends to tear up the road surface. Furthermore, it is quite obvious that tracks equipped with metal grousers or cleats will score, chip, abrade, and break up the surface of a hard pavement; but the effect of rubber blocked tracks is often underestimated. Ground pressures exerted by tracked vehicles upon pavements are very low as compared to pressures under the tires of heavy wheeled vehicles, and the fact that the pressure is more or less continuous over two large areas rather than concentrated on several relatively small areas makes the deflection experienced by the pavement less severe under tracked vehicles than under wheeled vehicles of comparable gross weight. The manner in which the load is applied to the pavement, however, is quite different for the two types of vehicles.

In the case of wheeled vehicles, the load is applied smoothly to successive areas of the road as the vehicle advances, resulting in a deflection wave in the pavement that moves longitudinally with the vehicle. In the case of a tracked vehicle, however, the load is applied in a continuous succession of impacts as each track block strikes the ground (see paragraph 10-24.3 of Chapter 10). This produces a spalling of the pavement surface and sets up a vibration in the pavement that is transmitted through the base course to the subgrade and to the shoulders of the road.

The effect of this vibration depends upon the thickness of the surface course and upon the composition and density of the base and subgrade. In situations where the subgrade or base courses have not been sufficiently compacted, this vibration causes a redistributing of base course or subgrade material, resulting in voids under the hard surface course. Cracks eventually develop and the pavement will fail. This situation is aggravated when the base or subcourses consist primarily of cohesionless materials (see Chapter 4). Base courses and subgrades that contain cohesive materials are less affected by vibrations and, therefore, are less prone to failure under the continuous traffic of tracked vehicles. The surface course, however, is still subject to spalling due to the impact loading.

2-10.1.3 Secondary Roads

The effects of secondary roads upon vehicle suspensions are not materially different from those discussed in the preceding section. Grades may be steeper (up to 32 percent for short distances in high mountains) and road surfaces are less smooth. Greater rolling resistance is encountered on secondary roads than on hard surfaced roads as evidenced by the need for increased horsepower to maintain a given speed. Approximately 12 to 28 percent more horsepower was required to develop speeds of 10 to 25 miles per hour on gravel as compared with concrete pavement.

In general, less traction can be developed on secondary roads than on hard surfaced roads due to a reduced coefficient of friction between the contacting surfaces caused by the rolling or sliding of loose surface material. Full advantage cannot be realized from aggressive grousers on the wheels or tracks because the road is sufficiently hard to prevent grouser penetration. In addition, the loose material on the surface of secondary roads has an undesirable abrasive effect upon suspension components, increasing wear and shortening component service life. The sharp edges and corners of gravel used in secondary roads cause chipping, cutting, and chunking of rubber track blocks.

Secondary vibrations and noise caused by the track blocks impacting the ground are somewhat less severe on secondary roads than on hard pavements. This is due to secondary roads being generally softer and more readily deformed, thus tending to attenuate the dynamic forces and make the impacts less severe.

Effects of vehicles, both wheeled and tracked, upon secondary roads are more severe than upon hard surfaced roads. The spinning of wheels and the wheel loads cause the surface material to move toward the edges and shoulders of the road. This results in ruts, chuck holes, and washboard effects. Tracked vehicles disturb the wearing surface of the road with the grouser action of their tracks, the lateral scrubbing resulting during steering maneuvers, and with the severe vibrations the impacting track blocks produce. This vibration affects not only the wearing surface of the road, but is transmitted to the base course and subgrade as well. Road material is displaced outward leaving ruts, holes, and washboard conditions behind.

2-10.1.4 Cross-Country Operations

The term "cross-country operations" refers to overland movements of military vehicles and foot troops without the benefit or use of roads or any type of temporary or improvised matting or other expedient device. A large number of factors affect movements of this type; such as slopes, soil characteristics, vegetation, streams, cultural features, and climate. The effects of these are discussed individually.

The salient aspect of cross-country movements, however, is that vehicle requirements vary quite widely for different situations, particularly with respect to the suspension system. The requirements for movement over hard, rocky terrain are quite different from those needed for movement through deep mud, or from those needed for operations in snow, or on ice. Furthermore, the situation is very sensitive to climatic conditions and weather, so that a change in weather may bring about an entirely different set of requirements. Thus, a certain amount of versatility must be designed into the system to make the vehicle useful under a maximum number of conditions. In addition, it should be capable of rapid modification—through the use of appropriate kits—to render it effective in other situations.

2-10.1.5 Water Operations

Two general types of water operations are encountered by military vehicles which affect the suspension systems; namely, fording and swimming. In a fording operation, the vehicle crosses the water obstacle by propelling itself through contact with the ground beneath the water in the same manner that it propels itself on dry land. Since ground conditions of the river bottom may vary from firm to extremely soft and from relatively smooth to boulder strewn, the tractive devices and suspension linkages must be designed to cope with these situations.

Additional tractive effort is required during fording operations to overcome the resistance of the water against the vehicle's frontal areas. Furthermore, additional rolling resistance is often encountered in the form of large boulders in the path of the vehicle, excessive sinkage into a soft bottom, and dense underwater vegetation. The development of tractive effort, however, is hampered by the buoyant forces that act on the submerged vehicle. The effect of these is to reduce the total downward force due to the vehicle's weight, resulting in reduced grouser penetration and reduced traction. Strong currents may produce a washing away of sand from beneath the tracks or wheels, thus further decreasing the tractive effort and increasing rolling resistance.

Steering, too, is more difficult during a deep fording operation. This is due to the reduced traction and increased rolling resistance. In the case of tracked vehicles fording a rock-strewn river, rocks and boulders may impede the characteristic yawing or slewing that are necessary in changing the direction of this type of vehicle.

Angles of approach and departure (defined in paragraph 2-10.1.6) are especially important in deep fording operations and should be made as large as possible. A vehicle with an inadequate angle of approach may ford a deep river successfully only to become incapacitated at the far shore by nosing into a steep bank that it is unable to climb because of structural interference. Similarly, a vehicle can become "hung-up" on entering the water by insufficient angle of departure.

Many military vehicles are required to have swimming capabilities. This requires that the vehicle have a buoyant hull, capable of floating it when fully loaded, and a means of water propulsion. This latter requirement is sometimes accomplished by providing the vehicle with a powered screw-propeller or a hydrojet. The most common practice, however, is to use the vehicle's wheels or tracks to achieve this end. Special grousers are sometimes added to the tracks and wheels to make them serve as paddle wheels. Directional control is also achieved by means of the tracks and wheels in much the same manner as is done on dry land. Speeds in the neighborhood of three knots are developed in this manner.

One of the most critical situations for an amphibious vehicle is the transition from operating as a land vehicle to operating as a water vehicle, or vice versa, which takes place when entering or leaving the water. When entering the water, the maximum tractive effort that can be developed by the ground-contacting elements (wheels or tracks) is progressively decreased as the buoyant forces support a greater proportion of the vehicle weight. When the vehicle is finally floating free of the bottom, there is zero tractive effort, and all propulsion must come from the water or other means. In situations where the banks are very soft, slippery, or marshy, the tractive effort may become insufficient to overcome vehicle resistance before the vehicle is waterborne. In these circumstances, the vehicle will be immobilized on the river bank—partially in the water and unable to move in further because of insufficient traction, and unable to move back up the bank because this direction requires additional tractive effort to overcome the grade. The situation is similar to that which may occur in a marsh or bog. There the ground is incapable of supporting the vehicle on its wheels or tracks as a land vehicle and yet offers sufficient resistance to prevent propulsion as a water-borne vehicle.

When leaving the water, a similar situation occurs although somewhat more difficult. In this circumstance, the transition is from a waterborne vehicle to a land vehicle. As a waterborne vehicle, the amphibian can develop only a very limited maximum driving force which is only a small fraction of its maximum drawbar pull on land. Initially, when the wheels or tracks first touch the upward sloping river mottom, this limited driving force together with the vehicle momentum are all that hold the ground-contacting elements against the bank and enable them to develop traction. As the vehicle moves up the bank, a progressively greater load is taken by the ground, resulting in a progressively greater available tractive effort. If the bank is steep, soft, or marshy, the vehicle may be unable to develop sufficient traction while still afloat to begin the ascent of the bank. It may thus become immobilized by a bank that it could otherwise negotiate satisfactorily if it could bring more of its weight to bear upon the ground. Factors that affect the vehicle's ability to emerge from the water at difficult landing sites are: waterborne drawbar pull, water speed, aggressive grousers, ability to shift vehicle's center of gravity, and an adjustable ground clearance.

2-10.1.6 Man-Made Obstacles

The most common man-made obstacles associated with terrain are trenches, ditches, and vertical walls. The difference between trenches and ditches, as the terms are used here, is the manner in which a vehicle crosses them. A trench is a horizontal void or cavity, relatively narrow with respect to the vehicle length, having vertical or near vertical walls (Figure 2-20). The vehicle makes use of its length, longitudinal rigidity, location of its center of gravity, and its tractive characteristics to bridge the gap. A ditch, on the other hand, is a horizontal cavity that is relatively wide with respect to the vehicle length and usually has sloping sides (Figure 2-21). Since its width is such that it cannot be bridged by the vehicle, a crossing must be accomplished by entering the ditch at one side, traveling to the bottom, and climbing out the opposite side. Ditch-crossing capabilities depend upon such factors as: slope climbing ability, maximum angles of approach and departure, location of vehicle's center of gravity, suspension stiffness, and the presence or absence of projections on the vehicle that might interfere with the ditch profile.

Vertical walls are obstructions that project above the ground and present a more or less vertical face to the approaching vehicle (Figure 2-22). They may be in the form of a plateau or step from one level to another, or they may be in the form of a true wall requiring the vehicle to climb up, over, and down the other side. Wall-climbing ability depends upon size of wheels or the geometry of the track, tractive ability, length of vehicle, location of vehicle's center of gravity, and, in the case of wheeled vehicles, whether rear wheels, front wheels, or all wheels are driving.

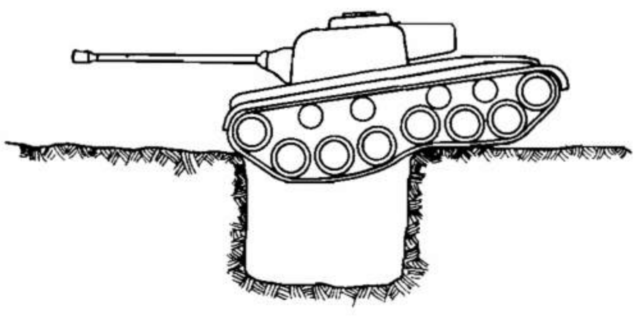


Figure 2-20. Vehicle Crossing Trench

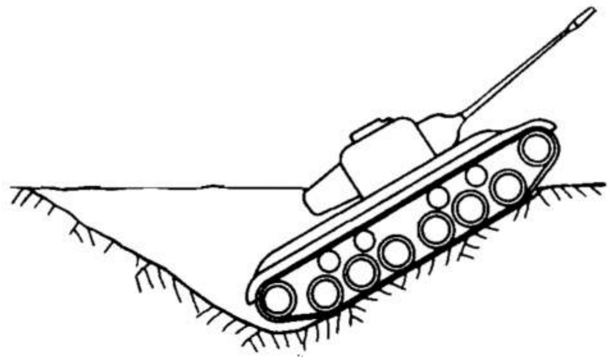


Figure 2-21. Vehicle Crossing Ditch

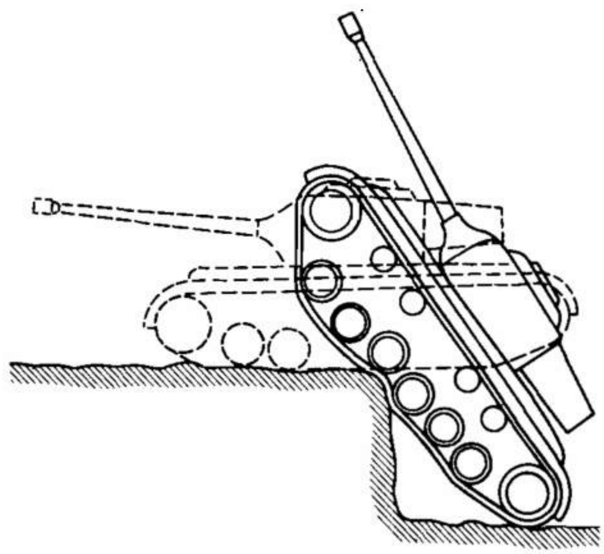


Figure 2-22. Vehicle Climbing Vertical Wall

The immediate effect of major obstacles, such as trenches, ditches, and vertical walls, is to cause the vehicle to slow down appreciably. A prudent driver will slow down to ascertain whether his vehicle can successfully negotiate the obstacle. Even when he knows he can cross the obstacle he will do so at a reduced speed to minimize the shocks to his vehicle, particularly to the suspension system. When crossing a trench, the unsupported portions of the suspension system drop down to the full extent of their restraining linkages, which places them well below the surface of the ground. When the opposite side of the trench is reached, these sagging suspension components experience an impact as they are forced back to their normal operating positions. The higher the speed with which the vehicle crosses the trench, the more severe will be this impact.

Similarly, when a vehicle encounters a ditch, particularly one with steeply sloping sides, the front end is initially unsupported, being balanced over the edge by the weight of the rear end. As the center of gravity passes over the edge of the ditch, however, the vehicle tilts downward until the forward suspension components strike the ditch walls. The resulting shock compresses the forward portion of the suspension system, producing a downward pitch of the vehicle even greater than the downward slope of the ditch wall. Under severe conditions, particularly when the vehicle has a soft suspension system and a high center of gravity or the vehicle noses into the ditch bottom, the vehicle may develop sufficient angular momentum to pitch over onto its back.

A further effect of ditches occurs at the ditch bottom. Since the ditch has sloping sides, the bottom is a modified V. The included angle formed by the walls of the ditch tests the vehicle's angles of approach and departure. Angles of approach and departure are defined as the maximum angle of an incline onto which a vehicle can move from a horizontal plane, for the angle of approach—or off from which a vehicle can move onto a horizontal plane, for the angle of departure—without interference with structural components such as bumpers or fenders (Figure 2-23). Thus, the steepest ditch that a vehicle can enter without becoming immobilized by nosing into the bottom is one whose sides make an included angle equal to, or greater than, the supplement of the angle of approach (180° minus the angle of approach (Figure 2-24)). Similarly, if the included angle formed by the sides of the ditch is less than the supplement of the vehicle's angle of departure, the rear end of the vehicle will hang up against the near

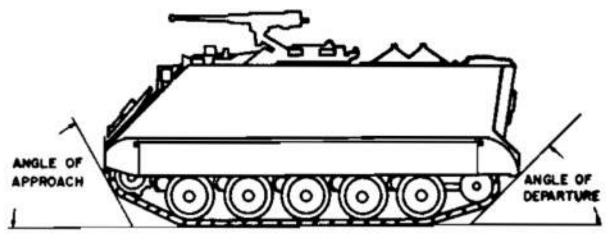


Figure 2-23. Angles of Approach and Departure

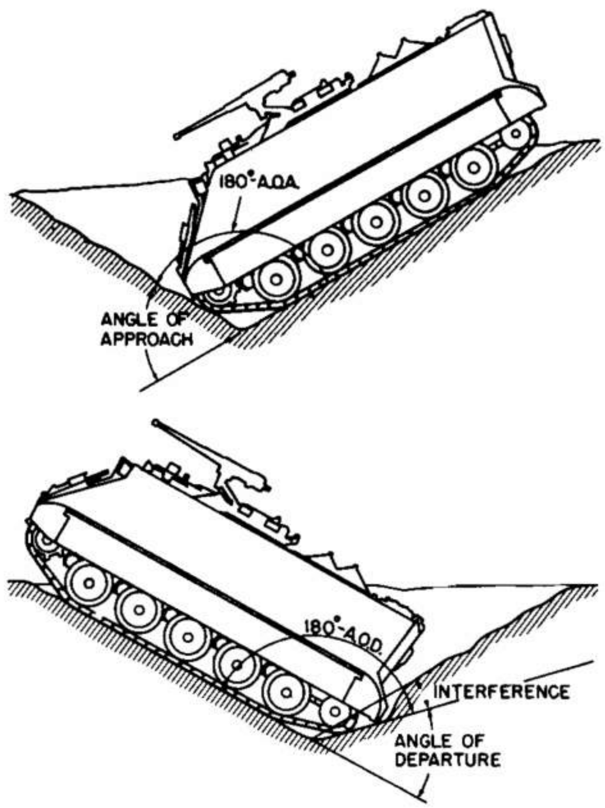


Figure 2-24. Ditch Crossing and Angles of Approach and Departure

wall as the vehicle tries to climb out of the ditch. The lower part of Figure 2-24 shows a tracked vehicle with insufficient angle of departure to clear the particular ditch shown. The rear fender will dig into the ditch wall and may cause sufficient resistance to arrest further progress of the vehicle.

Another situation that occurs at the bottom of a ditch is a loss of traction due to bridging. Figure 2-25 shows a tracked vehicle that has entered a V-shaped ditch. The vehicle's angle of approach was adequate to prevent nosing into the opposite bank, and the vehicle started to climb the opposite slope. Since the vehicle body is rigid, and the bottom of the ditch is narrow, the vehicle body cannot conform to the ditch profile and will need to move upward as it moves forward to effectively bridge the apex of the ditch. The suspension system, having a certain amount of vertical flexibility, will tend to conform to the ditch profile within the limits of its restraining linkage. Once this limit is exceeded, the ground-contacting elements will leave the ground if the vehicle continues to move forward as shown in the figure. This places the full weight of the vehicle on the contact areas at A and B. If this load exceeds the strength of the soil at these points, traction will be lost and the tracks will spin futilely causing the vehicle to dig into the ditch slopes. This will increase the rolling resistance to be overcome by the vehicle, making extrication even more difficult. Since the vehicle cannot move either forward or backward, it is hopelessly immobilized.

The point to be noted, however, is the possibility of a vehicle becoming immobilized by a ditch whose sides do not present otherwise unacceptable grades and whose internal angle is not unsuitable to the vehicle's angle of approach. The main cause of the vehicle becoming immobilized, in this situation, is the loss of tractive effort resulting from the loss of uniform ground pressure which, in turn, is due to the relationship between the ditch profile and the vehicle length.

2-10.2 SURFACE, SUBSURFACE, AND ABOVE SURFACE CONDITIONS

2-10.2.1 Soil Physics

The soil over which a cross-country vehicle is operating is probably the most significant environmental factor in vehicle mobility. The most significant portion is the topmost foot—particularly the layer from the 6- to the 12-in. depth, known as "the critical layer." The thickness and location of the critical layer may vary somewhat with vehicle weight and soil characteristics. In general, however, it is considered to be the lower half of the topmost foot of soil; although the soil to a depth of about two feet, or even deeper, may be of some consequence. Nearly all immobilizations of vehicles in soil are caused by insufficient traction to permit development of the forward thrust necessary to overcome vehicle resistance. It is important, therefore, that the vehicle designer—particularly the suspension designer—have some understanding of the engineering characteristics of soils.

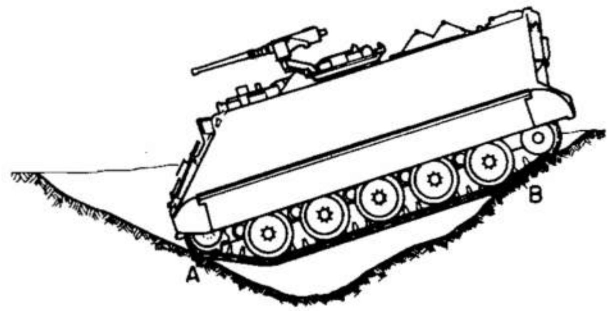


Figure 2-25. Loss of Traction in Ditch Due to Bridging

Individuals of different technical disciplines consider soils from different aspects. Thus, the geologist considers soil in terms of its origin and manner of formation; the mineralogist is interested in the physical, chemical, and crystalographic properties of the soil constituents; while the agronomist is concerned primarily with the chemical and structural make-up of the top few inches of the soil with respect to their ability to support crops. The engineer, on the other hand, is concerned with soils as they relate to the supporting of structures and to their resistance to horizontal forces.

An engineer's definition of soil describes it as ". . . a heterogeneous accumulation of uncemented, or loosely cemented, mineral grains enclosing voids of varying sizes. These voids may contain air, water, or both; or organic matter in various amounts." From this definition it is readily apparent that the engineer is concerned not only with the different mineral grains, but also with the voids between these grains and with the material that exists in these voids.

The mineral grains which comprise the solid portion of soils are particles of rock. All rock found on the earth fall into one of three basic types; namely, igneous, sedimentary, and metamorphic.

Igneous rocks are those that are formed from the molten material, called magma, which is found beneath the earth's crust. This material exists as a dense fluid under extreme conditions of high temperature and pressure. When the earth's crust which confines the magma is ruptured from internal strains, this molten material escapes until a stable condition of pressure is re-established. The molten magma will normally reach a region of lower temperature and solidify into *extrusive* igneous rock. Basalt, obsidian and rhyolite are typical examples of extrusive igneous rock.

Another, and more common, action of the magma is to migrate slowly upward through the crust until it becomes trapped in the crust before it reaches the surface. The cooling takes place at a much slower rate than in the former case; and this decreased rate of cooling, together with the influences of various materials with which the molten mass comes into contact during the cooling, affect the type of grain structure that is developed. To differentiate the igneous rock formed by this slow process from that formed in the volcanic process, it is referred to as *intrusive* igneous rock. Examples of this type of rock are granite, syenite, and trap.

Sedimentary rocks, the second basic type, were produced by the accumulation and cementation of pre-existing rock particles or the remains of plants and animals. Particles that form sedimentary rocks were deposited principally by water; although, some were deposited by winds and some by glacial action. Cementation was the result of clays, or of chemical actions such as that of calcium carbonate. Typical examples of sedimentary rocks are: sandstone, shale, limestone, and coral.

Metamorphic rock result from pre-existing rocks which were subjected to extreme heat and pressure within the earth. Under these conditions, the rock may change into new and completely different rock, hence the name "metamorphic." Typical examples of metamorphic rock are: quartzite, which is metamorphosed sandstone; slate, which is metamorphosed shale; marble, which is metamorphosed limestone; schist, and gneiss.

The reduction of rocks to the minute mineral grains which constitute soils is accomplished by *weathering.* When exposed for a sufficient length of time to the atmosphere, all rocks undergo disintegration and decomposition and ultimately become a loose, incoherent mixture of gravel, sand, and finer material. The changes involved are partly physical and partly chemical. The processes involved may be simple or complex, and they are confined almost entirely to the zone of weathering which extends from the surface of the ground to the level of the ground water. They result wholly from atmospheric agents (moisture and gases) which penetrate the rock and cause disintegration and decay. The total weathering process is divided into two categories: mechanical weathering and chemical weathering.

The soils which result from the weathering processes are divided into two broad categories based upon whether they remained in place during their formative weathering process or were moved during this process. These conditions influence the characteristics of the soils formed. The names given to these two general soil classifications are: *residual soils* and *transported soils.*

Residual soils are soils resulting from a weathering in place and which have not been moved (by nature) during the weathering process. Thus, a residual soil will reflect the characteristics of the parent rock from which it was derived. Cumulose deposits, formed from the deposition of organic matter with small amounts of rock waste (such as deposits of peat and muck in lakes and ponds), are also classified as residual soils.

Transported soils, as their name implies, are soils that have been moved from their place of origin and redeposited. The properties of transported soils have usually been influenced by the mode of transportation that redeposited them. Transport agents are water, wind, ice, and gravity. The most widely distributed of the transported soils are the waterborne soils. These are further subdivided into *alluvial, marine,* and *lacustrine* soils in accordance with the type of water body from which they deposited.

Alluvial soil is a type that is formed when a stream loses its carrying capacity due to a decrease of its velocity. In slowing down, it does not have sufficient power to keep the large rock particles moving and they become deposited on the river bed. As the velocity continues to decrease, ever smaller particles settle; until, as the river becomes slow and sluggish in the lowlands where the gradient is almost nil, only the finest of particles are still held in suspension. Finally, even these are deposited at the river mouth forming fine-grained deltas. The flood plains of the Mississippi and Nile Rivers, their deltas, and fan-shaped deposits where streams slowed down on leaving precipitous mountains, are examples of alluvial soils.

Marine soils are formed from materials carried into the seas by streams and by material eroded from the beaches by wave and tidal action. Part of the material is carried out and deposited in deep water and part is heaped up on beaches along the coast. The continual back and forth motion of sand on a beach is a very effective sorting agent which causes most beach soils to consist of sand and gravel deposits of fairly uniform grain size. Soil deposits from the Old Permian Seas that once covered Kansas and the Midwest are typical of marine soils deposited away from a coastline.

Lacustrine soils, the third subdivision of the waterborne soils, are deposits found in and around fresh water lakes. Generally speaking, these are fine-grained soils resulting from the deposition of material brought into the lake by small streams and rivers.

Aeolian soils are composed of grains that have been transported by the wind. Sand deposited by the wind is called a dune; and finer particles, which are generally carried further, are deposited into formations known as loess. Dune deposits seldom contain gravel (particles of pea size and larger) and are usually of uniform grain size. Loess has the unique ability to stand in nearly vertical walls because of its vertical drainage characteristics. It is a buff to yellowish-brown loam found throughout the gulf coast and the lower Mississippi River region.

Glacial soils consist of materials carried with or upon an advancing ice sheet or those materials pushed ahead of it. As glaciers melt, deposits of various physiographic forms occur such as moraines, kame terraces, eskers, and outwash plains. Moraines consist of mixtures of unstratified boulders, gravels, sands, and clays which are called till. Other forms of glacial deposits consist of somewhat stratified and partly sorted stream gravels, sands, and fines transported outward from the glacier by streams during the period of melting.

Colluvial soils are mixed deposits of rock fragments and soil material that accumulate at the base of steep slopes through the influence of gravity. Colluvial soils also include soil creep and local wash deposits.

Clay is a term applied to a natural, fine-grained mineral which develops plasticity when mixed with limited amounts of water. Like most soils, it consists primarily of decomposed igneous and metamorphic rock. In terms of particle size, however, it is considered comprised of the smallest sized particles that exist in soils (about 2 microns is upper limit).

Clays are generally considered composed of extremely small crystalline particles of one or more members of a small group of minerals that have come to be known as clay minerals. These are essentially hydrous aluminum silicates with magnesium or iron substituting wholly or in part for the aluminum in some minerals and with alkalies or alkaline earths present as essential constituents in some of them. Some clays are composed of a single clay mineral, but in many there is a mixture of them. In addition to the clay minerals, clay soils contain varying amounts of nonclay minerals of which quartz, calcite, felspar, and pyrite are important examples. Also, many clay soils contain organic matter in water soluble salts. The mineralogy of clay is beyond the scope of this handbook and many references exist on the subject (Refs. 33 to 35).

The important point to be remembered is that the plasticity of a soil is the result of the action of water on the clay minerals present. It permits the soil to be remolded without crumbling or rupturing. Plasticity is considered a colloidal property since no mineral possesses plasticity unless it consists of particles of colloidal or clay size. Even then, many minerals do not develop plasticity regardless of how small their particles are. On the other hand, all clay minerals are plastic; and, since practically all fine-grained soils contain some clay minerals, most fine-grained soils are plastic.

The engineering properties of soils depend upon four physical properties that all soils possess; namely, particle size, particle shape, gradation of particles, and moisture content. These are discussed in some detail in Chapter 7 along with a discussion of soil classification systems. In general, it can be stated that soil strength depends upon soil density; coarse-grained soils have greater density than fine-grained soils if there is sufficient gradation of particles to fill all the voids. Angular soil particles impart greater strength to the soils than do round particles. Furthermore, coarse-grained soils are less affected by water than fine-grained soils since the larger voids will drain more easily and capillarity is no problem. Secondly, since the particles in sandy and gravelly soils are relatively large (in comparison with silt and clay particles), they are heavy in comparison with the weight of water that surround them. On the other hand, the small, sometimes macroscopic, particles of fine-grained soil weigh so little that the water within the voids has considerable effect on them. Fine-grained clay soils may have different properties at different moisture content. A given clay may act as a liquid mud, a remoldable plastic, or crumble, depending only upon the amount of water it contains.

2-10.2.2 Effect of Vegetation

For purposes of this discussion, vegetation is divided into three categories:

(a) low growing vegetation less than 3 feet tall,
(b) brush or tall grass more than 3 feet tall, and
(c) trees and forest.

Low growing vegetation that is less than 3 feet tall has no significant effect on the movement of vehicles except where it may obscure obstacles, such as large boulders, stumps, large logs, or ditches.

The mat that is formed by grassy sod tends to strengthen the surface of weak soils, but this strengthening is of very little consequence except in the case of very light vehicles making only single passes over the ground. In general, military vehicles are so heavy that they quickly cut through the sod if the soil below is weak. For this reason the vehicle designer should place no value on the strengthening effects of sod. Sod may be a handicap by presenting a slippery surface to wheels or tracks. This often causes trouble, particularly if the slope is steep. In some locales of low vegetation, unusual difficulties may arise such as tire punctures caused by cacti needles in certain desert areas.

Brush or tall grass that is more than 3 feet tall affects cross-country mobility by impairing the visibility of drivers of wheeled and tracked vehicles and by obscuring obstacles, such as stumps boulders, logs, or ditches. The movement of wheeled vehicles is impeded considerably by tall, thick brush and may immobilize them completely. Tracked vehicles experience little or no deterring effects from tall, thick brush. In fact, brush may enhance conditions for tracked vehicles by spreading the load over a greater ground area.

Trees and forested areas have the greatest effect, where vegetation is concerned, on cross-country movement of vehicles. Trees with trunk diameters less than 3 inches are only a slight hindrance to tanks and most tracked vehicles. The maximum diameter of trees that it is practical for medium tanks to push over is considered to be between 6 and 8 inches; although trees of less than 6 inches in diameter may present unsuitable conditions if they are close together. The maximum diameter of trees considered feasible for a 2½-ton truck to push over is considered to be between 1 and 2 inches (Ref. 36).

Larger trees than those cited can be pushed over under suitable circumstances. The species of the tree, nature of its root system, soil conditions, and height of the tree are all factors that must be considered. But the capability of pushing over trees is not a criterion of whether a vehicle can go. A lone tree is obviously not an obstacle that needs pushing over, for it can be easily bypassed. The forest, however, does present obstacles if the trees are too close together to permit the passage of vehicles between them. The overturning of trees within forests can create other problems. When several trees are pushed over, some may not fall clear but will interlock with others to form a new barrier. Other barriers are developed by the protruding root systems of the overturned trees. And, finally, the trunks of fallen trees can present obstacles that the vehicles cannot cross.

The turning radius of vehicles determines the minimum spacing of large trees in a forest between which a vehicle can maneuver. For current vehicles this minimum distance is considered as between 15 and 20 feet for both wheeled and tracked vehicles. This distance includes an allowance for turning. The width of the current main battle tank, M60, is just under 12 feet (143 inches).

2-10.2.3 Precipitation Accumulation

2-10.2.3.1 *Snow*

Snow exists in many forms that vary from a solid icy crust through granular, so-called "sugar snow," to a soft plastic mass. Since it is really a form of ice, it is a solid that is slippery by its very nature. This characteristic is due to the fact that ice is not a homogenous substance, but a three-phase system in which solid, liquid, and gaseous states coexist in thermodynamic equilibrium at a given temperature and pressure. Any change of pressure or temperature causes an immediate shift in the phase composition, thus involving fluctuations in the structure of the matter. Since snow is an aggregate of ice crystals and air, the factors which involve the phase change are particularly active. An increase of pressure and temperature between a sliding body and snow (or ice) will invariably increase the liquid phase and, thereby, will provide the rubbing surfaces with a lubricant which reduces the friction coefficient considerably. This is why a ski or sled slides easily on snow or ice. The coefficient of friction between objects sliding on snow decreases as the temperature or density of the snow, or the vertical load on the snow, increases; and it increases as the velocity of sliding increases.

Furthermore, the mechanical properties of snow change in response to various factors. Compression, increase in grain size, or decrease in snow temperature all result in an increase in viscosity. Tensile strength increases with compression and with rising temperatures below the freezing point, and decreases as grain size increases (Ref. 37). There are practical limits of snow compaction, however, at different temperatures. In the temperature range of 12° to 30°F, this compaction limit is approximately 5.6 psi. Higher compaction pressures do not alter the snow structure as far as its density is concerned, in the temperature range stated. The granular, so-called "sugar snow," does not compact under pressure. It can be compacted by vibration, and otherwise behaves very much like fine sand.

Snow possesses a shear strength which is a function of pressure, time, and temperature. Within the ranges of pressures encountered in vehicle design, the shear characteristics are very similar to those of soil except, of course, generally much weaker.

Thus, the effects of snow accumulation on vehicle mobility are:

(a) Greater rolling resistance due to snow compaction.
(b) Additional resistance due to bulldozing of snow by front of wheels or tracks and by vehicle hull.
(c) Reduced traction due to low shear strength of snow.
(d) Reduced traction due to slippery characteristic of snow.
(e) Reduced steering control.
(f) Reduced resistance to lateral forces.
(g) Reduced slope climbing capabilities.
(h) Reduced braking capabilities.

2-10.2.3.2 *Ice*

The effect of ice accumulations on vehicle mobility are similar to those of snow; however, the increased rolling resistance encountered in deep snow due to snow compaction and bulldozing are not present in ice operations. The chief problems of movement over ice stem from the low coefficient of friction which leads to reduced tractive action, steering and braking instability, and instability to lateral forces.

When the ice consists of a relatively thin crust over a firm base, tracks equipped with metal grousers, or tires equipped with chains, may penetrate through the ice crust and obtain adequate purchase in the soil beneath. When the ice is thick, however, particularly when the temperature is low, it has sufficient compressive strength to resist the action of the grousers, and the vehicle will be supported on the grouser edges. The only benefit that can be obtained by the grousers in this condition is from the scarifying action that they may have upon the slick surface of the ice. Tread surfaces must be designed to be self-cleaning, for once the grousers become obstructed by impacted accumulations of ice, they become ineffective.

2-10.2.3.3 *Water*

Precipitation of water in the form of rain and its accumulation on the ground can affect vehicle operation in several ways. The effect of moisture content upon the physical characteristics of soils is discussed in Part Two of this handbook. The consideration here is the relatively sudden accumulation of water on the surface of the ground (or road) as by a sudden rainstorm.

Water acts as a lubricant, under certain conditions, and as such it reduces the adhesion between the ground-contacting elements of the vehicle and the ground. This is particularly true in the case of a rubber-tired wheel or a rubber-faced track operating over a hard surface. The film of water trapped between the two contacting surfaces decreases the friction coefficient appreciably. The pressure on the liquid tends to squeeze it out of the way, but the viscosity of the fluid and the tread pattern influence how rapidly and how effectively it is squeezed out.

Fresh rain results in a clear film of low viscosity which is relatively easy to squeeze out. When the water is mixed thoroughly with dirt, however, it coats the road surface with a high-viscosity fluid that is difficult to squeeze out. The time during which the pressure acts on the fluid is a factor, along with the viscosity, in determining whether the fluid will be squeezed out completely or not. At high speeds of locomotion, the tread bars or lugs are not in contact with the film for a sufficient time to allow the liquid to move out of the way. Instead of dry adhesion, a sliding on the viscous film will take place. Thus, the adhesion upon wet surfaces depends upon the speed of locomotion. This is a condition that does not exist on dry surfaces.

The tread pattern, too, enters into the consideration. In order for the fluid to move out from between the contacting surfaces, unobstructed channels must be provided in the tread. These channels, obviously, must not close under the weight of the vehicle, nor can they become plugged by dirt.

Tracked vehicles equipped with rubber track pads experience a similar loss of adhesion. Here, the time during which the blocks are in contact with the liquid film is of sufficient duration to permit the fluid to be displaced, but the ground pressure is usually too low to be effective.

2-11 CLIMATIC CONDITIONS

The climatic environment anticipated for military vehicles operating in all parts of the world is specified in considerable detail in AR 705-15 (Ref. 38). Operating conditions are divided into five climatic categories; namely, *intermediate, hot-dry, warm-wet, cold,* and *extreme cold.*

Intermediate operating conditions are those which exist at certain times and places in the most densely populated portions of the world and where major military activities have taken place in the past. All military combat and combat support materiel is required to be capable of satisfactory performance at all times under these conditions. In brief, these conditions consist of:

(a) Ambient air temperatures from −25° to 105°F.

(b) Relative humidity from 5% at 105°F to 100% at all temperatures from −25° to 85°F.

(c) Precipitation: wind-driven rain falling at specified rates for various time intervals to a maximum of 8 inches over a 12-hr period accompanied by intermittent winds, 35 knots and including short periods of torrential downpour; as specified in AR 705-15, Change 1.

(d) Snowloads of 10, 20, and 40 lb per sq ft depending, respectively, upon whether the equipment is portable (moved daily), temporary (moved often and cleared of snow between storms), or semipermanent (snow not usually removed between snowfalls).

(e) Wind velocities: 45 knots for a 5-minute period with gusts to 65 knots and, with provision of hold-down equipment, 55 knots for a period of 5 minutes with gusts to 85 knots.

(f) Falling snow crystals 0.05 to 20.0 mm in size; median size ranges from 2.0 to 5.0 mm.

(g) Blowing snow crystals 0.02 to 0.9 mm in size at wind speeds above 10 knots. Median particle size varies with temperature; 0.5 mm, above +14°F (−10°C) and 0.1 mm, below the same reference temperature.

(h) Blowing sand particles of 0.01 to 1.00 mm diameter at 15 knots or more.

(i) Blowing dust particles of 0.0001 to 0.01 mm diameter blowing at 15 knots or greater.

The remaining four climatic categories are less prevalent than the intermediate condition. Therefore, for operations in hot-dry, warm-wet, and cold conditions, modification kits will normally be developed to permit the use of the intermediate condition equipment. Specific equipment for the above three areas will be designed only if modification kits are impractical, or if such specific designs achieve meaningful improvement. The last category, extreme cold, occurs only in a few places in North America, Greenland, Siberia, and Antarctica. However, operations here require the design of specialized equipment, almost exclusively. For detailed specifications on all categories, consult AR 705-15 (Ref. 38).

2-11.1 EFFECTS OF AMBIENT TEMPERATURE

Ambient temperature has undesirable effects upon the suspension system at both the upper and lower limits of the range. In fact, the wide range itself presents problems that would be reduced if the total range were narrowed. For example: allowances must be made for thermal expansion when specifying clearances between running parts. This allowance can be applied in a manner that will result in either a freer or a tighter fit, whichever is the more tolerable to the functioning of the specific design. Obviously, the greater the temperature range, the less tolerable will be the design, at least at one temperature limit. The situation may even be such as to require the application of compensating devices to maintain the desired clearances throughout the temperature range.

Certain materials, particularly compounded materials, such as lubricants, hydraulic fluids, and elastomers, are more effective at certain temperatures than at others. They may freeze or become stiff under extreme cold weather conditions and may become thin and even deteriorate under extreme hot weather conditions. Rubber tires become hard and even brittle under extreme cold. Rubber tired vehicles that are parked for several hours in extremely cold temperatures develop semipermanent flat spots where they rested on the ground. These disappear when the tires warm up after a few miles of operation but cause a rough ride in the meantime.

In general, the principal effects of high ambient temperatures can be summarized as: (a) thermal aging resulting in oxidation, structural change, and chemical reactions of sensitive materials; (b) a softening, melting, and sublimation of materials; (c) viscosity reduction and evaporation; and (d) physical expansion of materials. These effects may lead to such failures as: insulation failures, alteration of electrical properties, structural failures, loss of lubrication properties, increased mechanical stresses, and increased wear on moving parts. The principal effects of low ambient temperatures are: increased viscosity and solidification of liquids, hydraulic fluids, and lubricants; ice formation; embrittlement; and physical contractions. These may lead to such failures as: loss of lubrication properties, alteration of electrical properties, loss of mechanical strength, cracking and sudden fracture, structural failure, and increased wear on moving parts.

2-11.2 EFFECTS OF HUMIDITY

The principal effects of humidity are the absorption of moisture by certain materials, which may lead to the swelling of parts and to chemical reactions, such as corrosion and electrolysis (electrolytic corrosion). These typical effects may lead to the physical breakdown of parts, malfunctioning of parts, loss of electrical strength, and the loss of mechanical strength. Low humidity produces desiccation which leads to embrittlement and granulation. These may cause such failures as loss of mechanical strength, alteration of electrical properties, and structural collapse.

High temperature combined with humidity tends to increase the deteriorating effects of humidity. Low temperature combined with humidity results in moisture condensation and, if the temperature is low enough, in the formation of frost or ice.

Sand and dust have a natural affinity for water. Therefore, a combination of high humidity with sand or dust increases deterioration of parts due to abrasion and wear.

2-11.3 EFFECTS OF WIND

The principal effects of wind upon the suspension system are those associated with the deposition of abrasive particles such as sand, dust, snow, and ice crystals. These result in mechanical interference between moving parts, clogging of cooling passages, contamination of lubricants, and a general acceleration of abrasion and wear of components. Sealing requirements are more difficult in the presence of wind-driven abrasive materials.

Heat loss is accelerated in the presence of wind due to the wind-chill factor. In conjunction with high humidity or rain, heat loss is further increased due to evaporative cooling. This accelerates the low temperature effects discussed in paragraph 2-11.1.

2-11.4 EFFECTS OF PRECIPITATION

Effects of accumulated precipitation of snow, ice, and water are discussed in paragraph 2-10.2.3. Direct effects upon the suspension system are accelerated abrasion effects caused by snow and ice crystals, particularly if wind-driven; accelerated low temperature effects caused by evaporative cooling of rain; and increased corrosion due to the removal of protective coatings by the abrasive action of snow and ice and by the dissolving and washing action of rain. In addition, rain enhances chemical reactions, thus also accelerating corrosion, and tends to dilute and wash away lubricants.

2-12 COMBAT CONDITIONS

2-12.1 HEAT BEYOND THE AMBIENT

Combat operations impose additional environmental requirements not normally encountered. Among these are temperatures considerably higher than those of the ambient, such as may be experienced within the sphere of influence of nuclear detonations and in, or at close proximity to, flames.

In a nuclear explosion, approximately one-third of the total energy yield is emitted from the fireball as thermal radiation. This enormous amount of energy liberated results in temperatures within the fireball estimated at tens of millions of degrees. For an air burst at altitudes below 50,000 ft, the thermal radiation is emitted in two pulses. The first, which is quite short, carries about 1 percent of the total radiant energy; the second, which is much more significant, is of considerably longer duration. The duration of the effective thermal pulse increases with the energy yield of the explosion. Thus, the pulse duration from a 10 megaton air burst is about 30 seconds, while from a 1 kiloton explosion it is roughly 0.3 second. The pulse characteristics and duration of explosions at higher altitudes change so that, at altitudes above about 100,000 ft, there is only one effective pulse and its duration is of the order of a second or less for explosions in the megaton range (Ref. 40).

The damage effect of this nuclear thermal radiation is in the ignition of combustible materials. The extent of thermal damage increases with the yield of the weapon and decreases as the distance from the burst increases.

When thermal radiation falls upon any material a part may be reflected, a part may be absorbed, and a part may pass through the material to ultimately fall upon other material. The portion of the radiant energy that is absorbed is transformed into heat; and it is this portion, only, which produces damage to the material.

The extent or fraction of the incident radiation that is absorbed depends upon the color and surface characteristics of the material. Light colors, highly reflective surfaces, and transparent substances do not absorb much of the radiation. Thin materials will transmit a large proportion of the radiation falling upon it and, thus, will escape serious damage. On the other hand, black or dark colored materials and rough or dull surfaces will absorb a greater proportion of the radiant energy. A light-colored material that chars readily in the early stages of exposure, however, will react essentially as a black substance irrespective of its original color.

Whether the suspension system will be seriously affected by this thermal radiation will depend upon the yield of the weapon, the distance to the detonation, and the duration of the pulse. The most vulnerable components are the tires (of both wheeled and tracked vehicles), rubber and fabric track components, seals, hydraulic fluids, and lubricants. If affected at all, these may either burn or become degraded by the intense heat. A secondary effect may take place in the form of a vehicle fire. This may occur through an initial kindling by the radiant pulse of such things as a fabric vehicle top, upholstery, or spilled liquid fuels or lubricants. Once kindled, the fire may spread to the fuel tank or ammunition storage, and the entire vehicle may become involved. The resulting conflagration will destroy seals, evaporate hydraulic fluids, burn away lubricants, destroy rubber tires and rubber track pads, and may seriously affect the strength of structural components and the spring characteristics of the suspension system by degrading the physical properties of the metals used.

Flame-spewing weapons, such as rocket launchers and flame throwers, that are mounted on vehicles produce considerable heat when they are fired. It is, therefore, necessary to shield vulnerable components from this intense heat.

2-12.2 EXTERNAL FORCES

2-12.2.1 Recoil of Weapons

The external forces to which vehicles may be subjected in a combat environment are many and varied. Perhaps the first to come to mind are those produced by the recoil of weapons fired from the vehicle. The chief among these, of course, are the large caliber guns that constitute the main armament of self-propelled artillery, tanks, and self-propelled antiaircraft artillery; but weapons of smaller caliber, particularly rapid firing automatic weapons, also subject the vehicles upon which they are mounted to shock and vibratory loads.

Recoil forces are transmitted through the weapon mounting to the vehicle structure, and thence to the suspension system. The springs and shock absorbers help to attenuate this recoil. Some vehicles are equipped with suspension lock-out provisions, the effect of which is to add the unsprung mass to the sprung mass of the vehicle. This results in a more stable gun platform permitting a faster and more accurate delivery of fires.

Weapon recoil forces can be calculated by the application of well established methods of dynamics and are confirmed by test firing. The procedure is an application of the principle of the conservation of momentum. The momentum (mass times velocity) of the projectile plus the momentum of the propelling gases (and unburned powder) are equated to the momentum of the recoiling parts as they move in a direction opposite that of the projectile.

$$M_pV_p + M_gV_g = M_rV_r \tag{2-6}$$

The velocity of the recoiling parts V_r can be determined as:

$$V_r = \frac{M_pV_p + M_gV_g}{M_r} \tag{2-6a}$$

where the symbols M and V represent mass and velocity, and the subscripts p, g, and r represent the projectile, gases, and recoiling parts, respectively.

Once the velocity of the recoiling parts is determined, and knowing the length of the recoil stroke, L_r, the duration of the recoil can be calculated from

$$t_r = \frac{L_r}{V_r} \tag{2-7}$$

Most recoil mechanisms are designed to give a constant resistive force throughout their working stroke. This force can be determined by equating the momentum of the recoiling parts to the impulse it produces on the trunnion mounting.

$$M_rV_r = F_rt_r \tag{2-8}$$

from which

$$F_r = \frac{M_rV_r}{t_r} \tag{2-8a}$$

A more detailed procedure for determining weapon recoil forces is given in Ref. 41. Recoil forces developed by 50-caliber and 20-mm weapons are given in Ref. 42. Weapon characteristics and the internal ballistics of the ammunition used must be known before their effects upon the suspension system can be evaluated accurately. Consideration must also be given to firing at different angles of elevation and azimuth to determine the most serious conditions affecting the suspension system.

2-12.2.2 Impact of Obstacles

Violent impacts with obstacles, such as boulders, logs, walls, bottoms of ditches, and the pancaking-down of vehicles driven rapidly over the brow of a hill or a parapet are quite common in the combat environment. The magnitude of the forces involved depends upon the speed with which the vehicle encounters the obstacle, the size of the obstacle relative to the vehicle wheels or track, the rigidity of the obstacle, and the rigidity of the suspension system.

In determining the effect of forces upon bodies, the procedures generally used are based upon the relationships between force and acceleration and work and energy, and upon the assumption that the forces act upon rigid bodies during definite (comparatively large) intervals of time. Furthermore, when the forces vary in magnitude during the time period over which they act, the manner in which they vary is known, and this is incorporated into the mathematical procedure used. However, forces often act for extremely short (indefinite) periods of time during which neither their value at any instant nor their manner of variance is known. These forces are known as impulsive, or impact, forces and are the type experienced by the vehicle when it encounters the types of obstacles mentioned.

Impact forces produce extremely high pressures on the bodies upon which they act, resulting in appreciable deformations which make it incorrect to assume the bodies to be rigid. In most cases, the effect of impulsive forces on the motion of bodies is so great in comparison with the effect of the other forces that may be acting that the effect of the other forces on the motion of the body, while the impulse lasts, may be neglected. The only details of the change in the motion of a body that can be determined, when the change is caused by an impulsive force, are the initial and final velocities of the body. The distance traveled during the impact is indefinitely small, the time interval is also indefinitely small; and hence, the acceleration produced is indefinitely large, since the change in velocity is a finite amount. Thus, the distance, time, and acceleration factors are indeterminate. There is, however, a definite (appreciable) change in velocity; although the manner in which the velocity changed during the interval of impact is unknown, and only the initial and final velocities can be determined. Therefore, the momentum (mass times velocity) of the body at the beginning and end of the impact period are definite quantities. Since these quantities are involved in the principles of impulse and momentum, problems which involve impact forces yield to this method of solution. Numerous text references are available that cover this subject (Refs. 45-49). The principles are summarized in the following equations:

$$\Sigma F_x \, \Delta t = M(v_{x_2} - v_{x_1}) \qquad (2\text{-}9)$$

$$\Sigma T_o \, \Delta t = I_o(\omega_2 - \omega_1) \qquad (2\text{-}10)$$

where

x = displacement in any direction, ft

F_x = force component in a given direction, x, lb

Δt = duration of the impact, sec

M = mass of the body concerned, lb-sec^2/ft

v_{x_2}, v_{x_1} = final and initial velocities of the body in the direction, x, fps

T_o = turning moment about an axis, O, ft-lb

I_o = mass moment of inertia of the body about axis, O, ft-lb-sec^2

ω_2, ω_1 = final and initial angular velocities of the body, rad/sec

The left-hand part of Equation 2-9 is the x-component of the linear impulse of the external force system acting on the body, and the right-hand member is the change in momentum of the body in the x direction. Similarly, in Equation 2-10, the left-hand member is the moment of the impulse (angular impulse) about the axis of rotation of the force system acting on the body; and the right-hand member is the change in the moment of the momentum (angular momentum) of the body with respect to the axis of rotation. These basic equations can be used to evaluate the impact forces resulting from the vehicle encountering obstacles.

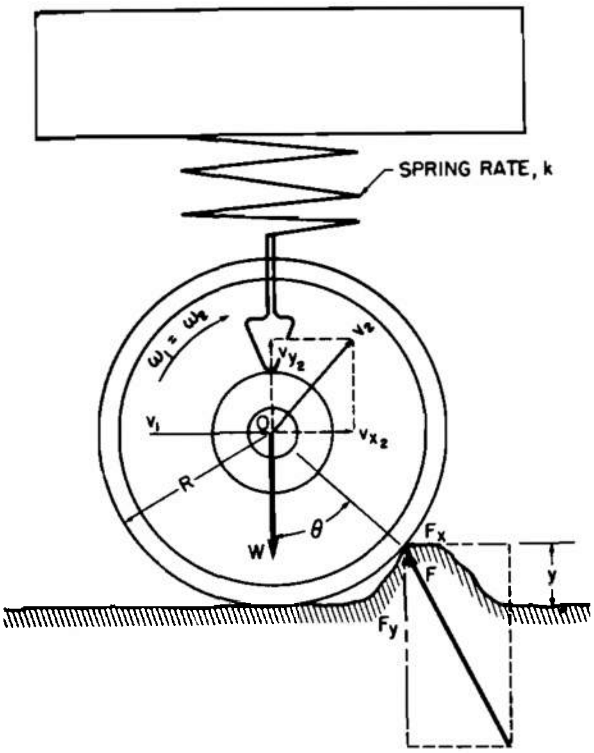


Figure 2-26. Forces on Wheel Striking Obstacle

The following illustrates the procedure as applied to a wheel encountering an abrupt bump in its path. Figure 2-26 shows the forces acting on the wheel at the instant of impact. Applying the principles given in Equation 2-9 we have

$$-F_x \Delta t = M(v_{x_2} - v_{x_1}) \qquad (2\text{-}11)$$

The magnitude of $v_2 = v_1$ and only the direction has changes, therefore

$$v_{x_1} = v_1$$
$$v_{x_2} = v_1 \cos \Theta$$
$$\Delta t = \frac{\Theta}{\omega_1} = \frac{R\Theta}{v_1}$$

and substituting these in Equation 2-11 results in

$$F_x = -\frac{Mv_1^2(\cos\Theta - 1)}{R\Theta} \qquad (2\text{-}12)$$

Similarly from the principles of Equation 2-9

$$\Sigma F_y \Delta t = M(v_{y_2} - v_{y_1}) \qquad (2\text{-}13)$$
$$v_{y_1} = 0$$
$$v_{y_2} = v_1 \sin\Theta$$

Substituting these values in Equation 2-13 and adding the forces acting along the y-axis results in

$$(F_y - W - ky)\,\Delta t = M(v_1 \sin\Theta - 0) \qquad (2\text{-}14)$$

substituting $\Delta t = \frac{R\Theta}{v_1}$ and solving for F_y

$$F_y = \frac{Mv_1^2 \sin\Theta}{R\Theta} + W + ky \qquad (2\text{-}15)$$

The last two terms on the right-hand side of Equation 2-15 represent the weight of the wheel assembly and the downward force developed by the spring system as the wheel rises over the obstacle, respectively. These two forces, particularly the former, are often omitted, because their contribution to the total impact force is minor in comparison to that of the first term. The ky term may or may not be significant depending upon the spring rate (or k factor). The following sample calculation illustrates this.

The total force F experienced by the wheel in going over the bump is the resultant of the two components, F_x and F_y, or

$$F = \sqrt{F_x^2 + F_y^2} \qquad (2\text{-}16)$$

Sample Calculation

Consider a tracked vehicle with 25-inch diameter road wheels moving at 30 mph when it encounters an abrupt bump 5 inches high. If the effective weight of the dual road wheel, tire, and road wheel arm assembly is 700 lb and the spring rate at the wheel of the torsion bar suspension spring is 1,800 lb per inch of deflection, calculate the impact force experienced by a front road wheel. (The impact loads are more severe on the leading road wheels because the action of the track tension to distribute the shock to adjacent wheels does not affect the leading wheels.) If we use Figure 2-26 as a reference and the same symbols as have been used thus far, the following is known

$$W = 700 \text{ lb}$$
$$M = \frac{W}{g} = 21.7 \text{ lb-sec}^2/\text{ft}$$
$$I_o = 12.7 \text{ ft-lb-sec}^2$$
$$R = 13 \text{ inches } (= 1.08 \text{ ft})$$
$$k = 1{,}800 \text{ lb/inch } (= 21{,}600 \text{ lb/ft})$$
$$y = 5 \text{ inches } (= 0.417 \text{ ft})$$
$$v_1 = 30 \text{ mph } (= 44 \text{ fps})$$

$$\Theta = \text{arc cos}\frac{(R-y)}{R} = \text{arc cos } 0.61538 = 52° = 0.9076 \text{ rad}$$

Substituting the appropriate values in Equations 2-12, 2-15, and 2-16 and solving, we find

$$F_x = -\frac{21.7 \times 44^2 (0.61538 - 1)}{1.08 \times 0.9076} \quad (2\text{-}17)$$
$$= 16{,}485 \text{ lb}$$

$$F_y = \frac{21.7 \times 44^2 \times 0.78823}{1.08 \times 0.9076} + 700 + 1800 \times 5 \quad (2\text{-}18)$$
$$= 33{,}114 + 700 + 9{,}000$$
$$= 42{,}814 \text{ lb}$$

$$F = \sqrt{16{,}485^2 + 42{,}814^2} \quad (2\text{-}19)$$
$$= 45{,}878 \text{ lb}$$

This is equivalent to a shock of over 65 g. Equation 2-18 illustrates the relative importance of the three terms. In this case, the magnitude of the last term is appreciable, and the second term is negligible. The designer must exercise engineering judgment in deciding whether terms should be omitted or not.

2-12.2.3 Ballistic Impacts

Ballistic impacts, i.e., impacts by projectiles and large fragments, are unique to the battlefield. Their severity depends upon the mass and velocity of the missile at the time of impact, upon the resistance offered by the surface that is struck, and upon the mass of the vehicle. In general, ballistic impacts against the vehicle are not a serious consideration when evaluating the loads experienced by the suspension system as Case II of the following Sample Calculations demonstrates. The momentum of the missile is easily absorbed by the mass of most combat vehicles; and thin-skinned vehicles, such as trucks, are easily pierced by the projectile, and thus, do not produce a sufficient change of momentum of the projectile to experience a serious loading of the suspension system.

The most serious effects of ballistic impacts are the direct damage they can inflict to suspension components. Pneumatic tires can be punctured; wheels, hubs, driving sprockets and idlers can be broken; tracks can be damaged or dislodged from sprockets or guides; and spring and shock absorber systems can be damaged. These components, however, are not generally protected—to do so effectively would greatly penalize the design from a weight and usable space standpoint.

The average force of the impact against a component can be calculated with reasonable accuracy by applying the principles of impulse and momentum discussed in paragraph 2-12.2.2. The mass of the projectile and its velocity with respect to the vehicle before and after the impact must be known; and by applying Equation 2-9, the magnitude of the impact $F\Delta t$ can be calculated. The units of impact are pound-seconds, since impact is a force acting for an interval of time. The average force of the impact can be found by dividing the value of the impact by the time interval Δt either measured, calculated, or assumed.

Sample Calculation—Case I

Consider the requirement of determining the average force experienced by the housing of a hydraulic shock absorber when it is struck by a 50-caliber round of ball ammunition. We will assume (a) a normal impact, (b) the projectile mushroomed against the housing and fell to the ground, (c) the velocity of the projectile at the time of impact was 2200 ft per sec, and (d) the weight of the projectile was 1¼ oz. Using the same symbols as in Equation 2-9 we can write the following:

$$F\Delta t = \frac{1.25}{16 \times 32.2} \times 2200 = 5{,}338 \text{ lb-sec}$$

and assuming $\Delta t = 5$ msec

$$F = \frac{5.338}{0.005} = 1{,}067 \text{ lb}$$

Sample Calculation—Case II

Consider now the impact of a 12-lb hypervelocity projectile against the side of a medium tank. Assume that the path of the projectile is horizontal and its velocity v_1 is 3,000 ft per sec just prior to the impact and that it ricochets upward at an angle of 45° to its original path with a departing velocity v_2 of 1,000 ft per second. Assume further that the point of impact on the tank is 8 ft above and 4 ft to one side of the vehicle's roll center, the tread width of the vehicle is 10 ft, the spring rate at the road wheels of the suspension system is 1,800 lb per inch of vertical deflection per road wheel, and the vehicle has six road wheels on each side. Let it be required to determine the effect of this impact upon the suspension system.

Again applying the principles of impulse and momentum, we know that the change of momentum experienced by the projectile results in an impulse (or impact) against the vehicle. Since this impulse occurs at some perpendicular distance from the roll center, it has an effective moment arm and, therefore, is an angular impulse which imparts an angular momentum to the sprung mass about the roll center. As the sprung mass begins to rotate about the roll center, one side of the suspension system experiences an increasing load while the other side experiences a corresponding load decrease. The couple thus developed is an angular impulse than counteracts the angular momentum of the sprung mass and returns it to its original position.

Figure 2-27 is a free body diagram showing the forces acting on a simplified representation of the vehicle. Again applying the principles of Equation 2-9 to the projectile and Equation 2-10 to the sprung mass of the vehicle we can write the following

$$\Sigma F_x \Delta t_1 = M_p(v_1 - v_{2_x}) \qquad (2\text{-}20)$$

$$\Sigma F_y \Delta t_1 = M_p(0 - v_{2_y}) \qquad (2\text{-}21)$$

$$\Sigma T_o \Delta t_1 = I_o(0 - \omega_2) \qquad (2\text{-}22)$$

where

M_p = mass of the projectile

Δt_1 = duration of the impact

I_o = mass moment of inertia of the sprung mass about the roll center.

Subscripts 1 and 2 designate conditions just prior to and immediately after impact, respectively. Subscripts x and y designate components along the x- and y-axes; for dimsions of terms, see paragraph 2-12.2.2.

Equation 2-22 can be written as

$$H(F_x \Delta t_1) - B(F_y \Delta t_1) = I_o \omega_2 \qquad (2\text{-}23)$$

where H and B are the distances indicated in Figure 2-27. Substituting from Equations 2-20 and 2-21, we find the angular momentum of the sprung mass about the roll center is

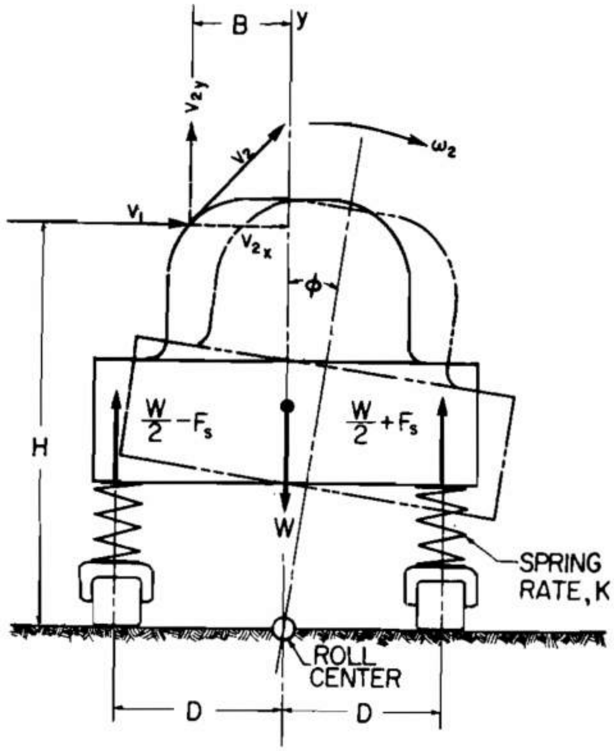


Figure 2-27. Effect of Ballistic Impact

$$I_o \omega^2 = M_p \left[H(v_1 - v_{2_x}) - B v_{2_y} \right] \qquad (2\text{-}24)$$

If F_s represents the total change in load experienced by one side of the suspension system (increased load on one side and decreased on the other), the restoring couple is

$$T_R = 2\,DF_s \qquad (2\text{-}25)$$

and the angular impulse experienced by the suspension system in absorbing the angular momentum of the sprung mass is

$$T_R \Delta t_2 = 2\,DF_s \Delta t_2 \qquad (2\text{-}26)$$

where Δt_2 is the duration of the angular impulse. Equating Equations 2-24 and 2-26, F_s can be evaluated as

$$F_s = \frac{M_p \left[H(v_1 - v_{2_x}) - B v_{2_x} \right]}{2\,D\,\Delta t_2} \qquad (2\text{-}27)$$

The term Δt_2 will be one-quarter of the natural period of the suspension system. Since vehicles are usually designed to have a natural frequency of between 1 and 2 cps, Δt_2 will be between ¼ and ½ sec. If ¼ sec is used, Equation 2-27 becomes

$$F_s = \frac{2M_p}{D}\left[H(v_1 - v_{2_x}) - B\, v_{2_y}\right] \quad (2\text{-}28)$$

This analysis is not rigorous in that it assumes that the spring rate of all road wheels is the same and that the effect of the lateral impact will be distributed uniformly to all road wheels. If a more exact solution is required, the basic Equation 2-27 can be readily modified.

The application of Equation 2-28 to our hypothetical ballistic impact problem results in the following:

$$\begin{aligned} F_s &= \frac{2 \times 12}{32.2 \times 5}\Big[\, 8(3{,}000 - 0.707 \times 1{,}000) \\ &\quad - 4 \times 0.707 \times 1{,}000 \,\Big] \\ &= 2{,}312 \text{ lb} \end{aligned} \quad (2\text{-}29)$$

This is obviously a negligible load, especially since it is divided among six road wheels and their suspension systems. If the assumption were made that the projectile did not ricochet and all of its momentum was transferred to the vehicle, Equation 2-29 would have been

$$\begin{aligned} F_s &= \frac{2 \times 12 \times 8 \times 3{,}000}{32.2 \times 5} \\ &= 3{,}578 \text{ lb (still quite negligible)} \end{aligned}$$

The angle of roll, ϕ, can be easily evaluated as follows:

$$\begin{aligned} \phi &= \frac{F_s}{Dk} \quad (2\text{-}30) \\ &= \frac{2{,}312}{5 \times 1{,}800 \times 12} \\ &= 0.0214 \text{ rad} = 1.2^\circ \end{aligned}$$

2-12.2.4 High Energy Blast

High energy blast is another consideration that is unique in the combat environment. Its effect on the suspension system can result in serious damage to components. Wheels, tracks, and tires can be blown off; suspension linkages can be damaged, misaligned, or destroyed; and hydraulic or pneumatic cylinders and shock absorbers can also be destroyed. Here too, since protective measures would exact an unacceptable price if really effective, very little is done to prevent this damage. A certain degree of resistance to blast is achieved through the selection of high strength materials and through the application of design techniques aimed at the development of rugged components. In general, when the suspension system is designed to function satisfactorily in its primary role on the battlefield, it will possess considerable inherent resistance to high energy explosives.

2-12.2.5 Shocks Encountered During Airdrop

Shocks experienced by vehicles during an airdrop are of three types; namely, the snatch force, the opening shock, and ground impact shock. The snatch force occurs at the instant the parachute suspension lines are fully deployed and just prior to the inflation of the canopy. It arises from the comparatively rapid deceleration of the deploying parachute in relation to the slow deceleration of the suspended load. Thus a differential velocity exists which is brought to zero when the lines become fully extended—hence the shock. The snatch force shock precedes the opening shock by anywhere between 0.1 to 1.0 sec. At high aircraft speeds, particularly where modern canopy designs are used, snatch force shocks can exceed opening shock. Furthermore, opening shocks can be reduced through the application of special reefing, venting, and collapsing techniques to the parachute. Snatch forces, however, are much more difficult to control and will become the limiting factor in future airdrop operations (Ref. 43). A method for calculating the snatch force is given in Refs. 43 and 44. The same references give methods for calculating opening shocks; although, for the reasons mentioned previously, opening shock is generally not of primary importance where modern canopies are employed.

Landing shock is quite another matter. The objective of aerial delivery systems is to get the material being delivered to the ground in the least possible time without damage, so as to achieve maximum drop accuracy, minimum dispersion of dropped loads, and with minimum effect of wind drift. In addition, speed minimizes the time during which the aircraft is vulnerable to enemy fire and reduces the chances of the enemy pinpointing the drop zone. Thus, the ideal aerial delivery system would dispense with all deceleration devices, such as parachute canopies or retro-rockets, and would be capable of absorbing all of the landing shock on impact. Since such a system is not feasible for bulky or heavy cargoes, an acceptable compromise is employed which combines a parachute canopy, which gives a degree of deceleration, with shock-absorbing devices that absorb the energy of impact. Most aerial delivering systems are designed to attain a rate of descent of about 25 ft per sec. This is varied somewhat in accordance with the fragility of the cargo, the requirements of the tactical situation, and the efficiency of the impact absorption system used. Ground impact forces of up to 100 g have been measured on cargo platforms descending at approximately 30 ft per sec.

Vehicle suspension systems are not generally designed to withstand impacts of this magnitude. Therefore, when vehicles are made ready for airdrop, suitable blocks and lashings are employed to completely immobilize the suspension system and to isolate it from landing shocks. Otherwise, suspension system components (tires, wheels, wheel rims, bearings, axle housings, bump stops), and even the vehicle frame or body, may suffer serious damage unless designed for these loads. With proper rigging and shock absorption techniques employed, the airdropped vehicle will still experience a landing shock of approximately 16 g.

SECTION IV MILITARY CHARACTERISTICS

2-13 DEFINITION

The general term "military characteristics" refers to those physical and operational features of equipment which determine its ability to perform desired military functions. In engineering circles concerned with the development of new and improved items of military equipment, Military Characteristics (MC's) is the designation given to an official Department of the Army document that specifies the physical and operational requirements desired of an item of materiel that is to be developed. These requirements are given in general, nontechnical terms that indicate the physical and operational characteristics desired but do not give technical specifications.

2-14 DEVELOPMENT

The initiation of the development of a new item of materiel for the Army begins with a statement of a military requirement. This statement may originate anywhere in the military or civilian society where a specific need is first recognized. It is channeled through appropriate military agencies where the requirement is appraised in the light of the budget, anticipated military situation, anticipated enemy capabilities, current and anticipated technological capabilities, logistical and troop training implications, and compatibility with current and future materiel, tactical doctrine, tripartite, Navy, Air Force, and Marine Corps development activities, and treaty agreements with allied nations. This comprehensive appraisal of the original requirement results in either a rejection or in an official statement, approved by the Department of the Army, expressing a military need for the development of the new materiel. Objectives, requirements, and specifications are given in only the most general terms, at this time, since the feasibility of the requirements has not been adequately determined.

Following the issuance of this official statement of need, various using agencies, technical services, and field agencies conduct comprehensive studies to determine the feasibility of the general concept and to establish the essential and also the desired requirements of the new equipment. These activities eventually culminate in the formulation and issue of the approved Military Characteristics for the required item. A detailed explanation of the preliminary steps that are involved in the preparation of the MC's and the agencies responsible for these steps are given in Refs. 50, 51, and 52.

2-15 NONTECHNICAL NATURE OF MC'S

It has already been stated that the requirements enumerated in the Military Characteristics are given in general, nontechnical terms that indicate the desired physical and operational characteristics but are not given as technical specifications that can be applied directly in the construction of a prototype. This is done for several reasons. It allows the designer maximum leeway as to the possible courses his design may take and it allows different design groups to pursue different design courses. The result is more than one solution to the overall requirement, permitting the different approaches to be evaluated with respect to each other to determine which is the most desirable. Furthermore, although various technical agencies were involved in the preparation of the MC's, their primary concern was the verification of the statement of need, a determination of the physical and operational requirements that must be met by the new equipment in order to fulfill the recognized need, and a general consideration as to the feasibility and practicability of the overall concept. The primary concern of the designer is to determine the best utilization of space, materials, equipment, techniques, and dollars to meet the requirements of the Military Characteristics. Thus, at least in theory, his design should be more feasible, more practicable, and more economical than the preliminary concept evaluated by the agencies that developed the MC's.

2-16 FORMAT AND CONTENTS

Military Characteristics are prepared in accordance with a standard format given in Appendix III of AR 705-5 (Ref. 51). The information and requirements are grouped into four categories or sections. Section I presents a general statement of the requirement of the equipment desired; a brief narrative of the operational and organizational concepts planned for the desired equipment; information regarding Navy, Air Force, Marine Corps, and Allied interests or developmental activities on a similar item; a statement relative to the feasibility of the development; and background information covering such items as relevant knowledge gained from past experiences, related equipment, industrial capabilities, possible parallel approaches, and personnel implications.

Section II of the MC's deals with the operational characteristics of the equipment to be developed. It defines and describes the general configuration and classification of the materiel that is required, e.g., "a highly mobile and maneuverable, cargo carrying type of wheeled vehicle." The crew and cargo space requirements are stated, and any contemplated alternate uses of the desired materiel that are to be considered in the development of the overall configuration are indicated. For example, the proposed cargo carrier in the foregoing illustration might be also considered for alternate functions as a wrecker, missile launcher or transporter, self-propelled carriage for artillery, or a self-propelled van for electronic equipment. Limits of overall size and weight are specified as are the method of operation and required performance characteristics along with specifications on transportability, durability, reliability, vulnerability, and maintenance requirements. The latter includes specifications regarding in-storage maintenance as well as the degree of maintenance to be performed by the user, the expendability of the components, and the ease of maintenance required. As guidance for the designer in making trade-off decisions during the development of the design, an indication is usually given of the relative importance of the requirements listed. This is usually done by indicating the various requirements as "essential" or "desirable."

Section III of the Military Characteristics deals with various associate considerations; such as modifications to or development of associate equipment that will become necessary as a result of the new device or equipment (includes training aids); consideration relating to special requirements of nuclear, bacteriological, and chemical warware; safety criteria; a recommended priority of development; special time considerations if pertinent; and the remarks of other interested agencies that may have a bearing upon the subject.

Section IV of the MC's gives an order of priority to the major characteristics. Since most design is a process of compromise in which the designer must forfeit the attainment of certain ends for the attainment of others that are not compatible, this section gives him guidance in deciding which features should receive design emphasis. Furthermore, the cost of producing an end item, though not as important a factor in the production of military materiel as in the production of commercial materiel, must be within an acceptable range. Thus, information regarding an acceptable end item unit cost is often included in Section IV of the Military Characteristics as a development objective.

SECTION V DESIGNING FOR MAINTENANCE

2-17 CATEGORIES AND ECHELONS OF MAINTENANCE

The maintenance concept approved by the U. S. Army is given in AR 750-1 (Ref. 53). The term maintenance, as it is used by the Armed Forces, is all action taken to retain materiel in a serviceable condition or to restore materiel to serviceability. This includes such actions as inspecting, testing, servicing, classifying as to serviceability, repairing, overhauling, rebuilding, modifying, and modernizing. Maintenance is divided into four major categories; namely, organizational, direct support, general support, and depot maintenance. These divisions result in maintenance strata that relate the difficulty of the maintenance tasks to the skills of the personnel performing the maintenance and to the facilities available at each level.

Organizational maintenance is that maintenance which is authorized for, performed by, and the responsibility of a using organization on its own equipment. Normally, this maintenance consists of inspecting, cleaning, servicing, preserving, lubricating, adjusting, and the replacement of minor parts that does not require highly technical skills. It encompasses that degree of maintenance which is performed by the equipment operator or the operating crew, as prescribed by pertinent technical publications, or that can be performed by specially trained personnel in the using organization (organizational mechanics and technicians).

Direct support maintenance (formerly designated as third echelon of field maintenance) is that maintenance which is authorized for and performed by designated maintenance activities and maintenance units that are in direct support of the using organizations. Normally, this category of maintenance is limited to the replacement of unserviceable parts or subassemblies in order to restore an end item (vehicle, etc.) to serviceability on a return to user basis.

General support maintenance (formerly designated as fourth echelon of field maintenance) is that maintenance which is authorized for and performed by units that are organized as permanent or semipermanent shops to serve lower echelons of maintenance. Normally, these units overhaul or repair material to required maintenance standards in a ready to issue condition.

Depot maintenance is that maintenance which is required for the repair of materiel that requires a major overhaul or complete rebuilding of parts, subassemblies, assemblies, or the end item as required. Additional information on maintenance responsibilities, shop operations, definition of maintenance terminology, and maintenance criteria are given in Refs. 53-56.

2-18 THE DESIGN ENGINEER'S RESPONSIBILITY

It has long been recognized that only marginal improvements to the Army Maintenance System can result from future improvements in operational, organization, and personnel activities. The only prospect available for an appreciable reduction in the maintenance requirements of military vehicles is in design and development. Design engineers, while striving for performance, must give adequate consideration to the maintenance requirements and the maintainability characteristics of their design *during the equipment design phase* of the development program. Failure to do so results in costly training programs for maintenance personnel, long equipment downtimes because of maintenance services, neglected maintenance leading to early equipment failure, and an increased logistical support burden during the life of the vehicle. Maintenance-minded design engineers must be concerned with the following objectives:

(a) Improving the reliability of the end item so as to reduce the need for maintenance.
(b) Reducing the frequency of cyclic preventative maintenance.
(c) Improving and simplifying maintainability to reduce equipment downtime.
(d) Reducing the logistical burden necessary to support the equipment in the field.

The design engineer is the controlling factor in determining reliability, simplicity of operation, and ease of maintenance. These qualities can be attained only through serious considerations of the maintenance problem. He should acquire sufficient information from maintenance engineers regarding the capabilities and limitations of the operating and maintenance personnel that will be assigned to the end item of equipment, the anticipated effects of environmental factors upon maintenance, and what maintenance support will be available. He should have frequent consultations with maintenance engineers during the design phase to assess the maintainability of his design. The general subject of designing for maintainability is covered quite comprehensively in Ref. 57. Some pertinent points that are specifically applicable to the design of suspension systems, from this and other sources, are given in the paragraphs which follow.

2-19 MAINTENANCE DESIGN REQUIREMENTS

2-19.1 GENERAL MAINTENANCE OBJECTIVES

In January 1959, the U. S. Army Tank-Automotive Center issued an official document entitled *The Maintenance Criteria for Ground Vehicles* which established certain maintenance objectives that were to be the design goals for all agencies engaged in the design and development of military vehicles. Since that time, these objectives have been approved by the Department of Defense and incorporated in MIL-STD-1228, *Maintainability Criteria for Tank-Automotive Materiel* (Ref. 56) for mandatory use by the Departments of the Army, Navy, and Air Force effective 27 September 1962. These maintenance criteria are as follows:

The Design Facility shall have as [its] maintenance goal that ground vehicles will accomplish the following environment, unless otherwise directed:

(a) Wheeled, tactical vehicles: 25,000 miles without field or depot maintenance.
(b) Tracked vehicles: 5,000 miles without field or depot maintenance.
[Field and depot maintenance are defined in paragraph 2-17.]

Maintenance Skill Level. The Design Facility shall design maintenance procedures which can be accomplished by an individual with general tenth grade education and two years of general military experience.

Maintenance Manhours. The Design Facility shall have, as a maintenance goal, that the total scheduled and unscheduled maintenance manhours shall be a constant ratio and not to exceed the following, within the requirements of [a or b of the foregoing]:

(a) Self-Propelled Wheeled Vehicles. Seven percent (7%) of the operational hours. The average distance negotiated by the vehicle shall be considered to be twenty (20) miles for each hour of operation.
(b) Self-Propelled Tracked Vehicles. Twenty percent (20%) of the operational hours. The average distance negotiated by the vehicle shall be considered to be ten (10) miles for each hour of operation.

2-19.2 MAINTENANCE DESIGN CRITERIA

The following design criteria have been extracted from MIL-STD-1228 to be considered for incorporation into all design and development projects where it is practical and economical to do so:

(a) Use of modular or throwaway components and techniques.
(b) Reduction in weight and quantity of parts, components, and assemblies.
(c) Simplification of operator and maintenance functions.
(d) Culmination of component design prior to its application.
(e) Increased use of standardized, pre-tested components.
(f) Wear-out limits defined to a degree that shows a definite margin of performance over the maintenance objective.

(g) Use of self-lubricating principles where practicable.
(h) Use of sealed and lubricated components and assemblies where feasible.
(i) Use of built-in testing and calibration features for major components.
(j) Use of self-adjusting mechanisms where feasible.
(k) Use of gear-driven accessories to eliminate belts and pulleys.
(l) Minimum number and complexity of maintenance tasks (i.e., calibration, adjustments, inspections, etc.).
(m) Maximum use of simple design.
(u) Design for rapid and positive recognition of malfunctions or marginal performance.
(o) Design for rapid and positive identification of the replaceable defective components, assemblies, or parts.
(p) Design to eliminate torque specifications, at organizational level (see paragraph 2-17) and minimize need for all other torque specifications.
(q) Design to minimize skills and training requirements of maintenance personnel.
(r) Design to minimize the types and number of tools and test equipment (both special and standard) required to perform maintenance.
(s) Design for optimum accessibility to all systems, equipment, and components requiring maintenance, inspection, removal, or replacement.
(t) Design for maximum safety and protection for personnel and equipment involved in the performance of maintenance.
(u) Design to minimize the net mean time required to accomplish scheduled and unscheduled maintenance to assure operation availability.

2-20 ACCESSIBILITY

Accessibility is a fundamental requirement for maintainability and must be designed into the equipment; for adding access provisions for maintenance purposes after the equipment is built is inefficient, costly, and often inadequate. Accessibility is defined as the *relative ease* with which an assembly component can be reached for inspection, service, repair, or replacement. If it can be reached quickly, requiring the use of a few or no tools and only a few simple steps, the item is accessible; if it requires many tools, special tools, many or difficult operations, the item is inaccessible even though it is possible to eventually reach it. After all, since all equipment originated through the assembly of individual components, it can be disassembled through a reversal of the assembly process. This by no means constitutes accessibility, however, as the term is used with respect to maintenance. Furthermore, the disassembly or removal of parts that are in the way of easy access to a part needing maintenance is highly undesirable, especially when done under field conditions. Adequate space is usually not available for laying out parts as they are removed. This increases the possibility that they will be lost, damaged, or contaminated; and further malfunctions will be introduced into the system.

Inaccessibility also has psychological aspects. The more difficult or involved a maintenance task is, the more readily will an operator or mechanic put it off in preference in less demanding tasks. Periodic maintenance activities, such as checks, adjustments, or general troubleshooting may be unduly postponed or neglected entirely. Thus, inaccessibility is a human factors engineering problem as well as a problem for the design engineer.

Another aspect of accessibility often overlooked is the "whole body" accessibility features involved in a particular maintenance operation. It is not enough to provide a suitable access opening to a component for the insertion of a tool, a light, a hand, etc.; but consideration must be given to the position of the maintenance man with respect to the equipment while he is performing the particular maintenance task. Can he perform the task while standing on the ground; does he need foot holds and hand holds on the sides of the vehicle; is perhaps an impossible body position required? If an awkward body position cannot be avoided, can the man exert the force necessary to accomplish the maintenance task from the awkward position?

Does the man have visual access as well as access for his hands or tools? The designer must determine whether it is necessary for the technician to see what he is doing while he is doing it. If this is a requirement, it may be necessary to make the access large enough to accommodate hand, arms, tools, and still leave an adequate view; or an auxiliary viewing port may be necessary.

The environment in which maintenance tasks will be performed must also be considered by the designer when determining the size of access openings. Access for a gloved or mittened hand is considerably larger than for a bare hand. Similarly, the wearing of gloves or mittens seriously limits the dexterity and skill with which a technician can perform his duties; certain simple tasks may be impossible to perform while wearing arctic mittens. The wearing of arctic clothing, in general, creates many problems. Men become less agile and bulkier in arctic clothing; their hulking bodies become wedged in narrow hatches; ponderous feet stumble awkwardly on steps and ladders and fumble with small, closely-spaced pedals; and awkward hands are unable to operate intricate hand controls with deft and precision. Anthropometric data for normally clad and arctic clad personnel can be found in Ref. 58.

2-21 DESIGN SUGGESTIONS FOR MAINTAINABILITY

The following design suggestions are recommended for improved maintainability of vehicle suspension systems. Obviously, all of the suggestions given are not applicable to every type of vehicle, nor is this list a complete checklist of "things to do." It is merely a compilation of some of the suggestions made by vehicle designers, maintenance engineers, and maintenance technicians as a guide in future designs.

(a) Provide maximum accessibility to the following:
   (1) All servicing operations, particularly those required daily.
   (2) Drain plugs and bleeder valves in hydraulic assemblies or systems.
   (3) Lubrication fittings.
   (4) Pneumatic tire valves
   (5) Track tension adjusting devices.

(b) Provide for visual inspections without entailing the disassembly of components.

(c) Provide for repair by the exchange of assemblies or components (the throwaway concept).

(d) Use self-adjusting devices wherever practical.

(e) Make provisions for locking manually adjusted devices without disturbing the adjustment.

(f) Secure covers and access plates with the minimum number of fasteners compatible with the sealing requirements. Use quick release type fasteners for cover and access plates wherever suitable.

(g) Use same size and type grease fittings throughout an end item. Locate grease fittings to facilitate servicing with standard grease guns without special attachments.

(h) Protect lubrication points against accidental damage.

(i) Minimize number of lubrication points needed.

(j) Make provisions for the removal of all components weighing 50 lb or more with the use of slings and hoisting equipment. Provide suitable lifting eyes, or tapped holes for lifting eyes, on all heavy components whose general shape is not convenient for the attachment of slings.

(k) Identify complex hydraulic or pneumatic systems by a system of color coding to assist maintenance personnel in quick recognition.

(l) Provide means for checking pressures without disconnecting lines or fittings.

(m) Design track guards, skirting plates, fenders, mud flaps, etc. for quick removal in the field with standard tools.

(n) Design track assemblies for ease of installation, removal, and adjustment with standard tools.

(o) Design wheels and tires so they can be changed without special tools or equipment.

(p) Provide means for lifting the road wheel and road-wheel arm, such as by extending the spindle.

(q) Make adequate provisions for removing final drive assemblies of tracked vehicles by hoisting from directly above the unit. Permanent, over-hanging fenders are not desirable.

(r) Indicate wear pattern and degree of wear permissible directly on the sprocket to eliminate need for special gaging equipment.

(s) Indicate maximum permissible wear of road wheel flanges due to center guides directly on the flanges to eliminate need for special tools.

(t) Design track-tension adjusting devices so they can be turned with common tools. Provide maximum accessibility.

(u) Provide for easy removal of wheel bearings on both tracked and wheeled vehicles.

(v) Provide for easy removal of shock absorbers.

SECTION VI DESIGNING FOR PRODUCTION

2-22 DEFINITIONS AND GENERAL DISCUSSION

The term "production" is basically defined as the process of gathering and fabricating raw materials into finished articles that satisfy some human want. Inasmuch as human wants involve people, the demand for the finished articles resolves into a matter of quantity manufacture; hence the term "production" connotes mass manufacture, or the manufacture of large numbers of identical units. Military wants, too, are demands for large quantities of finished articles; and so the concept of mass manufacture is also applicable here.

"Mass production" is a term often used with reference to quantity manufacture, and it is generally interpreted as involving outputs of tens of thousands, or even millions, of units. This need not always be the case, however, for the principles of mass production can be applied to any quantity of units. The underlying difference between mass production and unit production is in the application of the principles of interchangeable manufacture rather than those of the "custom" or hand-fitted and selected assemblies. The physical dimensions and surface finishes of mating parts must be so specified as to assure proper mating and ease of assembly.

Due consideration must be given, however, to the inherent variations present in each process which set the economical dimensional tolerance limits and surface finish obtainable with each. The practice of specifying unnecessarily close tolerances and high grade finishes, without giving serious consideration to the factors attendant to maintaining them, invariably results in excessive costs due to reduced output, extra operations, high cost of tool upkeep, and a high rejection rate or scrap loss. In general, any tolerance closer than plus or minus 0.005 inch should be closely scrutinized to determine whether such accuracy is absolutely essential. Once the degree of accuracy needed is established, it should be compared with the accuracy normally obtainable with the processing method being contemplated for the part under study. If the contemplated process cannot be relied upon for the accuracy or surface finish required, the part should be redesigned for a more suitable production process; or, where the increased costs can be absorbed, a more sophisticated production method can be utilized to obtain the desired result.

The quantity to be produced, together with the general design characteristics of the parts, influence the selection of the manufacturing methods for the most economical production of the units required. Regardless of quantity, however, the problem facing the design engineer remains the same: to produce the desired product in the required quantities, at minimum cost, and possessing all of the required physical and functional attributes. The minimum cost factor does not imply "cheap" components but, rather, components that are adequate for the intended service and which utilize in their manufacture those processes that offer the desired results at the most economical expenditure of time, materials, and labor.

The design engineer engaged in the design of military equipment is faced with an additional consideration which is more unique to his sphere of activity than that of the designer of commodities for the general civilian market. This is the requirement that his design be compatible to efficient and economical manufacturing techniques under the limited production required in time of space and, also, to the very high rate of production, conducted with relatively unskilled workers, which is required in time of war. When establishing material specifications, too, the military equipment designer must bear in mind that the supply of many highly desirable materials becomes critically short in time of war. The specification of these critieal materials may become a serious production obstacle at a time when production efficiency is paramount.

The design process embraces two broad fields; namely, *functional design* and *production design*. Functional design is that design activity concerned with the conception of ideas and their development into a mechanism, or other end item, that will perform a required function. In general, the primary aim of functional design is to accomplish the desired purpose, function satisfactorily, and meet the necessary practical performance requirements. Production design, on the other hand, usually follows functional design and is comprised of the study and redesign of the functional design with the aim of simplifying manufacture. It is a process of fitting the parts for the simplest, most rapid, and most economical method of production.

Considerable time and money can be saved, however, by merging functional and production design; or where a complete merging is not possible, by giving careful consideration to the producibility of the parts and avoiding manufacturing problems through the application of good design techniques. Good design must take into account the broad limitations inherent in the various processing methods that are available and must capitalize on their strong characteristics. Some unusual special parts may necessitate novel production methods to attain economy of manufacture; but usually, these will be in the minority. Most designs can be produced with the highly satisfactory methods that are already available.

It is highly imperative, therefore, for the up-to-date designer to keep abreast of modern production methods. However, the rapid rate of technical advancement in the basic processing technique, and the development of new techniques during recent years, has made this task rather difficult. Nevertheless, the point has often been made that a mechanism or part that is impractical to manufacture is just as much an indication of poor design as one that fails to function properly. A comprehensive discussion of the nature, fundamental characteristics, advantages, limitations, and design considerations of the basic production methods applicable to the manufacture of military equipment, mechanisms, and components is far beyond the scope of this book, even if it were to be presented in a synopsized form. Numerous excellent references are available on this subject, however, a few are listed at the end of this chapter (Refs. 59 to 66).

Economical and thoroughly practical production-designed parts, therefore, must be based upon a wide knowledge of the various basic manufacturing methods. Primarily, then, by concentrating on basic operations available through the many readily tooled and adapted production methods rather than solely on the product or part to be made, maximum economy, flexibility, accuracy, and speed in manufacture can be realized. Without this concentration on basic production methods during the initial design phase, a long and arduous second phase is necessary during which the parts must be redesigned or otherwise changed to bring them into the scope of economy through a roundabout means from the shop, production planning, or tool design department; primarily because the designer ignored, or was unaware of, the limitations and capabilities of methods available.

It is readily recognized that, under particular circumstances, some designs will require the development of new processing methods. The development of a new method of processing, however, should be demanded only after a thorough analysis has exhausted the possibilities of present methods. The so-called standard or universal production machines and methods provide the most versatile and readily adaptable means for the manufacture of most parts. Tooling and equipment for these basic processes are most easily provided and can be quickly brought into play. The designer should therefore plan, wherever possible, to take advantage of the economies afforded by these universally available methods.

Recognition of the economic possibilities and limitations of the various basic manufacturing methods available, thus, becomes one of the most important considerations for the designer. Reduced manufacturing costs, simplified processing, decreased material waste, and design improvement must originate in the engineering department and depend primarily upon the extent to which the parts have been processed so as to lend themselves to minimum cost manufacture.

2-23 GENERAL RULES FOR PRODUCTION DESIGN

Sound design practice takes careful cognizance of functional requirements, production requirements, and the characteristics of the available processing methods to achieve maximum economy and efficiency in manufacture. This necessitates consideration of a number of basic rules which are outlined below. The order in which they are given in no way indicates their relative importance.

2-23.1 DESIGN FOR SIMPLICITY

Design for maximum simplicity of all functional and physical characteristics is required, and production cost has a direct relation to the complexity of the design. This point is rather self-evident, but it is often overlooked or neglected by designers. Too much attention to the functional requirements, to the exclusion of adequate consideration of production problems, is often the reason for this. After the designer has developed a concept that satisfies the functional requirements, he should take additional time to see how his design can be simplified. Service life of components is generally improved considerably where parts can be simple and sturdy. Furthermore, where the design configurations of parts are simple and uncomplicated, the computation of stresses and the load carrying capacities of the components are more certain.

Ease of assembly and maintenance requirements should be evaluated in terms of design complexity. Combining parts into one part obviously reduces the number of parts that must be assembled on the assembly line and is an acceptable procedure provided the resulting part does not become complex or costly to produce. The opposite possibility should not be overlooked, i.e., breaking down a complicated part into two, or more, simple parts. The resulting parts may be cheaper to produce and assemble.

2-23.2 DESIGN FOR MOST ECONOMICAL PRODUCTION METHODS AVAILABLE

Once an end item is conceived, the various methods by which each part can be manufactured should be considered seriously. Such factors as quantity to be produced, physical features, accuracy required, material, production speed, equipment available, and amount of scrap produced should be carefully evaluated. The ratio of the weight of material required per piece to the weight of the finished part often is a clue to possible cost reduction. For example, a part machined from bar shock may require considerably more raw material than the same part made from a forging or a die-casting, and a considerable reduction in machining time will be added to the savings. The designer should be thorough in his consideration of the various available production methods. Often, the most satisfactory method from a cost and efficiency viewpoint is not a very well-known or widely used one; e.g., electroforming, chemical milling, or die rolling. Therefore, no possibility should be overlooked. It is a worthy challenge to the ingenuity of the designer to recognize the most economical and efficient production method and to be able to modify his design to take advantage of it.

2-23.3 OBSERVE LIMITATIONS INHERENT IN AVAILABLE PRODUCTION METHODS

The inherent limitations in each manufacturing process should be determined and compared to the requirements of the design. If the requirements are unnecessarily stringent, they should be relaxed; or perhaps the design can be modified to make certain features less critical and thus suitable for a particular production process that is otherwise desirable. Close dimensional tolerances and high quality surface finish specifications are often beyond the capabilities of certain production processes, such as flame cutting, contour sawing, or sand casting. Requirements for pressure-tight components may make it unwise to use sand castings; rejections may be costly—especially if machining operations are necessary before the parts can be adequately tested. Similarly, processes that result in porous structure, coarse grain size, blow holes, or slag inclusions—such as are normally associated with sand castings—should not be specified where high-quality surface finishes or plating is contemplated. Grinding and buffing costs will run high and will uncover defects in the material which will cause the items to be rejected.

2-23.4 SELECT MATERIAL TO SUIT PRODUCTION METHOD AS WELL AS THE DESIGN REQUIREMENTS

When specifying the material for a part, it is important that the designer give careful consideration to the machinability, workability, or formability (whichever is applicable to the part) of the material to be used. Designs that are most adaptable to manufacture on the screw machine or turret lathe-type of production equipment had best be made of a free machining material; otherwise machining costs will rise and quality of finish will drop. Oftentimes, a substantial savings can be affected by specifying a more expensive, but better machining. material over an inexpensive but troublesome material. The designer should bear in mind that the ratio of material cost to labor costs on most parts requiring machining is usually quite low. Thus, maximum emphasis should be given to the reduction of labor time. For example, alloys and aluminum alloy forgings cost somewhat more than steel, but the time required for machining aluminum alloy parts is approximately one-third that required for steel parts of similar design. Magnesium alloys provide an even greater machining advantage requiring only about one-sixth of the machining time required of steel.

When the design requirements are such that a material of poor machinability must be used, other production methods should be considered where machinability is not a factor. Perhaps the material has a high formability rating, or a suitable material can be selected that has a high formability rating, and the part can be produced by such processes as cold heading, stamping, drawing, rolling, or pressing from metal powders.

In situations where design requirements allow very little or no leeway in the selection of materials, the design configuration should be carefully evaluated with regard to the limitations and requirements of the applicable production methods. For example, foundry processes require that the design possess certain physical attributes such as uniform sections, absence of sharp corners, simplified gating and location of risers, simple parting and coring, etc. Forging processes require proper consideration be given to draft, corner and fillet radii, webs and ribs, and material considerations. In situations where the necessary material is extremely hard to machine, production processes must be selected that yield close dimensional tolerances and smooth surface finishes to minimize machining operations as much as possible.

2-23.5 DESIGN FOR MINIMUM NUMBER OF SEPARATE OPERATIONS

It is quite obvious that the fewer the operations required to produce a part the lower will be the production cost. Every feature that requires some operation be performed should be carefully scrutinized to determine whether it is really necessary, merely decorative, or entirely extraneous. This may lead to a modification of the feature so that it can be produced in an already existing operation, or it may be possible to eliminate it entirely. Sharp corners (and radii, too,) often require extra operations to produce and often involve special tools. If not really necessary to the function of the design, they needlessly increase production costs.

A very important consideration in quantity production is the number of times an item must be repositioned during its production cycle. Repositioning or relocating in supplementary jigs or fixtures wastes time and may introduce errors or inaccuracies. Castings, forgings, and weldments should be designed so that they can be machined from as few directions as possible. Finished openings, threaded or reamed holes, and other features requiring machining should be placed in the same plane as the major machining operations, if possible, to allow completion of the part at one set-up.

2-23.6 DESIGN FOR EASE OF HOLDING, SET-UP, AND HANDLING

One consideration that is overlooked by the designer is how the part will be held or located in a fixture during the processing operations. It may become necessary to add lugs, alter the contour, or otherwise modify the design to make handling, holding, and set-up an easier and less costly operation. Some degree of familiarity with the processing functions involved and the various aspects of fixturizing, holding, and feeding of parts is necessary on the part of the designer to affect reduced jig and fixture costs and thus reduce production costs per part. Where a designer lacks such knowledge, he should consult tool designers and industrial engineers; in fact, such consultations are advisable in any situation where extensive production runs are involved.

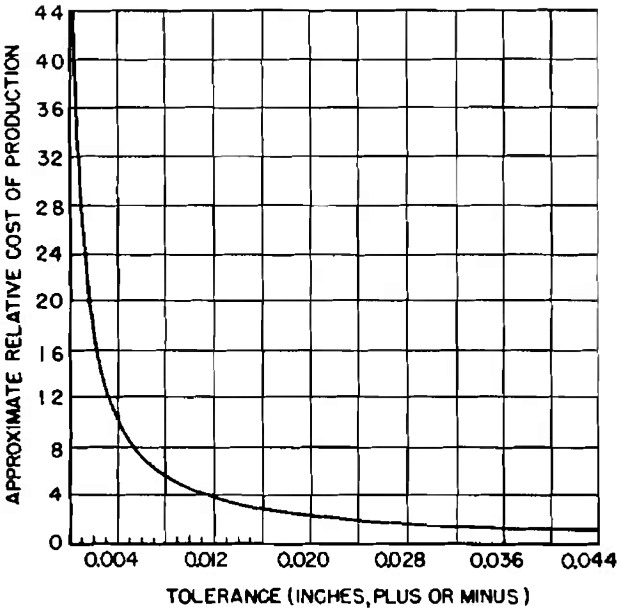


Figure 2-28. Relationship Between Production Cost and Accuracy of Parts (Ref. 61)

2-23.7 SPECIFY FINISH AND ACCURACY COMMENSURATE WITH DESIGN REQUIREMENTS AND WITH PRODUCTION METHOD CONTEMPLATED

The principle of interchangeable manufacture—upon which low cost, mass production techniques are based—depends on the reproduction of mating parts to predetermined dimensional tolerances. However, production costs increase exponentially as dimensional tolerances are tightened, as illustrated graphically in Figure 2-28. The chart is intended merely to show the general relationship between production cost and various degrees of accuracy, but the importance of judicious selection and specifications of tolerances is readily apparent. In actual practice, each process must be analyzed individually with reference to the particular production item. By designing an item to be particularly suited to a specific process, the desired degree of accuracy is often obtained without an undue cost penalty.

In general, fine surface finishes and close tolerances do not result with operations where metal removal is extremely heavy. Once the functional specifications and the necessary dimensional accuracy for satisfactory performance are determined, it becomes relatively simple to decide upon the production methods that will be required to produce the desired part. The required production quantities will usually narrow these further to but one or two methods, depending upon the flexibility of the design.

In order to be assured of the lowest production costs and highest rate of production, neither dimensional tolerances nor surface quality should be specified to closer limits than function, design, or interchangeability dictate. The roughest acceptable finish should be contemplated and specified. If at all possible, the design should be so arranged that the inherent average finish quality characteristic of the basic production method used would be acceptable. Figure 2-29 shows the basic metal removing and finishing processes with their full range of surface finishes encountered in general commercial practice. Also shown are the narrower ranges of surface finish normally expected in average production from well-kept machines and tools without incurring undue costs. An explanation and discussion of surface roughness, waviness, and lay can be found in MIL-STD-10A (Ref. 67) and in Standard B46, 1-1947 of the American Standards Association.

The lower portion of Figure 2-29 shows the range of accuracy that can be normally expected for each of the production methods. It is obvious that there must be a relationship between surface roughness and dimensional accuracy, but this fact is often overlooked by the designer.

Another factor of surface finish that is often overlooked by the designer is that each process produces its own characteristic type, or types, of surface pattern. The effect of these surface patterns should be considered with respect to the functional requirements of the design. It is possible to have a surface that is too perfect. Table ways can be finished so smooth and flat to the point where lubrication fails to enter between the surfaces. Similarly, hydraulic valves can be fitted so closely by lapping that they will fail to function under normal system pressure due to the absence of a lubricant-retaining scratch pattern.

A characteristic range of surface roughness figures for various applications is given in Table 2-9. These values can be used as a guide to what constitutes a practical finish.

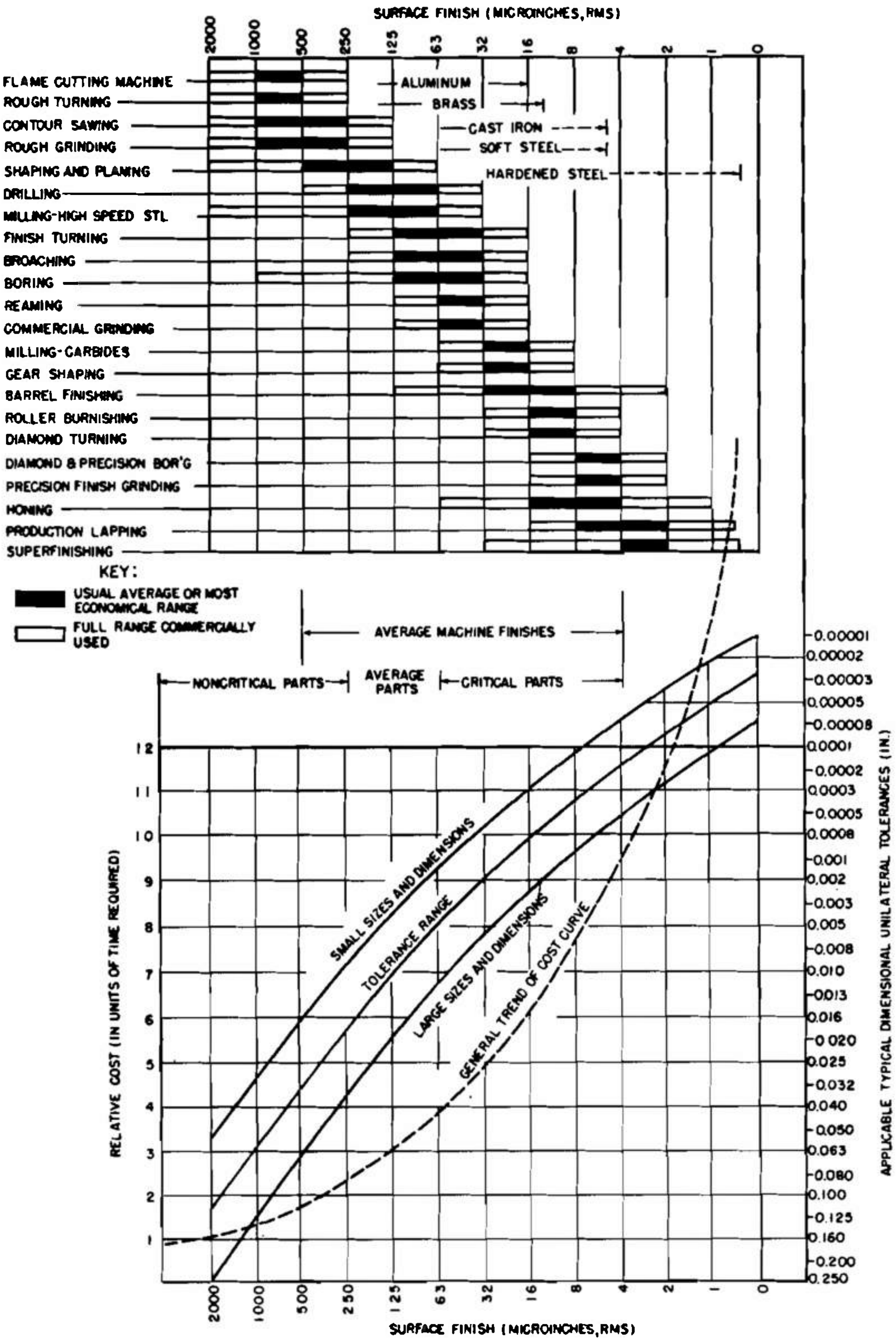


Figure 2-29. Practical Guide to Surface Finishes and Tolerances (Ref. 68)

TABLE 2-9
MAXIMUM SURFACE ROUGHNESS CHARACTERISTICS OF COMMON MACHINE PARTS (Ref. 61)

Part or Surface	Max. Roughness, μ in., rms
Bearings, antifriction, seats for	10
Bearings, journal (general)	32
Bearings, journal (precision)	16
Bearings, pressure lubricated	5
Bushings, rod (automotive)	20
Cam lobes (automotive)	16
Clearance surfaces (machined)	250
Crankpins	8
Cylinder bores (automotive)	13
Cylinder bores (O-ring or leather packing)	16
Datum surfaces (tolerances under 0.001 in.)	63
Datum surfaces (tolerances over 0.001 in.)	125
Fluid seals, surfaces of (sliding or rubbing)	5
Friction surfaces (brake drums, clutch plates, etc.)	16
Gaskets, surfaces for copper	32
Gaskets, surfaces for soft	125
Gear teeth (heavy loads)	16
Gear teeth (ordinary service over 10 D. P.)	63
Gear teeth (ordinary service under 10 D. P.)	32
Housing fits (no gasket or seal)	125
Mating surfaces (brackets, pads, faces, bases, etc.)	125
Pistons	16
Piston pins (automotive)	5
Piston rods (O-rings or leather packing)	16
Press fits, general (keys and keyways)	63
Push fits	32
Ratchets, teeth of	63
Rolling surfaces, general (cams and followers, etc.)	63
Rolling surfaces, precision heavy-duty	8
Rotating surfaces, general (pivot pins and holes, etc.)	32
Rotating surfaces, precision	16
Slide ways and gibs	32
Sliding surfaces of mating parts, general	32
Sliding surfaces of mating parts, precision	16
Threads, screw (chased)	250
Threads, screw (die cut or tapped)	125
Threads, screw (ground)	16
Threads, screw (milled)	63
Threads, screw (rolled)	8
Valve seats	8
Valve stems (automotive)	16
Worm gears, general	32
Worm gears, heavy loads	16

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PART TWO
LAND LOCOMOTION

CHAPTER 3
BASIC PHILOSOPHY OF LAND LOCOMOTION MECHANICS*

SECTION I DEFINITION OF THE PROBLEM

3-1 AN APPRAISAL OF OFF-ROAD CAPABILITIES

The subject of land transport as it applied to man's movement over the face of the earth is as old as the existence of man himself. He was endowed by nature with a basic means of locomotion, namely foot movement, which he later augmented with animals to carry him and his cargoes, and with carts and wagons to better support his burdens. Water barriers were crossed by wading or swimming and by poling or rowing various rafts and boats. The use of animals permitted heavier loads to be transported, conserved man's energies, and permitted increased speeds; although the increased speed was possible for only limited periods and required compensatory periods of rest. Thus, the use of animals in land transport resulted in certain improvements, but not without their price. Animals require feeding, rest, and care; and these requirements tend to offset the gain that the use of animals bring.

It is easy to misjudge the efficiency or true value of a transportation system if a broad viewpoint is not maintained. A horse walking, for example, can carry about three times the load that a man can carry; and he can walk with the load at about $1\frac{1}{2}$ times the man's walking speed (Ref. 1). If the man walks beside the horse, however, the horse's speed must be reduced to that of the man, making the net gain only that of the comparatively greater weight being carried by the horse. If the man rides the horse to take advantage of the horse's faster walking speed, the man's weight becomes part of the horse's load and reduces his effective payload to about 1.4 times the man's normal load capacity. When one considers that part of this payload capacity must be allocated to the horse's requirements, the net payload advantage is not very great.

Furthermore, the overall average speed of a march column is influenced not only by the speed of the marching elements but also by the total length of the column; for the movement isn't completed until all of the elements reach the finish line. Since a mounted column is considerably longer than when dismounted, it requires additional time to close into its destination, thus, reducing its overall average speed. Animal drawn vehicles in the column increase the load carrying potential but at a corresponding increase in column length. Furthermore, the interaction of the wheels and terrain during off-road operations creates a greater resistance for the draft animals to overcome, resulting in a further reduction in speed. Hence, the overall average speed of the mounted column is not significantly greater than that of the column on foot.

Similarly, the average overall speed of a modern mechanized column may not be significantly increased over a nonmechanized column. The presence of the vehicles increases the column length appreciably. In addition, a mechanized column has an almost insatiable appetite for fuel, lubricants, and repair items all of which must be carried along with the column or somehow supplied to it by supporting agencies. It is, therefore, not surprising that the marching rate of armies has not changed

*Written by Rudolph J. Zastera of the IIT Research Institute, Chicago, Ill.

significantly over a period of two thousand years.

The legions of ancient Rome were primarily foot troops with comparatively modest requirements, and who, therefore, traveled unencumbered by long supply trains. History tells us that these legions advanced at an average rate of about 12 miles per day. Napoleon's armies, almost 2,000 years later, amply equipped with animals and animal drawn vehicles for transporting men, equipment, and supplies, advanced at about the same rate. American forces during the Revolutionary and the Civil Wars, also had essentially this same rate of movement. The same applies to the rate of movement of the Mormon wagon trains, a nonmilitary force, migrating westward across the American continent early in the 19th century. General Patton's forces, much celebrated for their dramatic "dash" across France during World War II, actually had an overall average speed of 12 miles per day (Ref. 2). Obviously, certain elements of Patton's force could forge ahead at a much faster rate, but this rate could be sustained for only a limited distance determined by the availability of fuel and other needed services. Mormon outriders, too, could scout several days march ahead of the main column and return in a single day. The fact remains, however, that the overall average rate of movement is governed by the rate of the slowest element and is greatly influenced by the length of the column. Even though Patton's supply trains, too, were mechanized, they themselves had enormous fuel requirements and added to the length of the columns.

This must not be interpreted, however, to indicate that there has been no improvement since the days of the Roman foot soldier. The modern mechanized army moves into action with a preponderance of weapons, equipment, and supplies far greater than the ancients ever dreamed of and does so with a minimum expenditure of human physical energies. Thus, overall speed of movement, though important, is not the only gage of mobility nor of the value of mobility. Furthermore, the fact that assault echelons of modern mechanized or armored units can swiftly penetrate 50, 75, or 100 miles into the enemy positions to seize objectives is a definite improvement over an assault by foot troops or horse cavalry of a few hundred yards, even though they must wait after the assault for supplies to catch up to them.

There is one area, however, in which very little real progress has been made. This is the area of off-the-road, rough terrain, soft soil locomotion. Despite the fact that modern military vehicles are capable of top speeds in excess of 35 miles per hour, in off-the-road situations involving rough terrain or soft soils, their operating speeds drop to less than 8 miles per hour, with 5 miles per hour being a common average speed.

The chief factor responsible for this limited speed on rough terrain is the physical endurance of the operators and crews to the violent jostling to which they are subjected by this type of operation. Structural failure of vehicle components, particularly of suspension system components, is a secondary factor; but this is usually an indication of inadequate design. Physical tolerance limits of human operators cannot be altered. Recent experimental suspension concepts which emphasize maximum attenuation of ground shocks have permitted significant increases in vehicle speed over rough cross-country terrain. Thus a marked improvement in cross-country speed may not be far off.

In short soils, whether soft sand or silt, deep powdery snow, slippery wet clay, or deep mud, the reduced speed is due to a combination of factors. There is a reduction in the maximum tractive effort that can be developed by the wheels or tracks due to the low strength characteristics of the soil. For similar reasons, an appreciable sinking of the wheels or tracks takes place resulting in an increased rolling resistance and an increased resistance to steering control. And finally, if the sinkage is so great that the vehicle body or hull contacts (or sinks into) the ground, an additional "bulldozing" resistance is introduced along with a further reduction in tractive effort due to the flotation effect of the hull or body. The satisfactory solution of these problems requires an analytical investigation of soil-vehicle relationships and the matching of vehicle design to the environmental conditions anticipated for the vehicle.

3-2 LOCOMOTION IN NATURE

Since scientific attempts at the development of flying machines were preceded by studies of bird flight, and the development of efficient undersea vessels were preceded by studies of the swimming of fish, it is appropriate to begin a discussion of land locomotion with a brief consideration of the locomotion of animals. It is especially appropriate since the greatest difficulties in vehicular mobility are encountered in cross-country operations, and the locomotion of animals is exclusively of this type.

Animals in all parts of the world move about in their particular environment with a remarkable facility; although there is considerable degradation of this facility, in some cases, when the animal is taken out of its natural environment. Thus, the slender legs and sharp hooves of deer and elk endow these animals with an unbelievable agility in brushy forests and on steep mountain slopes; but they become completely immobilized in deep snows where the snowshoe rabbit and the wolf, on the other hand, have little trouble moving about. Bighorn sheep and mountain goats are equipped to scale precipitous rocky crags where even cats, equipped to climb trees, dare not follow; yet a more ridiculously helpless sight cannot be imagined than a goat high in a tree.

Some interesting observations can be made from considerations of animal mobility. For example: practically all animals have a fairly high degree of mobility over a wide range of environments; and, in most cases, can exceed the performance of modern military vehicles under identical conditions. Consider the elephant as a case in point. Though a denizen of tropical forests, his movements are not hampered significantly by either mud nor deep snow; and even though he sinks 3 or 4 feet deep in snow, he is still capable of plodding on at better than 5 miles per hour. Or consider a horse a hoofed animal primarily at home on firm ground. He can negotiate an appreciable depth of snow, dune sand, or mud although not in his best element, and can climb appreciably steep slopes as well. The performance of burros in mountains, deserts, and jungles is almost legendary; while the cross-country speed of a hunting cheetah is unequalled by any animal or machine.

It is interesting to note that in this natural system of animal mobility, the existence of a wheel or continuously rotating joint is nowhere to be found. The locomotion of all organisms in nature, with the exception of protozoa, is produced through a system of levers. By contrast, all man-made devices for off-the-road locomotion are based almost exclusively on wheels and wheel-like devices (tracks). Man himself is perhaps the most versatile of all land animals in his natural cross-country mobility capabilities. He is not superior on the basis of speed, endurance, or load capabilities, however, but makes use of his intelligence and ingenuity to overcome these deficiencies. In pursuit of these objectives, he may have taken an unsound course by basing his locomotion concepts upon the man created system of wheels rather than opon the natural system of levers. It is a point to ponder.

By making certain assumptions regarding the mechanics of translation, Bekker (Refs. 3 and 4) was able to determine the order of magnitude of the power requirements of various types of animal locomotion. He expressed the power requirements in terms of horsepower per ton of weight, thus permitting a comparison of the efficiencies of the various types of locomotion—including mechanical locomotion. Five basic types of natural locomotion were considered, namely, leaping, as a kangaroo or rabbit leaps; crawling, as a caterpillar crawls; sliding, as a snake slides; running as a horse runs; and walking, as a man walks. The following are the expressions of the power required for these types of locomotion.

For leaping:

$$(P)_{leap} = 0.91 \; \frac{v}{\sin \alpha \cos \alpha} \; \text{HP/ton} \quad (3\text{-}1)$$

where

v = velocity of locomotion, fps

α = angle that the trajectory makes with the horizon at the start of the leap, deg

0.91 = conversion factor to change units to HP/ton

For crawling:

$$(P)_{crawl} = 3.63\, fv \left(0.04 \frac{l_c}{l_d} + 1 \right) \text{HP/ton} \quad (3\text{-}2)$$

where

f = unit resistance of body weight to the forward motion of the stretching portions of the animal's body, lb/ton

v = velocity of translation of the animal, ft/sec

$\frac{l_c}{l_d}$ = ratio of the animal's (caterpillar) body length l_c, to length of that portion of the body l_d which is progressively lifted, stretched forward, and lowered

3.63 = conversion factor to change units to HP/ton

For sliding:

Here it was assumed that the animal (snake) was sliding with a sinusoidal motion of his body such that the tangent to the sinusoid at the intersection with the axis of motion formed a 20° angle with that axis. The unit power required is:

$$(P)_{sliding} = 3.63\ v\ \tan\mu\ \text{HP/ton} \qquad (3\text{-}3)$$

where

v = velocity of locomotion, ft/sec

μ = angle of friction between the snake's belly and the ground, deg

3.63 = conversion factor to change units to HP/ton

For running:

$$(P)_{running} = 0.91\ v\ \tan\alpha\ \text{HP/ton} \qquad (3\text{-}4)$$

Here the symbols have the same significance as in Equation 3-1.

For walking:

$$(P)_{walking} = 3.63\ v\ \frac{l_w}{p_w}\left[1-\sqrt{1-0.25\left(\frac{p_w}{l_w}\right)^2}\,\right] \text{HP/ton} \qquad (3\text{-}5)$$

where

v = velocity of locomotion, ft/sec

$\frac{l_w}{p_w}$ = ratio of length of the leg l_w, from the hip joint to the length of one step p_w

3.63 = conversion factor to change units to HP/ton

Suitable values were substituted for the various parameters in the foregoing equations and the power requirements for the five means of natural locomotion were compared to the power requirements of railroad cars, and of tracked and wheeled vehicles *on smooth, hard ground* at the same speed. Walking and running are the most efficient of the natural means of locomotion. Tracked vehicles, it was found, approached the efficiency of walking and running while wheeled vehicles exceeded it under the assumed conditions. *On soft ground,* however, the efficiency of wheeled vehicles deteriorates very rapidly while that of tracked vehicles deteriorates much less rapidly. Where the tracked vehicle required about one half again as much power (150%), the wheeled vehicle required about five times more (500%). The natural modes of locomotion required comparatively negligible power increases when changing from hard to soft ground operations.

Another consideration was vehicular speed. In addition to sinkage, the roughness of the ground is a serious factor affecting the speed of mechanical vehicles due to the violent pitching and shocks that it causes. Experience and theory both demonstrate that ground roughness can reduce the speed of both tracked and wheeled vehicles to almost the speed of walking. (The speed of wheeled vehicles is more sensitive to ground roughness than is that of tracked vehicles.) The speed of animal locomotion, however, is practically unhampered by the roughness of the terrain. A mountain goat can leap and run as swiftly in rough mountainous terrain as it can on smooth ground.

It may, therefore, be concluded that the animal kingdom, operating exclusively in an off-the-road environment, has developed its own methods of locomotion which do not make use of the wheel because its value is determined by the smoothness and hardness of the road—and roads are not produced by nature. The highest evolutionary stage of animal locomotion is running and walking, and the efficiency and versatility of these has not been surpassed in off-road locomotion by any man-made system of wheels or tracks. To be realistic, however, there are many reasons why we must assume that the wheel is here to stay as a universal means of locomotion; and it is, therefore, imperative to study the entire problem more systematically in order to improve its solution.

3-3 LOCOMOTION ON WHEELS

3-3.1 THE WHEEL IN ANCIENT TIMES

The true origin of the wheel is lost in antiquity; neither its inventors nor the circumstances leading to its invention are known. Like the cooking of food and agriculture, it was probably conceived independently by many persons in different parts of the world. The belief that the concept of a wheel developed from observations of round stones or tree trunks rolling downhill is not accepted by all authors. Primitive tribes exist today who, in their native state, do not know the wheel. Our American Indians were an example of this. Although a fairly intelligent and advanced people with ample opportunity to witness the rolling of round stones and tree trunks, prior to the coming of white men, they used travois to transport their cargoes without incorporating a wheel. The oldest records of wheeled carriages date to 3200 B.C. They are pictures of vehicles built to transport warlords and chiefs. And so, the wheel as applied to the locomotion of military vehicles has a long history, indeed.

3-3.2 EARLY OBSERVATIONS OF SOIL-VEHICLE RELATIONSHIPS

The very first wheeled vehicles were probably propelled by manpower; although there is no record of this.

With the domestication of the horse and ox, and the development of harness, came the first animal drawn vehicle on which man could transport many times the weight that the animal could normally carry. This was perhaps the first major revolution in land transportation; and, as far as off-the-road transportation is concerned, was the turning point which led to the vehicles of today and to our elaborate system of highways.

In those early days of man-propelled and animal-propelled vehicles, the man and the animal were the prime movers while the carts and wagons constituted towed vehicles. Although the feet and legs of these prime movers continued to exhibit the superior off-road performance characteristics developed into them through countless eons of evolution, the cart wheels, which were only recently invented by man, did not perform nearly as well. The softer the soil became, the greater their sinkage and the higher their rolling resistance. Rocks, logs, and ground irregularities, which the feet could easily step over or around, became obstacles in the paths of the wheels. This, then, was man's first exposure to the incompatibility of the wheel to the off-road environment. That man recognized the problem is evidenced by early developments of the wheel and by the development of roads.

Early wheels were narrow and large in diameter. This configuration is conducive to decreased sinkage, less rolling resistance, better obstacle-crossing capabilities, and, in self-propelled vehicles, to the development of greater tractive effort. These wheel developments are evidences of man's early awareness of the interrelationship between the soil, the surface geometry, and the vehicle.

When man removed the first rock or fallen log from the path of his first wheeled vehicle, he was unwittingly recognizing the incompatibility of his wheeled invention to the environment in which it was to operate. By removing the obstacle (which, incidentally, is not an obstacle to natural animal locomotion) he was modifying the operating environment and admitting the inferiority of his invention. Furthermore, by bridging mud holes with brush, logs, or gravel to reduce the sinkage and rolling resistance of his wheels he was further modifying the operating environment. Thus began man's roadbuilding activities which reached a very high state of advancement during the days of the ancient Roman Empire.

When mechanical power was added to the wheeled vehicle, another phase began in the evolution of vehicles away from the natural environment. Heretofore propulsion was through the tractive effort developed by the animal prime movers, and the wheels were required to provide a moving support for the load. Now, with the animal prime movers gone, the wheels were also required to provide traction. It became quickly apparent that in this respect, too, the wheels were inferior to the animal's hoof or the pedestrian's foot, particularly in soft soils. Add to this the requirement for greater tractive effort to overcome the rolling resistance of the wheels due to sinkage, and the wheel is sorely inferior.

On hard ground, the tractive capacity of wheels is greatly improved and the rolling resistance is minimized. Furthermore, the increased speed, potentially available through the addition of mechanical power to the vehicle, cannot be fully realized on rough ground due to the harshness of the ride. Here, too, the improved environment of the paved road permits acceptable performance from the wheels.

It is, therefore, not surprising that man should engage in extensive road building activities to create suitable operating environments for his vehicles, and to pursue vehicle developments based upon this operating environment. The proficiency of the modern highway engineers in developing safe, smooth roads offering good traction and gentle slopes has had a great influence upon the development of wheeled vehicles. Unfortunately, the requirements of military vehicles are for maximum off-road capabilities in a wide range of geographical terrain and under all climatic conditions. Thus, the designers of modern automotive vehicles for civilian use and the designers of modern automotive vehicles for military use have diverging interests. As civilian automotive vehicles become more and more dependent upon good roads, their suitability for military operations decreases.

It is necessary, therefore, for a thorough scientific study to be made of soil-vehicle relationships to bring about a clear understanding of the mechanics of the resistances encountered and of rational means of overcoming them. Here lies the only hope for a practical and economical solution to the age-old problem of cross-country locomotion. A great deal has been done in this respect in the field of aerodynamics and hydrodynamics but relatively little in terramechanics. Perhaps this is largely due to the fact that the operating environments of the first two disciplines do not lend themselves to limited modification as does the last discipline. The workers in the first two fields could not avoid the problems that their environments introduced. They were forced to understand their working environment and make their designs compatible to it. With this approach they found success.

SECTION II VEHICLE MORPHOLOGY

Observations of evolutionary changes in the animal world show that evolutionary processes tend to eliminate the undesirable and to improve the desirable attributes of each species according to the requirements of the environments in which each live. The physical forms of the species that are capable of locomotion tend to evolve into a form that offers the least resistance to motion. This is also true in the world of technology as is illustrated by a comparison of the forms of ancient wooden carts with those of streamlined racing cars, aircraft, and submarines. Any attempts at rationally adapting self-propelled land vehicles to their medium of operation should be preceded by a survey of present vehicle concepts to point out the evolutionary trends and the direction future developments should take. Bekker accomplished such a study (Ref. 5) and developed three morphological indexes; namely, the form index, the specific weight index, and the size and form index. These are summarized in the paragraphs which follow.

3-4 FORM INDEX

The general forms of vehicles were described by Bekker in terms of three ratios; namely, the height aspect, the width aspect, and the frontal aspect. The *height aspect* is the ratio of the vehicle height to its length h/l; the *width aspect* is the ratio of the width to the length w/l; and the *frontal aspect* is the ratio of the frontal area to the vehicle length wh/l. The length l is that dimension which is oriented along the direction of motion.

A survey of a large number of sea, air, and land vehicles which compared their height and width aspects, h/l vs w/l, showed that the majority of vehicles tend toward a one-to-one relationship between height and width. Military tanks and tracked tractors displayed a tendency toward wider shapes, while trucks and passenger cars displayed the opposite tendencies. In total, however, the study indicated a prevailing trend of the form of land vehicles toward a short, wide prism, sometimes slightly elongated in the direction of motion. This trend is opposite to that displayed by sea and air vehicles which favor a typically streamlined shape with low h/l and w/l ratios.

The need for streamlining is usually associated with speed, and the need for the streamlining of slow-moving land vehicles seems unnecessary. Subsequent chapters of this handbook, however, show that long, narrow shapes are more rational in land locomotion than are short, wide ones. This supports the long, narrow form of the ski as highly efficient. The general laws governing the morphological evolution of moving bodies remain the same whether they apply to fish, submarines, aircraft, or land vehicles. The lack of compliance with this law should be of concern to those interested in improving the mobility of land vehicles.

The softness of the ground is not the only consideration in cross-country mobility. Other factors, such as the surface contour of the ground, the presence of large trees or boulders, narrow passes and mountain trails, are often sufficient obstacles to seriously slow down or stop all locomotion. In these situations, the *frontal aspect*, wh/l, plays a significant role. It is obvious that the types of obstacles just mentioned can be tolerated more readily by vehicles of smaller frontal aspect than by those exhibiting large height or width aspects, h/l and w/l. Thus, it seems that streamlining as expressed by low values of the form indexes, h/l, w/l, and wh/l, is as essential to effective cross-country-type land vehicles as it is to high speed vehicles moving against aerodynamic or hydrodynamic resistances.

Agricultural tractors are the greatest violators of these form principles. Since they perform their function of plowing and harvesting on open plains at slow speeds, these form indexes have minor importance, and agricultural tractors are traditionally short, stubby vehicles. When performing their functions in special operations such as in jungles, forests, mountains, or narrow passes, their stubby forms are not practical. This is especially true with regard to heavy tractors since they are the most likely candidates for these types of operations; yet their height, width, and frontal aspects are the highest.

3-5 SPECIFIC WEIGHT

The specific weight of a vehicle is the ratio of its total weight to its total volume, and is expressed as pounds per cubic foot. It is of interest when buoyancy characteristics are under consideration, as in the design of amphibious vehicles, and when considering the shipping requirements of land vehicles. Since the function of vehicles is to transport some form of cargo, the gross vehicle weights should be used in calculating specific weights, particularly in cases dealing with amphibian wheel design.

When determining the specific weight of a vehicle with regard to shipping requirements, the volume that is considered is the volume of a so-called "shipping prism." This is a representative rectangular prism having length, height, and width equal to the corresponding maximum dimensions of the vehicle. The maximum density vehicles are tractors, with "shipping" specific weights ranging from approximately 12 to 40 lb per cu ft, followed closely by tanks, with approximate specific weights from 18 to 30 lb per cu ft. Trucks come next, having specific weights from about 8 to 20 lb per cu ft. Passenger cars have the lowest specific weights, 5 to 8 lb per cu ft.

It is interesting to make several comparisons here. The specific weight capacities of cargo aircraft, i.e., the total cargo weight capacity divided by the total cargo volume capacity, are from about 4 to 10 lb per cu ft depending upon the aircraft and the distance to be flown. This indicates that passenger cars and the lighter weight trucks are economically compatible cargo for aircraft. Similarly, the specific weight capacities of ocean-going vessels fall between 35 and 45 lb per cu ft depending upon the type of vessel and the availability of deck space for cargo. This makes ships economically suitable for the shipping of tanks and tractors.

The true volumes of vehicles are approximately 35 percent of the volumes of their respective shipping prisms, which makes their actual specific weights about three times their shipping specific weights. Thus, it is seen that the true specific weights of passenger cars, trucks, and light tanks do not exceed the specific weight of water (64 lb per cu ft). This indicates that it should be comparatively simple to make vehicles of these categories amphibious, but medium and heavy tanks and tractors require assistance in the form of enormous floats.

3-6 SIZE AND FORM

In a discussion of the morphology of things, it is apropos to examine the relationship between the sizes and general forms of the objects and to note the evolutionary changes that have occurred. Thus, it is seen that ocean vessels, for example, have undergone considerable change from the days of the ancient Phoenicians to modern times. As the capabilities of the vessels increased, not only did their size increase but their configuration changed radically in adaptation to the medium in which the vessels operate. Similar evolutionary changes in the form of aircraft are evident paralleling their spectacular progress. An examination of land vehicles, however, does not reveal a similar history of change. It is not surprising, therefore, to find a disturbing lack of progress in cross-country performance.

This lack of appreciable change in the form of land vehicles is partially the result of limitations imposed by the widths of roads. These widths were established in ancient times when animal drawn vehicles were the only means of locomotion. Since that time, the only changes in road width resulted from considerations of safety made necessary by increased vehicle speeds. Thus, the modern concept of roads, highways, bridges, and even railroads is built up from the basic dimensions of the ancient Roman four-horse chariot.

It is admittedly too late to change for obvious economic reasons; and a change is probably not necessary since the artificial environment of the highway and the man-made wheel are reasonably compatible. But the rationality of allowing the wheel spacing of a Roman chariot to influence a modern military vehicle, whose design is to emphasize cross-country mobility, should be carefully examined.

The width of highway vehicles has remained practically constant at about 80 inches for passenger vehicles of approximately $1\frac{1}{2}$ to $2\frac{1}{2}$ tons gross weight and at about 90 inches for trucks of approximately 3 to 9 tons gross weight. The lengths of highway vehicles are approximately 2 to 3 times their width. These dimensional relationships also apply, generally, to off-road vehicles with the exception of some agricultural tractors and special industrial off-road vehicles which violate the limitations of the road width rule.

The width of tracked vehicles shows some relationship to the gross vehicle weight and has been expressed by Bekker (Ref. 5) with the following approximate equations:

$$w_{tank} = 90 + 1.0\ W \tag{3-6}$$

and

$$w_{tractor} = 49 + 3.75\ W \tag{3-7}$$

where w and W represent the vehicle width in inches and gross weight in tons, respectively. Similarly, the length-to-weight relationship was found to approximately agree with the following expressions:

$$l_{tank} = 170 + 1.9\ W \tag{3-8}$$

and

$$l_{tractor} = 85 + 6.5\ W \tag{3-9}$$

where l is the vehicle length in inches. From a combination of Equations 3-6 to 3-9, the length-to-width relationship l/w of tracked vehicles is found to vary from approximately 1.7, for tractors; to 1.9, for tanks. Thus, while the width of highway vehicles is relatively constant regardless of gross vehicle weight, tracked vehicles follow a rule of relatively constant length-width ratio.

The reason for maintaining a length-to-width ratio for tracked vehicles between 1.7 and 1.9 relates to the steering requirements of a tracked vehicle. Since about 1904—when Messrs. D. Roberts and Richard Hornsby applied an automobile differential and two brakes, one to each track driving shaft, to a tracked vehicle—tracked vehicles have been steered by the application of unequal forces to each track to affect a slewing couple M of sufficient magnitude to cause the vehicle to turn. If F represents the track force causing the slewing couple and w is the tread width of the vehicle, the turning moment is

$$M = wF \tag{3-10}$$

It is later shown that the resisting moment to turning is directly proportional to the track length l, the gross weight of the vehicle W, the coefficient of friction to lateral movement μ_l, and to a certain function ζ, of the load distribution on the track, or

$$M = \mu_l W \zeta l \tag{3-11}$$

The driving force F which is required to develop the steering moment is limited by the load on each track $\frac{W}{2}$ and by the coefficient of adhesion μ_a that can be developed between the track and the soil, or

$$F = \frac{W}{2}\mu_a \tag{3-12}$$

Combining the last three equations, results in the following length-to-width relationship:

$$\frac{l}{w} = \frac{1}{2\zeta}\left(\frac{\mu_a}{\mu_l}\right) \tag{3-13}$$

It has been shown (Ref. 6) that the factors μ_a, μ_l, and ζ are practically invariable in limiting conditions for tracked vehicles and, therefore, the steerability of tracked vehicles is determined by a nearly constant relationship of l/w. Values of l/w higher than 1.9 result in a vehicle that is almost impossible to steer (par. 5-4-4).

In conclusion, it must be pointed out that the length-to-width ratio of tracked vehicles is restricted, and hence the evolutionary development of vehicular form is restricted, by peculiarities of the steering concept that has been adopted for vehicle control and not by the physical laws and geometry of the medium in which the vehicles operate. This argument has been advanced by some to explain the relative lack of change in the form of cross-country vehicles.

3-7 VEHICLE FORM AND ITS ENVIRONMENT

The rather arbitrary manner in which vehicle form originated was discussed in preceding paragraphs. Since highway vehicles were placed upon a prepared operating surface, they were freed from the necessity of adapting to the operational requirements of their environment. Thus, vehicle lengths increase with increasing weight until a length of approximately 22 feet, at a weight of 5 or 6 tons, is reached. Thereafter, the length becomes more or less stabilized since such factors as maneuverability and the radii of curvature of roads now become decisive in determining maximum vehicle length. Safety, traffic control, production problems, and cost are other factors which influence the morphology of highway vehicles. Thus, the basic form of highway vehicles is the result of the dictates of public laws, of highway and bridge dimensions, and of the functional applications of the vehicles.

When establishing the form of a vehicle intended for cross-country operations, consideration should be given to the type of terrain in which the vehicle will operate. In extremely rocky, mountainous terrain, where the vehicle is forced to travel narrow tortuous paths, extreme length is an obvious disadvantage due to the inability of a long vehicle to execute abrupt changes of direction. This disadvantage can be ameliorated by designing the vehicle in the form of a tractor-trailer assembly of two, or more, relatively short units. In this respect, a statistical analysis of the geometrical characteristics of the terrain of operations is a valuable basis for the determination of vehicle form.

Similar considerations apply when the vehicle is to operate in heavily forested areas. Here the vehicle width comes up for consideration and must be compared to the spacing between trees—particularly the spacing between trees that are too large for the vehicle to knock down. However, the spacing of trees that the vehicle is capable of knocking down must not be ignored, either. Where a vehicle may have sufficient power and traction to knock down one or two trees of a given size simultaneously, three trees may immobilize it. Thus, the narrower the vehicle, the fewer will be the trees that the vehicle must engage simultanously.

The roughness of the ground and the presence of vertical obstacles are additional considerations that affect the length, width, and general configuration of off-the-road vehicles. These are also discussed in Chapter 2. The obvious effects of terrain irregularities are bouncing, pitching, and rolling of the vehicle. Since these effects increase in

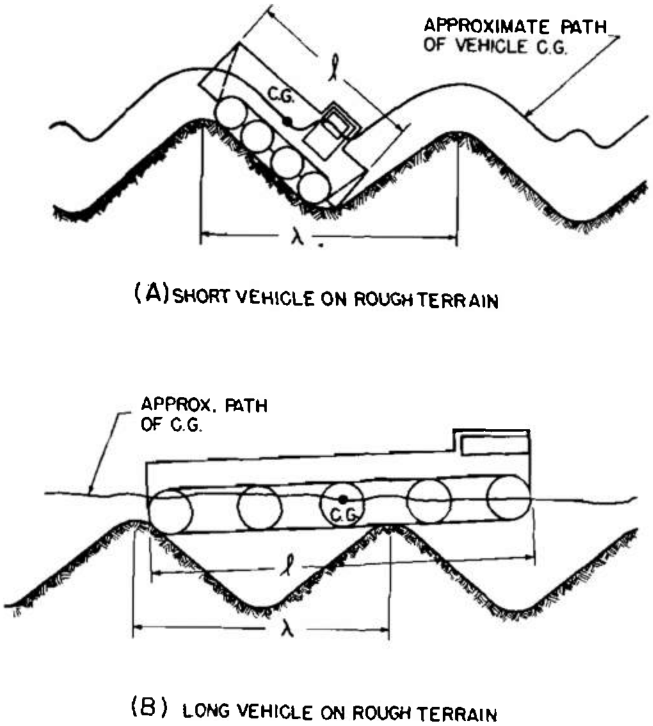

Figure 3-1. Effect of Vehicle Length on Ride Characteristics

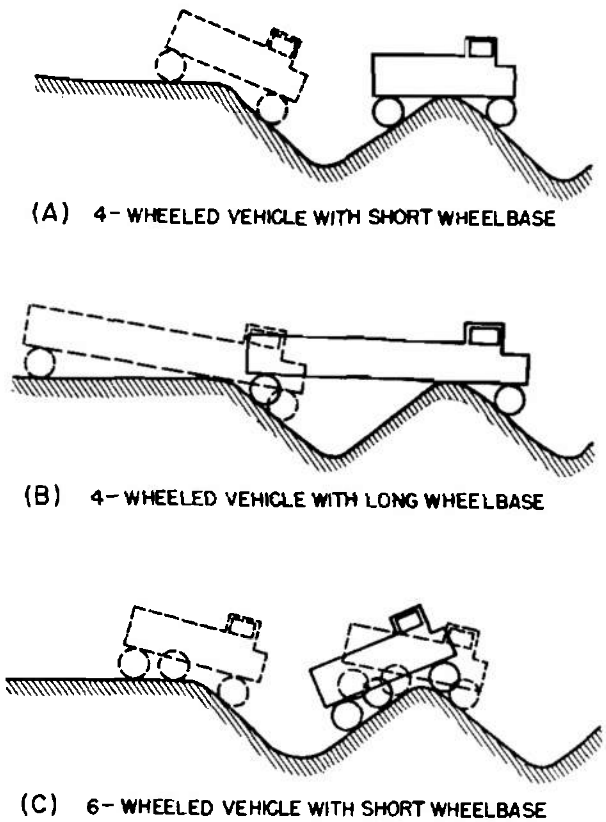

Figure 3-2. Effect of Ground Clearance and Wheelbase on Rough Ground Performance

violence with vehicle speed, they indirectly limit the vehicle speed to a level where the resulting shocks and vibrations are within the physical tolerance limits of the operator and crew.

Ground roughness, when sufficiently severe, can immobilize the vehicle. This occurs when the wavelength of the "ground wave" λ is longer than the overall length l of the vehicle, and if any structural element of the vehicle at a height within the amplitude of the "ground wave" protrudes forward of the ground-contacting elements (wheels or tracks). This is illustrated in Figure 3-1(A), which shows a tracked vehicle whose length l is considerably shorter than the length λ of the "ground wave." The vehicle shown will be immobilized when the front overhang digs into the rear slope of the next wave, unless the front of the vehicle is modified as shown by the dotted line on the figure. Similarly, the rear overhang requires modification to prevent it from digging into the bottom of the forward slope of the "ground wave" as the vehicle attempts to climb the next rise. With the vehicle ends modified as indicated, the vehicle will be able to proceed over the rough terrain. Its center of gravity will travel a path approximately as shown in the figure; obviously resulting in a rough ride necessitating a very low speed.

Figure 3-1(B) shows a tracked vehicle whose length is greater than the length of the "ground wave." This enables the vehicle to bridge considerable portions of the terrain with the result of a much smoother ride, as is shown by the approximate path traveled by the vehicle's center of gravity. If the vehicle is flexible in the horizontal plane (as a vehicle comprised of several short, coupled units), it will tend to conform to the ground contour, much as the short vehicle of Figure 3-1(A) does, and the resulting ride will again be rough.

Figure 3-2 illustrates the effect of ground clearance on performance over rough ground. The three vehicles shown have the same ground clearance and are shown on the same terrain. The four-wheeled vehicle shown in Figure 3-2(A) enters the first depression successfully, succeeds in crossing the bottom, but becomes immobilized when the front wheels pass over the next crest due to insufficient ground clearance. More ground clearance would obviously permit the vehicle to move on.

The amount of ground clearance needed by a particular vehicle is a function of the distance between successive wheel centers and the amplitude of the "ground wave." Figure 3-2(B) shows a four-wheeled vehicle with a long wheelbase attempting to cross rough terrain. It becomes immobilized soon after the front wheels enter the depression. Should it be pushed ahead, somehow, it will again become immobilized when astride the next crest, thus demonstrating that a vehicle with a long wheelbase is generally inferior when crossing rough terrain. It requires more ground clearance than a short wheelbase vehicle in the same terrain.

Figure 3-2(C) shows the effect of adding an additional axle between the two axles of the vehicle shown in Figure 3-2(A). Since this makes the wheel spacing considerably closer together, the vehicle will now cross the terrain with relative ease. Pitching will still be considerable, however, due to the short overall vehicle length making for a rough, slow ride. Additional wheels can be added to the vehicle of Figure 3-2(B) to enable it to cross the assumed terrain without increasing its ground clearance. If thus modified, this vehicle would give the smoothest ride and would permit the highest speed.

Streamlining a vehicle is usually associated with aerodynamic drag resulting from high speeds. It should be an important consideration, however, in the design of cross-country vehicles with a high degree of mud or snow mobility. In this case, the components beneath the vehicle's belly are pushed through the mud or snow, resulting in an appreciable drag on the vehicle. In fact, it is this bulldozing effect of standard wheeled vehicles that is the main cause for their bogging down in deep mud or snow. Thus, while streamlining of the body is important in reducing drag at high speeds, streamlining of the undercarriage is important at all speeds when mud or snow mobility is a requirement.

SECTION III DEFINITION OF MECHANICAL MOBILITY

The process of designing begins with a statement of the requirement; included with this statement are specifications of such factors as size, weight, appearance, performance, etc., as may be pertinent to the requirement. These specifications direct the creative efforts of the designer toward the desired goal. When a prototype of the new design is completed, it is evaluated against these same specifications to determine whether the requirement was satisfactorily met. Obviously, the more detailed and precisely stated the specifications can be, the more useful they become to both the designer and to the evaluators of the completed prototype.

The preparation of performance specifications for cross-country vehicles is a complex task. An ideal cross-country vehicle for military operations is one that can perform effectively in deep mud, snow, and over rough terrain, that can surmount significant vertical obstacles, is amphibious, air-transportable, and that can operate on roads as well. Its performance specifications must, therefore, include all of these conditions. Soft-ground performance embraces an almost infinite number of soil conditions; rough terrain involves the description of ground profiles; and obstacle performance requires the dimensions of walls and ditches as well as descriptions of the attendant soil conditions. To this collection of performance areas to be covered are added requirements which deal with the vehicle's functional performance, i.e., to what extent it must fulfill its primary purpose. In the case of a simple cargo carrier, an example of functional requirements would be specifications as to the weight and bulk of the maximum load the carrier is required to carry; in the case of a mobile weapon system, specifications as to the required effectiveness of the vehicle as a weapon platform; and in the case of an armored carrier, specifications of the degree of protection required for passengers and crew.

It is evident that the overall performance of a cross-country vehicle is comprised of performance characteristics of two general categories. The first category includes characteristics that relate to the vehicle's capacity or competence to move, while the second category deals with factors of functional performance. While the latter category is of importance in the evaluation of total vehicle performance, it is the first category which is concerned with mobility.

Mobility, then, is a term representing the competence of a vehicle to move over any given terrain. The manner of expressing this competence, in order to be meaningful and useful in vehicle design, must be based upon quantitative expressions in units of length, mass, and time. This has not been done in the past. Instead of this, the practice has been to specify mobility in descriptive phrases using vague, qualitative generalities; such as "be capable of all season, off-road operation, over a large variety of difficult terrain including mud, sand, marshes, tropical swamps, muskeg, tundra, and be capable of crossing vertical obstacles of a maximum height consistent with aspects of the designs." When scrutinized for quantitative data, none can be found. "To be capable of" merely indicates a "go" proposition with no indication of how fast or with what efficiency the vehicle should "go." "Off-road operation" indicates nothing as to the degree of terrain roughness, the severity of slopes, nor the profile of ground depressions. Terms such as "mud, sand, marshes, swamps, muskeg, tundra," are descriptive but have no quantitative interpretation that can be used in calculations of thrust, sinkage, and resistance to motion.

In order to describe mobility in reasonable terms, it is necessary to examine the elements which are involved and determine their quantitative characteristics. Thus, we find that off-road mobility is comprised of the following factors:

(a) Soft-ground performance
(b) Rough-ground performance
(c) Obstacle performance
(d) Slope performance
(e) Water performance
(f) Agility
(g) Air transportability

The first of these factors, soft-ground performance, has been the most difficult to specify in the past. This was chiefly due to a general lack of knowledge of the physical interrelationships between vehicles, particularly the ground-contacting elements of vehicles, and the engineering properties of various soils. Intertwined with this lack of knowledge was the lack of a satisfactory system for describing soil types and soil conditions that conveyed engineering characteristics of soils in quantitative terms useful to the vehicle designer. To date, however, several approaches exist to the study of vehicle mobility. Some attempt to arrange various vehicle characteristics, judged to be pertinent to the problem of mobility, into a logical sequence resulted in a numerical value designated as an index of mobility (Chapter 9, par. 9-43.3 and Ref. 7). Others relate measured soil properties and measured vehicle performance to various vehicle factors which determine vehicle performance; such as size of engine, type of transmission, wheel or track loadings, dimensions and aggressiveness of ground-contacting elements, ground clearance, etc., and arrive at a numerical mobility index (Refs. 8-10). Still other approaches define pertinent soil properties—after investigating the relationships that exist between soils and wheels, or tracks—and develop equations which include both soil and vehicle parameters (Refs. 11-13). Scale model studies and dimensional analysis techniques are also used in soils, sand, and snow (Refs. 14-15). These various methods of describing the soil-vehicle relationships are discussed in subsequent chapters of this handbook.

The second mobility factor, rough ground performance, is quite amenable to being specified. A complete specification must include a minimum acceptable speed over some standard, reproducible ground profile. Ground profiles can also be described geometrically and the vehicle performance can be evaluated analytically (see Section IX of Chapter 8).

Specifications of obstacle performance require a description of the obstacle geometry, such as a vertical wall of stated dimensions, or a ditch or bump of particular configuration, dimensions, and slopes.

Slope performance has been stated quantitatively for a long time, however, the nature of the slope and condition of its surface have usually not been specified. The only purpose served by a slope specification that neglects to define the nature of the slope is to establish power levels and braking capabilities, and in actuality, these could be established by drawbar-pull tests. The real intent of the slope specification is to define the degree of slope in nature that the vehicle is required to negotiate; therefore, in order for this specification to be meaningful, the nature of the slope surface must be included.

Water performance is usually specified as the ability to ford depths of a given amount, or as the ability to float with a specified minimum freeboard and the ability to attain a stated minimum water speed. To be truly useful as an amphibious vehicle, however, it must have a fairly high capability of entering and leaving the water unassisted. The margins of many bodies of water are often soft and marshy, on the one hand, or steep on the other. When entering or leaving the water, there is a critical transition zone during which buoyant forces acting on the vehicle reduce the tractive effort of the ground-contacting elements. If sufficient traction cannot be maintained to overcome the vehicle resistances, the vehicle will become immobilized. Thus, to completely specify the amphibious requirements of a vehicle, the maximum slopes and nature of the banks must also be specified.

Agility is a term used to define a quickness of maneuverability. Maneuverability is fairly simple to specify by giving minimum turning radii, requiring tbe ability to follow a defined circuitous course, and stating the maximum acceptable reduction of forward speed as a direct result of negotiating a turn. (This latter is particularly apropos to tracked vehicles.) Agility can be specified by associating certain minimum speeds with the stated elements of maneuverability.

The last factor, air transportability, although an important element of overall mobility, is not an element of land mobility. Nevertheless, even this factor can be quantitatively specified. Necessary parameters are maximum allowable weight, size restrictions, and sometimes maximum allowable ground (floor) loading. Current practice in this respect is to specify that the vehicle must be transportable by a particular class of fixed wing or helicopter aircraft and thus shift the responsibility of determining maximum dimensions and weights to the designer. A cautionary word to the designer, therefore, is in order here. The maximum dimensions are often determined, not by tbe dimensions of the cargo space, but by the dimensions of the hatch openings and their relationship to the configuration of the loading ramp and the interior of the aircraft. Many vehicles are in existence that are of compatible size to fit inside particular aircraft, but they cannot be loaded.

In summary, cross-country mobility is defined as the competence of a vehicle to move over a given terrain, as measured by its best average speed over a prescribed route. Included in this specification are quantitative expressions of the vehicle's competence to cope with soft ground, rough ground, obstacles, water barriers, and its agility and compatibility to airborne operations. Mobility results from a combination of the driving forces and the forces which oppose the movement. Since both of these forces are dependent upon the environment, which itself is variable in nature, the resultant mobility will vary for different conditions. Thus, a mobility specification must include a specification of the environmental conditions in which the mobility is determined. It also, logically, follows that the environment specified should be representative of the environment in which the vehicle will operate. This admonition might seem ridiculously obvious were it not for the long established practice of evaluating the drawbar pull of farm tractors on concrete test courses.

It is thus evident that mobility is a complex factor which does not lend itself very readily to representation by a simple single index number. Furthermore, a simple single index of mobility would, in all probability, be of little use to a designer.

Bekker (Ref. 16) proposed a system for evaluating mobility by tabulating all conceivable, relevant, measurable performance characteristics—such as thrust, speed, payload, height of negotiable obstacle, flotation, etc.—of vehicles in various types of terrain representative of their actual operational environments. The resulting matrix of performance values would be a parametric representation of the mobility of the vehicles in the selected terrain conditions. This could be optimized, by the application of operations research techniques, into a single overall solution.

Other significant performance factors are: the payload delivery rate vp, represented by the product of the vehicle payload p, and its velocity v; and the payload delivery rate per unit quantity of fuel consumed vp/f, represented by the ratio of the payload delivery rate to the rate of fuel consumption f. This latter ratio should generally be maximized. These factors, too, can be included in the mobility matrix since the final selection of the most desirable vehicle for a particular type of operation and terrain will depend upon delivery rate, fuel economy, or speed of operations, whichever has the highest priority. Compromises can easily be made when mobility matrices are used. The frequency of obstacle occurrences, the effect of route selection, or the probability of the occurrence of terrain changes due to climatic or weather changes —all of which enter into a broader definition of vehicle mobility—all lend themselves to evaluation by means of the mobility matrix technique.

The foregoing illustrates the many criteria that may be selected for an overall definition of mobility. It should be remembered, however, that mobility criteria are meaningful only when expressed in measurable terms and only when related to a rationally described terrain environment.

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CHAPTER 4

EQUATIONS OF LAND LOCOMOTION MECHANICS*

SECTION I REQUIREMENTS FOR MEANS OF IDENTIFYING TERRAIN PROPERTIES

4-1 NEED FOR QUANTITATIVE IDENTIFICATION OF PHYSICAL PROPERTIES OF SOILS

The rational design of a vehicle for off-road locomotion necessitates that consideration be given to those properties of soil which affect trafficability. A soil must have sufficient bearing capacity to prevent the vehicle from sinking too deeply and sufficient traction capacity to provide the necessary forward thrust of the vehicle's wheels or tracks for propulsion. However, very often the off-road soil conditions are such that movement without sizable sinkage and slip is impossible.

Immobilization of a vehicle is usually caused by concurrent failure in bearing and traction. Traction failure can occur without appreciable sinkage, as when a wheeled vehicle spins its wheels on a firm soil which has a soft, slick overlay; but sinkage failure does not usually occur without being accompanied by traction failure since traction requirements increase with sinkage. The multiple passage of vehicles is also a frequent requirement. Many soils, which the first few vehicles can traverse, will weaken due to soil disturbance and not support subsequent vehicles of the same type.

Bearing capacity and traction capacity are both functions of soil shear strength. The shear strength of soil is achieved from two basic properties, cohesion and internal friction. Cohesion is a measure of the bonds between soil particles which are developed from complex phenomena and can be understood only by study of the surface physics and chemistry of the small mineral particles present in soil. Internal friction is that portion of soil shear strength developed by the interlocking of soil grains and the frictional resistance to sliding between the grains. The shearing strength developed by internal friction is proportional to the stress level; i.e., the greater the normal pressure applied to a surface of sliding in the soil, the higher the shearing resistance which may be mobilized along that surface.

Shear strength of soil is often considered independently from the associated deformations, as in stability problems. However, in problems dealing with the performance of vehicles on soil, the stress-strain characteristics of the soil, including deformations associated with failure, are important since they relate traction with slip, and sinkage with load-carrying capacity and rolling resistance. The soil failure is usually progressive and locally concentrated.

* Written by Dr. Ernest T. Selig of the IIT Research Institute, Chicago, Ill.

4-2 NEED FOR QUANTITATIVE IDENTIFICATION OF TERRAIN GEOMETRY

The remarkable speed range of motor vehicles on modern highways is due partly to the power available, the suspension system, and the straightness of the highway; but mainly, it is due to the smoothness of the surface upon which the vehicles operate. The importance of the smoothness of the surface is realized immediately when these vehicles are required to operate off-road. The vehicle then vibrates to such an extent that its speed must be radically reduced. Since the response of the vehicle to ground roughness is related to the terrain profile as well as the vehicle's dynamic characteristics, no fully rational vehicle design is possible if the geometrical properties of the ground are not considered. These properties can be described either statistically or by equations representing typical profiles.

SECTION II PHYSICAL SOIL VALUE SYSTEM

4-3 PROPERTIES OF SOILS

4-3.1 PURE COHESIVE SOILS (Ref. 1)

A pure cohesive soil is defined as one which, when unconfined, has considerable strength when air-dried and that has significant cohesion when submerged. Its shear strength is derived entirely from cohesion, i.e., strength which does not depend upon the applied loading. Such soil would be classified as an ideal clay, which is a fine-grained soil or the fine-grained portion of soil that can be made to exhibit plasticity (putty-like properties) within a range of water contents and that exhibits considerable strength when air-dry. The term "clay" has been used to designate particles in soil finer than 0.002 mm, but this usage is not preferred because the size alone does not guarantee that the important properties will be exhibited.

Because of its plasticity, the soil can be deformed beyond the point of recovery without cracking or appreciably changing volume. It will exhibit shrinkage upon drying, and swelling in the presence of water. Because of the small size of the particles (relative to other soil ingredients), the cohesive soils have low permeability to pore fluids. Hence, consolidation will not occur immediately upon application of load. For this reason, consolidation, which is very important in foundation engineering, will not be an important factor in cohesive soils subjected to vehicle traffic loads.

The shear strength of clay varies considerably depending on the amount of moisture present and, for *in-situ* soils, will decrease as moisture content increases. The state of the soil is generally spoken of in terms of its consistency as being soft, medium, stiff, or hard. Shear strength will also change with unaltered moisture content if the soil is subjected to a kneading action due to structural rearrangement. This effect is measured in terms of what is known as the sensitivity of the clay. This is indicated by the term degree of sensitivity S_t defined as

$$S_t = \frac{\text{cohesive strength, undisturbed}}{\text{cohesive strength, remolded}}$$

For most clays, S_t ranges between 2 and 4; for sensitive clays, it is between 4 and 8. Soils for which S_t is greater than 8 are termed extra-sensitive. If a kneaded soil is allowed to stand, it may regain its cohesive strength. This phenomenon is known as thixotropy.

A cohesive soil also tends to adhere to other materials. The shear strength of this bonding is referred to as adhesion. Depending on the moisture content of the soil, the strength of adhesion may be greater or less than the strength of cohesion.

4-3.2 PURE FRICTIONAL SOILS (Ref. 1)

A pure frictional or cohesionless soil is defined as one which has little or no strength when air-dried and little or no strength when submerged, i.e., the opposite of a cohesive soil. Its shear strength is a function of applied loading—the shear strength on any plane is zero when the normal stress on the plane is zero. Soils of this type are coarse-grained with particle sizes usually in excess of 0.002 mm in diameter. Sand, gravel, and silty soils are commonly assumed to be pure frictional soils.

The shearing resistance of the soil is designated by its angle of internal friction φ. The tangent of the angle of internal friction is analogous to the coefficient of friction between solids, but in soil mechanics the practice is to refer to the angle rather than the tangent. It is developed due to the interlocking of the soil grains and the frictional resistance to sliding between the grains. As such, the frictional angle is greater with irregular, rough-shaped (or angular) grains rather than smooth, rounded grains; and it increases with the density since this increases particle interlocking.

Shearing action in soil of this type characteristically produces a change in volume. This change may be either an increase or a decrease, depending on the density of the soil; hence the degree of the soil's confinement has a significant effect upon the behavior. If the grains are very loosely packed, they change to a more compact arrangement as they move due to shear. There is, of course, a definite limit to the volume decrease which can occur. When the grains are tightly packed, shearing causes the grains to move around one another, and an increase in volume (dilitation) occurs.

Pure frictional soils are not particularly sensitive to the presence of water. If the soil is not totally submerged, water will cause an apparent cohesion to exist between the soil grains due to capillary attraction; however, if the soil is totally submerged, the cohesive effect is lost and friction is reduced slightly due to lubrication between the particles.

Within the range of loading associated with vehicle design, the soil shear strength on an internal surface may be considered to be proportional to the normal load on the surface and independent of the rate of shearing displacement. Shear strength is thus expressed as

$$s = p \tan \varphi \qquad (4\text{-}1)$$

where

s = shearing strength, psi
p = effective normal stress on the surface of sliding, psi
φ = angle of internal friction, deg

4-3.3 GENERAL SOILS (Ref. 2)

Most soils are neither pure cohesive nor pure frictional but are a combination of the two. The properties of general soils are a combination of those discussed in paragraphs 4-3.1 and 4-3.2, depending upon the proportion of each type present. Relatively small quantities of silt or sand in a clay will introduce certain frictional effects, and small amounts of clay with sand or silt will produce some cohesion. It was recognized by Coulomb that soils, in general, derive strength both from cohesion and frictional resistance. The general expression for soil ultimate strength is known as Coulomb's law and is written:

$$s = c + p \tan \varphi \qquad (4\text{-}2)$$

where c is cohesion in psi. Actual soils do not generally behave in such a simple manner, but a simplified relationship is required to facilitate design. Determination of the proper c, p, and φ for a given set of conditions may be extremely difficult, requiring considerable judgment to obtain the correct results.

Stability problems in soil mechanics deal with

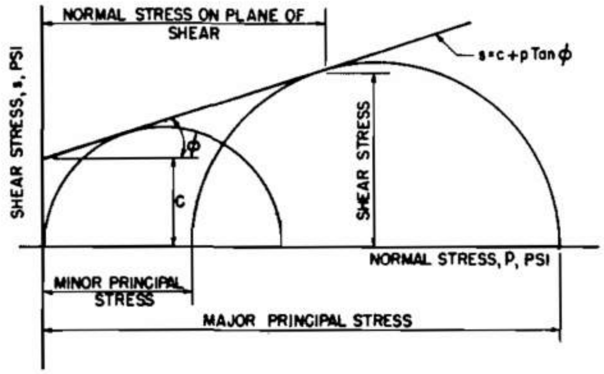

Figure 4-1. Mohr's Failure Circles for Soils

the state of stresses at the moment of incipient failure of the soil through plastic flow. In solving these problems, only strength is involved and no consideration is given to the corresponding state of strain, unless some limitations are imposed on soil deformations due to boundary conditions. However, as a vehicle moves over a soil, sizable sinkage and slip can occur. The resistance to vehicle motion is constantly changing with the degree of sinkage, and the tractive effort varies with the percent slip. Hence, strength alone is not sufficient and it becomes necessary to consider soil stress-strain relationships which govern sinkage and slip.

4-3.4 DETERMINATION OF SOIL VALUES c AND φ (Ref. 3)

The expression for Coulomb's law is the equation of a straight line which is the envelope of Mohr's failure circles for the soil (Figure 4-1). Such a diagram is normally obtained by performing a series of laboratory tests on specimens representing the *in-situ* soil. There are many types of strength tests for soil. Among the most commonly used are the triaxial and direct shear type.

The triaxial test is a compression test performed on a cylindrical soil specimen which is subjected to an all-around confining pressure while being loaded axially (Figure 4-2). In this case, the axial stress is the major principal stress and the other two principal stresses are both equal to the confining pressure. A series of tests performed at different confining pressures enables a series of Mohr's circles to be drawn. The cohesion c and frictional angle φ can then be determined graphically from the failure envelope.

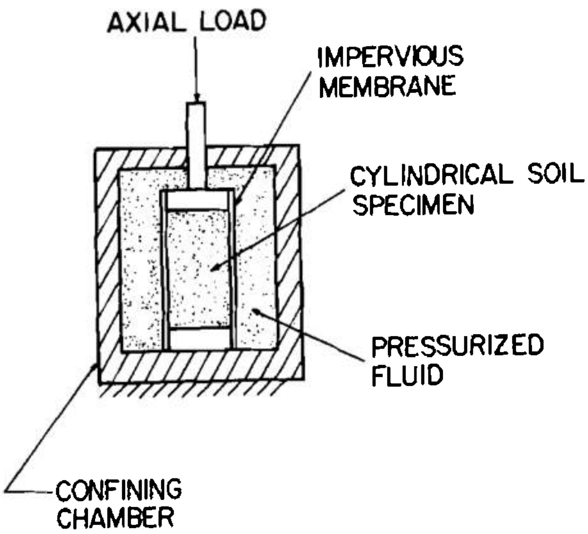

Figure 4-2. Schematic Representation of Triaxial Soil Strength Test

In the direct shear test, a specimen is forced to shear on a predetermined plane (Figure 4-3). Direct measurement is made of shearing strength and normal stress on the plane of shear. A series of tests performed with various normal loads determines points of tangency of the failure envelope for several Mohr's circles and, again, c and φ may be determined graphically.

The most important general consideration in soil strength determination is, obviously, to properly represent the *in-situ* conditions for the problem of interest. This includes such factors as degree of disturbance, pore pressures, effective confining stresses, and rate of loading. In practice, this can never be entirely achieved; hence, judgment is needed in the application of the test results.

4-4 STRESS-STRAIN RELATIONSHIPS OF SOILS

4-4.1 DISCUSSION OF SINKAGE EQUATION (Refs. 4, 5, 6)

An approximate relationship sometimes used in soil mechanics to relate breadth of footing, initial depth of footing, loading intensity, and settlement is

$$\frac{p}{z} = C_1\left(1 + \frac{2d}{b}\right) + \frac{C_2}{b} \qquad (4\text{-}3)$$

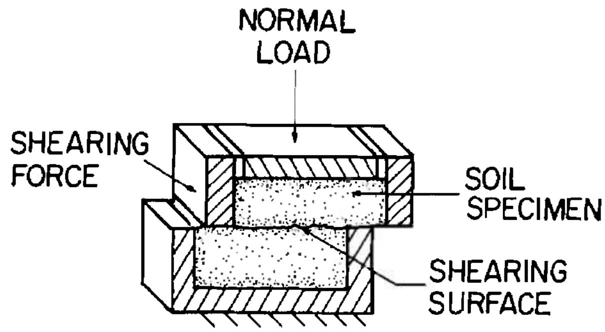

Figure 4-3. Schematic Representation of Direct Shear Soil Strength Test

where

p = loading intensity, psi
z = sinkage, in.
d = initial depth, in.
b = breadth of footing, in.
C_1 = constant due to soil frictional resistance, lb/cu in.
C_2 = constant due to soil cohesion, psi

The scope of this relationship is limited to small z for which linear relationship exists between load and sinkage. Within this range, however, the relationship is fairly independent of the size of the loaded area since size is taken into account by the characteristic dimension b.

For the case of a wheel at the soil surface, $d = 0$, and equation reduces to

$$\frac{p}{z} = C_1 + \frac{C_2}{b} \qquad (4\text{-}4)$$

However, in the case where vehicle sinkage becomes quite large with respect to the breadth dimension b, the load-sinkage curve is nonlinear and the above relationship cannot be expected to apply.

It has been suggested that load-sinkage relationships of a wheel or track can be expressed more generally by the following equation

$$p = \left(\frac{k_c}{b} + k_\phi\right) z^n \qquad (4\text{-}5)$$

where

k_c = modulus of deformation originally thought due to the cohesive characteristics of the soil, lb/in.$^{n+2}$

k_ϕ = modulus of deformation originally thought due to the frictional characteristics of the soil, lb/in.$^{n+1}$

n = exponent of deformation, dimensionless

The difference between Equation 4-5 and Equation 4-4 is just the exponent n, for the depth z, to account for the nonlinear relationships between pressure and depth when large sinkages are involved.

Other relationships which have been suggested to describe pressure-sinkage relationships are a simpler form of Equation 4-5

$$p = k\ z^n \tag{4-6}$$

where

k = modulus of soil deformation, lb/in.$^{n+2}$

n = exponent of deformation, dimensionless

and the hyperbolic equation

$$p = \frac{\dfrac{z}{C}}{A + B\dfrac{z}{C}} \tag{4-7}$$

where

C = perimeter of a plate, in.

A and B = intercept and slope, respectively, of the experimental plot of $\dfrac{z}{Cp}$ versus $\dfrac{z}{C}$, sq in./lb

In each of these cases, the unspecified coefficients and exponents are determined by fitting the equations to experimental pressure-sinkage data. In the case of rectangular areas specified by a single width dimension, the equations would only be valid if the other (length) dimension is much greater, or if the length-to-width ratio is held constant.

The most commonly used pressure-sinkage relationship for vehicle design has been Equation 4-5 which utilizes three parameters, k_c, k_ϕ and n, to describe the soil behavior. The effect of size of loaded area is accounted for by the characteristic dimension b. The soil values k_c and k_ϕ are assumed to be independent of both the size and form of the loaded area, and influence the relative magnitude of sinkage. Studies of pressure-sinkage relationships of various clays and mixtures of clays with silt and sand have shown that k_c and k_ϕ decrease in magnitude as water content increases and as the strength decreases. The exponent n influences the shape of the pressure-sinkage curve and generally will have values ≤ 1.

4-4.2 DETERMINATION OF k_c, k_ϕ, AND n (Ref. 5)

An instrument for the determination of k_c, k_ϕ, and n is shown schematically in Figure 4-4. Load W is applied, first to plate I and next to the wider plate II while a constant speed motor M, forces the respective footings down. The load is plotted as a function of sinkage z by recorder R. Thus, for each plate a separate pressure-sinkage curve is obtained. Plotting these curves on a logarithmic scale will produce practically straight parallel lines which form angle α with the z-z axis. It is evident that $\tan\alpha = n$. Thus, the exponent of sinkage can be easily determined. Abscissas a_1 and a_2 are equal to the respective quantities of the term $\left(\dfrac{k_c}{b} + k\right)$. Two equations result which may be solved for k_ϕ and k_c:

$$k_\phi = \frac{(a_2 b_2 - a_1 b_1)}{b_2 - b_1} \tag{4-8}$$

$$k_c = \frac{(a_1 - a_2) b_1 b_2}{b_2 - b_1} \tag{4-8a}$$

To facilitate the computation, an alignment chart shown in Figure 4-5 was developed.

In the process of determining these values, it was found that the variation of natural soils due to moisture content and other geological factors necessitates the presentation of results in a graph based on frequency distribution of values involved. This requires in turn a statistical evaluation of soil as the only practical basis of assessment of mobility in large areas.

4-4.3 DISCUSSION OF SLIP EQUATION (Ref. 5)

The interrelationships of thrust and slippage cannot be established on the basis of the sinkage parameters. This is because horizontal ground displacements are of a different character than that

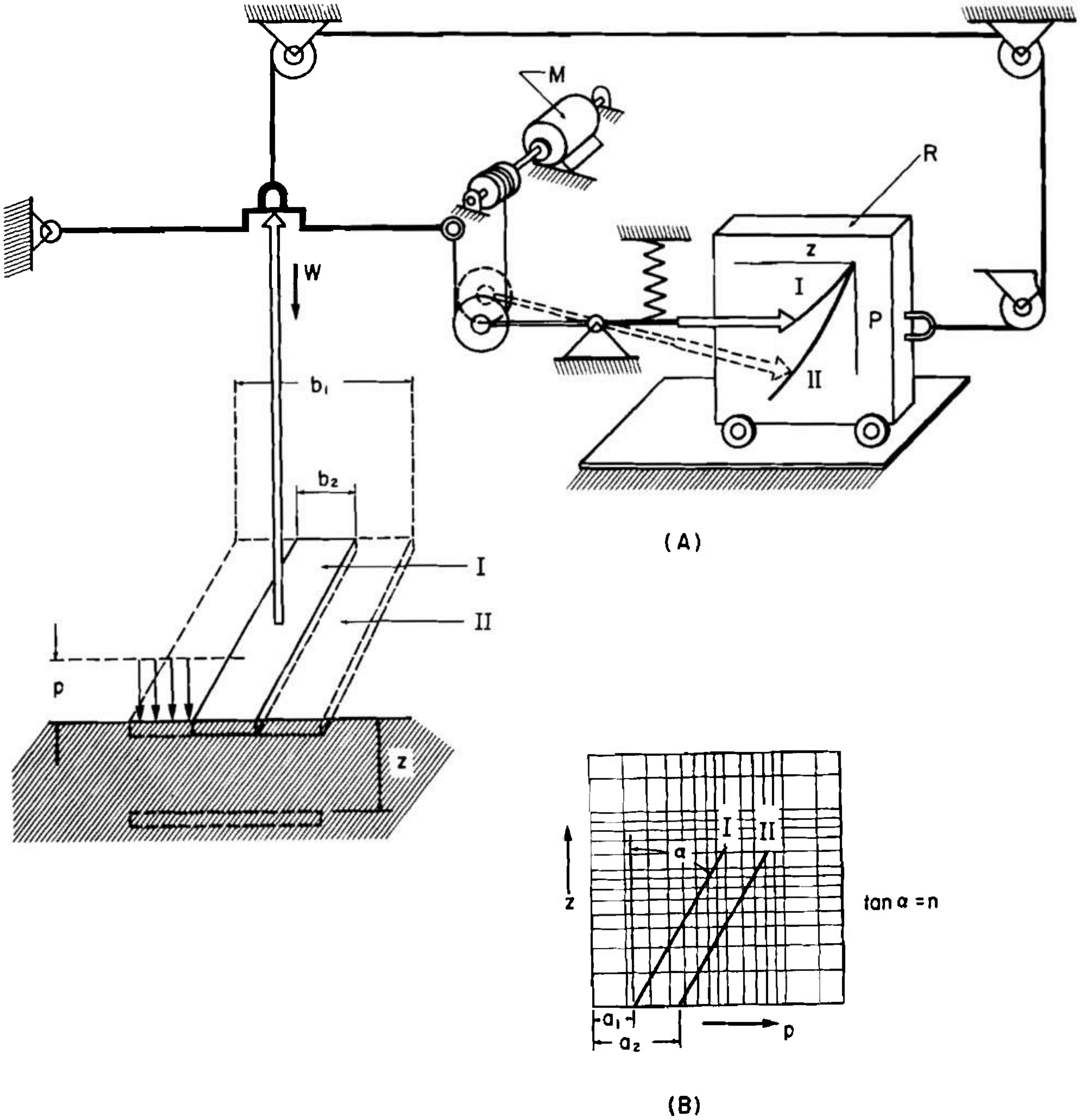

Figure 4-4. Apparatus for the Determination of Soil Values for Sinkage

represented by Equation 4-5. The maximum soil shear strength available to provide thrust is given by Coulomb's Equation 4-2. To determine the actual soil shear strength being mobilized at any instant, when it is everywhere less than maximum, requires a relationship between shear stress and shear displacements.

It has been suggested that Coulomb's equation be modified as follows to account for the change in shearing resistance as slippage takes place

$$s = (c + p \tan \varphi)\left(1 - e^{-\frac{j}{K}}\right) \qquad (4\text{-}9)$$

where

j = amount of shear deformation, in.
K = slip parameter, in.
e = base of Napierian logarithms

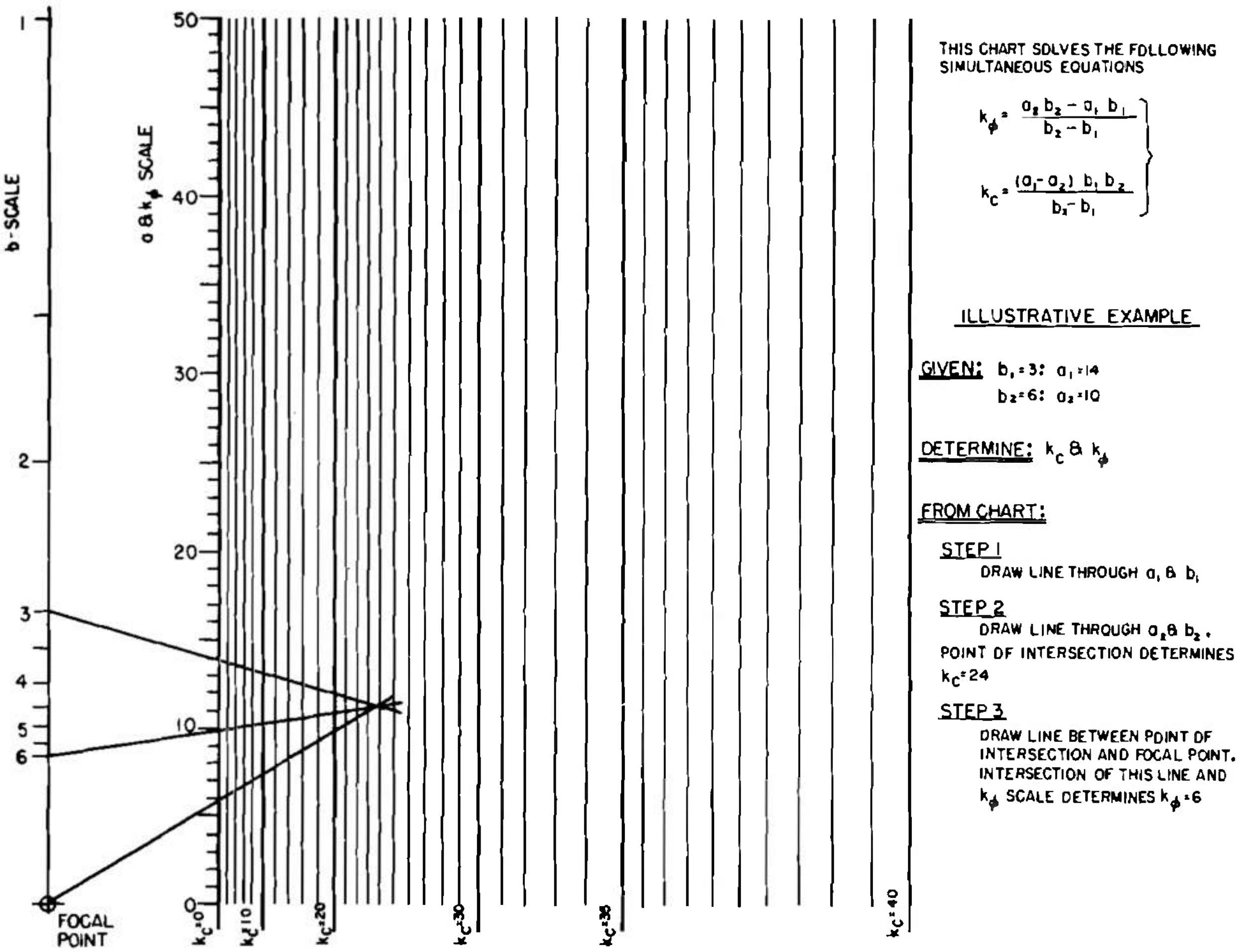

Figure 4-5. Alignment Chart for Computing Soil Values for Sinkage Equation

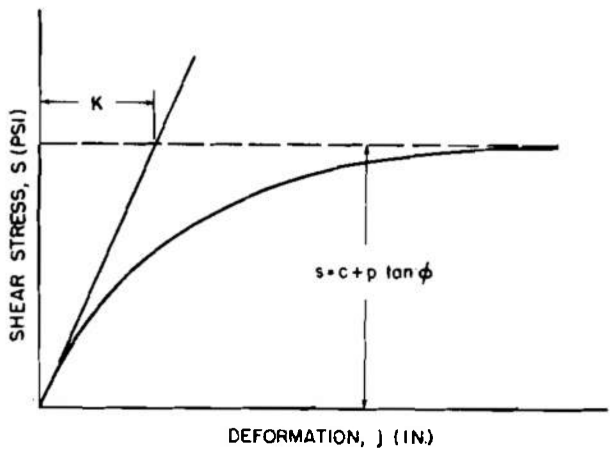

Figure 4-6. Typical Shear Stress-Deformation Curve

This equation describes a family of curves with the general form of soil shear stress-deformation relationships as shown in Figure 4-6.

4-4.4 DETERMINATION OF SLIP PARAMETER, K

The slip parameter K is obtained from the experimental shear stress-deformation curve for the soil (Figure 4-6) as the distance between the intercept of the tangent drawn at the origin and the horizontal line $s = c + p \tan \varphi$. However, the shape of the stress-deformation curve is rather erratic near the origin; therefore, an alternate method of determining K may be desirable. Equation 4-9 may be rewritten as

$$ln \left(1 - \frac{s}{c + p \tan \varphi} \right) = - \frac{j}{K} \quad (4\text{-}9a)$$

This equation plotted on semi-log paper produces a straight line the slope of which is K.

The value of soil shear deformation j may be expressed in terms of slippage i_o and distance x, from the beginning of ground-contact area to the point along the wheel or track where shearing strength s is being computed. The relationship is $j = i_o x$, which indicates that the shearing deformation increases in proportion to the distance along the contact area. The determination of slippage i_o follows the standard formula: $i_o = 1 - \frac{v_a}{v_t}$, where v_a is actual vehicle speed and v_t theoretical, based on number of revolutions and dimensions of wheel or sprocket.

4-4.5 REDUCING NUMBER OF PARAMETERS REQUIRED

To fully express soil-vehicle relationships in terms of loads, vehicle geometry, and soil strength and deformation parameters—which include consideration of both sinkage and slippage—six soil parameters are required: c, φ, k_c, k_ϕ, n, K.

The number of parameters may possibly be reduced, either through (a) discovery of explicit relationships between some of them, (b) by determination that some change so little over given ranges of conditions that they may be considered as constants, or (c) the contribution made towards accuracy of solution does not justify the complexities introduced by the consideration of the parameter. An example of item (c) has already occurred. Earlier study indicated that two parameters, K_1 and K_2, were necessary to account for affects of slippage; however, the additional complexity involved was not found to be justified by the accuracy of solution obtained, and hence, the two were combined into one parameter, K.

SECTION III TERRAIN-VEHICLE RELATIONSHIPS IN DEFORMABLE AND NONDEFORMABLE SOILS

4-5 THE CONCEPT OF FLOTATION

4-5.1 GENERAL DISCUSSION

Flotation is a term used to denote the ability of a vehicle to traverse soft soil without undue sinkage. Increased flotation is generally sought by increasing the vehicle's ground-contact area to reduce the ground pressure. The concept, however, is not always valid since experiments have shown that in some soil conditions vehicles with high ground pressure perform better than those with lower ground pressure.

Soil properties vary widely and, as a result, bearing-capacity as well as load-sinkage relationships cannot be universally expressed in a simple form. A vehicle surface load with which the vehicle has sufficient flotation for one soil condition may not be safe in another soil condition.

The task of evaluating flotation effects upon performance becomes more complicated when the variation of soil properties with atmospheric conditions is considered. A soupy mud, when frozen, will behave like concrete pavement. Hard soil, when wet, may become very soft. This may be caused even by a small precipitation since there may be considerable variation in strength when water content changes by only one or two percent.

4-5.1.1 Use of Soil Values in Evaluation of Flotation

Flotation is usually thought of in terms of bearing capacity of the soil. However, bearing-capacity equations are based upon limiting conditions of soil strength and do not usually consider associated sinkage. The ability of a vehicle to cross terrain with or without negligible sinkage as determined by the bearing capacity of soil, is defined by its angle of internal friction φ, cohesion c, and density γ. In addition, the smaller dimension b (normally representing width) of the loading area is an important factor.

For mobility problems, sinkage is an important factor; hence, to fully evaluate flotation, a numerical method of determining the load-sinkage relationships is also required. That is, once the safe loads for surface crossing are exceeded and the vehicle sinks, another set of soil values is required to determine the depth of sinkage and the ability of the vehicle to cross the given area with less sinkage. One such set of soil values is k_c, k_ϕ, and

n (Eq. 4-5). The dimensional parameter b is also required.

4-5.1.2 Effect of Track and Wheel Forms on Flotation

The above discussion has indicated that vehicle mobility cannot be compared only on the basis of ground pressure. It cannot be assumed that vehicles with the same ground pressure will have the same flotation. A track is comparable to a rectangular footing with length much greater than width. The form of ground-contact area of a wheel lies between circular and square, but closer to the circular. As such, in a frictional soil, a ground pressure that is safe for a track may not produce enough flotation for a tire; while in a cohesive soil, the opposite could be true. Hence, flotation characteristics, when considered in terms of bearing capacity, cannot be evaluated without a strict definition of the soil and form of the ground-contact area.

The same is true when flotation is viewed in terms of sinkage. Equation 4-5 indicates that the wider the ground-contact area of a wheel or track, dimension b, the deeper the area will sink for the same ground pressure p. This has been known from observations and theoretical evaluations in the field of soil mechanics where, for a given ground pressure, large foundations have shown a stronger tendency for settlement than smaller ones. Likewise, this lesson has been learned from field experience with motor vehicles; although, the reasons why the same ground pressure should cause a different amount of sinkage in the same soil were not always understood.

To further illustrate this point, take a numerical example. A standard vehicle equipped with tracks 10 in. wide and having the ground pressure, $p = 10$ psi, was considered unsatisfactory. To improve flotation, the ground pressure was reduced to 7.5 psi and the track was made 30 in. wide. The proving ground in which this design change was tested displayed the following soil values: $k_c = 30$, $k_\phi = 1$, and $n = 1$. The tests showed a deterioration rather than improvement of sinkage, which appeared to be paradox. But a check with Equation 4-5 discloses that such an outcome should have been expected:

$$z_{10} = \frac{10}{30/10 + 1} = 2.5 \text{ in.}$$

and

$$z_{7.5} = \frac{7.5}{30/30 + 1} = 3.75 \text{ in.}$$

The above example explains why a reduction in ground pressure alone may not necessarily bring a comparable improvement of flotation, i.e., reduce sinkage. Again, only a consideration of soil properties and the ground-contact area can fully explain the gains and losses involved.

4-5.1.3 Limitations of Equations

Both the sinkage equations and the bearing-capacity equations are based upon the assumption of a flat plate with uniformly distributed pressure. In reality neither tracks nor wheels satisfy these conditions; and hence, one can expect only approximately correct predictions. Total sinkage of a wheel, for example, will be greater than that of a flat plate with the same contact area and load.

One other important factor which complicates the problem is that the horizontal motion and soil reaction on the wheel or track will have considerable influence on the resultant sinkage, but the theoretical equations usually assume a vertical load on a motionless plate.

4-5.2 DISCUSSION OF SOIL-BEARING CAPACITY EQUATION (Ref. 7)

Flotation is associated with bearing capacity of the ground. As such, interrelationships of ground pressure and the form of the ground-contack area can be investigated by means of equations from the field of soil mechanics developed for the allowable bearing capacity of footings. For a rectangular footing (length > width), the most commonly used equation for bearing capacity of a surface footing is

$$(q_d)_{rect} = cN_c + 0.5\gamma\, b\, N_\gamma \qquad (4\text{-}10)$$

where

q_d = allowable ground pressure, psi
c = cohesion, psi
γ = soil density, lb/cu in.
b = footing width, in.

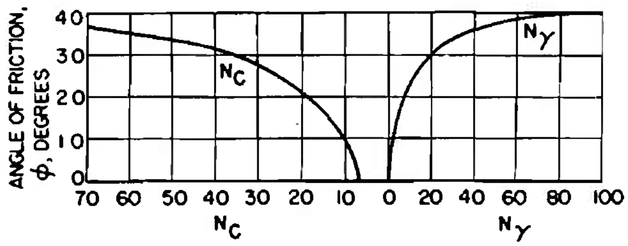

Figure 4-7. Bearing Capacity Factors

N_c and N_γ = bearing capacity factors which are functions of soil internal friction, φ, as shown in Figure 4-7, dimensionless

The equation assumes general shear failure of the soil rather than local shear failure (Figure 4-8). The former involves shearing of the soil along certain definite surfaces, while the latter refers to a progressive shearing diminishing with distance from the loaded area.

4-5.2.1 Effect of Form on Ground-Contact Area

On the basis of experimental results it was found that the equation could be empirically modified as follows to apply to square and round footings:

$$(q_d)_{square} = 1.3\, c\, N_c + 0.4\gamma\, b\, N_\gamma \qquad (4\text{-}11)$$

$$(q_d)_{round} = 1.3\, c\, N_c + 0.3\gamma\, b\, N_\gamma \qquad (4\text{-}12)$$

where, for the round footing, b is the footing diameter, in.

For a pure cohesive soil, $N_\gamma = 0$ (Figure 4-7), and the equations reduce to

$$(q_d)_{rect} = c\, N_c \qquad (4\text{-}13)$$

$$(q_d)_{square} = (q_d)_{round} = 1.3\, c\, N_c \qquad (4\text{-}14)$$

As such, it can be seen that a 30 percent greater ground pressure is permissible with the square and round footing than with the rectangular. There is not, however, any discernible difference between the form of the circle or square.

In a pure frictional soil, the cohesion c is zero, and the equations for allowable ground pressure become

$$(q_d)_{rect} = 0.5\, \gamma\, b\, N_\gamma \qquad (4\text{-}15)$$

$$(q_d)_{square} = 0.4\, \gamma\, b\, N_\gamma \qquad (4\text{-}16)$$

$$(q_d)_{round} = 0.3\, \gamma\, b\, N_\gamma \qquad (4\text{-}17)$$

In this case, in contrast to the pure cohesive soil, the form of the rectangular footing permits the greatest ground pressure; and a difference exists between the form of the square and round footing. It is also seen that the size of footing is a contributing factor to the permissible bearing capacity, the allowable bearing pressure increasing in direct proportion to size. For the case of the rectangular footing this is especially important since a given plane area can be arrived at by numerous combinations of length and width.

4-5.2.2 Effect of Local Shear Failure (Ref. 7)

For local shear failure the load-settlement curve has no well defined maximum. An approximate value of bearing capacity is obtained from the same equations by assuming that the cohesion c, and friction $\tan\varphi$, are equal to two-thirds of the actual value. Equation 4-10 then becomes

$$(q_d)_{rect} = \frac{2}{3} c\, N'_c + 0.5\, \gamma\, b\, N'_\gamma \qquad (4\text{-}18)$$

where N'_c and N'_γ are obtained from Figure 4-7 using, instead of φ, a value φ' such that

$$\tan\varphi' = \frac{2}{3} (\tan\varphi).$$

The other equations (4-11 to 4-17) are similarly modified.

4-5.2.3 Illustrative Examples

A few examples are given to illustrate the points presented in the preceding paragraphs. A soil tract is to be crossed without appreciable sinkage ($z = 0$) by both a tracked vehicle and one with low pressure tires. The tracks are 20 in. wide and the ground-contact area of the tire is approximately circular and of 20 in. diameter.

First assume the soil to be primarily frictional with $c = 0.1$ psi, $\varphi = 30°$, and $\gamma = 106$ lb/cu ft. From Figure 4-7, $N_c = 35$ and $N_\gamma = 20$. Substituting into Equations 4-10 and 4-12 gives as allowable ground pressures for the track and wheel:

$$p_{track} = 0.1 \times 35 + 0.5 \times \frac{106}{1728} \times 20 \times 20 = 15.8 \text{ psi},$$

$$p_{tire} = 1.3 \times 0.1 \times 35 + 0.3 \times \frac{106}{1728} \times 20 \times 20 = 12.9 \text{ psi}$$

In this case, the permissible track ground pressure is 22 percent greater than that for the tire.

Next assume a cohesive clay soil for which $c = 3$ psi, $\varphi = 5°$, and γ again $= 106$ lb/cu ft. N_c and N_γ from Figure 4-7, are equal to 7 and 1, respectively; and hence,

$$p_{track} = 3 \times 7 + 0.5 \times \frac{106}{1728} \times 20 \times 1 = 22 \text{ psi},$$

$$p_{tire} = 1.3 \times 3 \times 7 + 0.3 \times \frac{106}{1728} \times 20 \times 1 = 28 \text{ psi}$$

In this case, the permissible tire ground pressure is 27 percent greater than that for the track.

The significance of track dimensions for the frictional soil can also be seen from the first assumed conditions. If the ground-contact area of the track were kept constant by doubling the length and halving the width, the allowable ground pressure would change to

$$p_{track} = 0.1 \times 35 + 0.5 \times \frac{106}{1728} \times 10 \times 20 = 9.6 \text{ psi}$$

This is a significant reduction from the 15.8 psi permissible ground pressure previously computed. Hence, both tracks cannot be loaded to the same extent in frictional soils despite the fact that the ground-contact area is the same.

For the cohesive soil the allowable ground pressure would change to

$$p_{track} = 3 \times 7 + 0.5 \times \frac{106}{1728} \times 10 \times 1 = 21.3 \text{ psi}$$

which is a negligible reduction for practical purposes.

4-5.3 CONDITION OF SOFT PLASTIC LAYER OVER HARD SUBSTRATE

4-5.3.1 Determination of Allowable Ground Pressure (Ref. 8)

Previous considerations have not recognized any variance in soil conditions with depth. However, in general, the conditions of the soil surface vary considerably with seasonal changes and weather conditions. A soil condition which is quite firm in cold winter months can become impassible with spring thaw, then return to a firm condition as the soil dries in summer. Under the action of heavy rain the firm soil may again become very soft. These effects usually influence the soil only to a depth of a few feet and, hence, the soil cannot be considered a semi-infinite homogeneous mass. In actuality there is a transition from soft to firm which is usually gradual and not clearly defined. As such, rather than being a homogeneous continuum, the soil condition is more nearly analogous

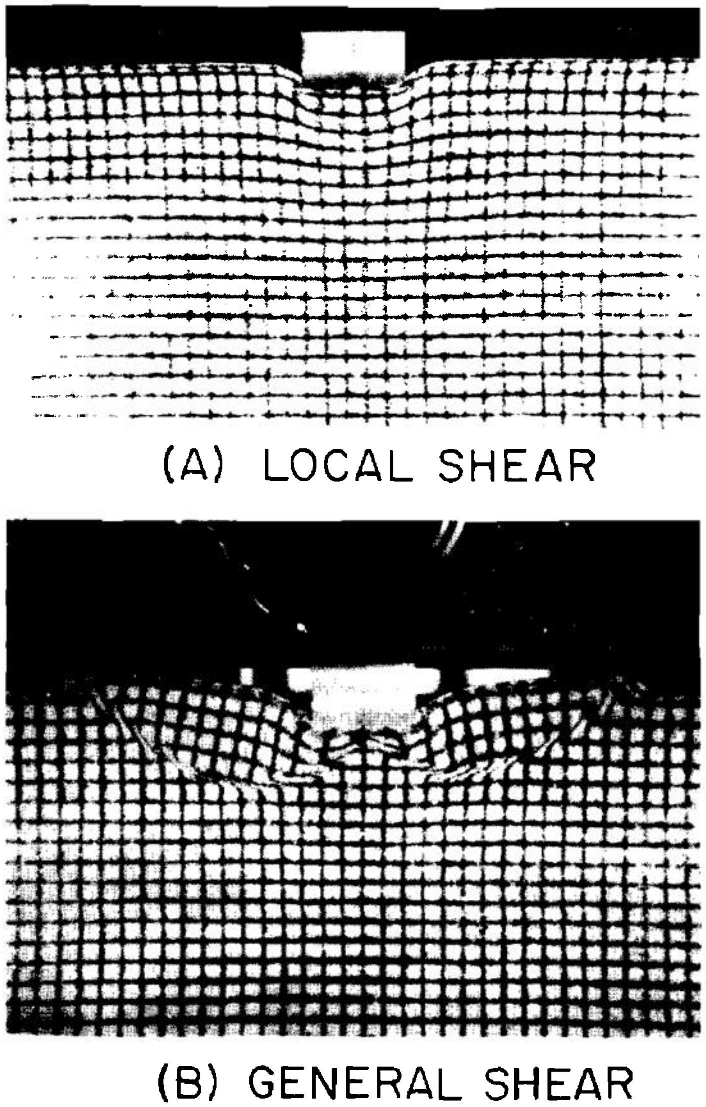

Figure 4-8. Failure of Soil Beneath a Loaded Area

to a very soft plastic layer resting on a rigid base.

The previously discussed bearing-capacity formulas cannot be used to determine safe loads or flotation for this condition. Based on the theory of plasticity and conditions for plastic flow, it has been suggested that the admissible ground pressure which a top layer of soft soil-underlain by a hard bottom-will support, can be determined by

$$p = \frac{sb}{2h} \tag{4-19}$$

where

p = average ground pressure, psi
s = shear strength, psi
b = width of loaded area, in.
h = thickness of soft layer, in.

The equation applies only when the length of the loaded area is much greater than the width (length 10 or more times larger than width).

4-5.3.2 Determination of Sinkage

Equation 4-19 determines an allowable ground pressure which permits a vehicle to float on top of a very soft soil. However, soft soil layers may be so weak that the stress value s permits only a prohibitively small ground pressure which could be achieved only through unacceptably large tracks or wheels. Hence, for some soil conditions the ground pressure for complete flotation must be exceeded. Equation 4-19 indicates that the thinner the soft layer, the greater the allowable pressure. As a first approximation we may assume that for a given ground pressure p sinkage would occur until the depth of the soft layer under the loaded area became reduced to $h_1 = \frac{sb}{2p}$. Sinkage z, in inches, would then be $(h - h_1)$ or

$$z = h - \left(\frac{sb}{2p}\right) \tag{4-20}$$

4-5.3.3 Interpretation of Equations

In Equation 4-19 the soil strength parameters are lumped in the terms of the strength s. For a constant p, an increase in ground-contact area will result in a decrease in sinkage. This is in contrast to the relationship of Equation 4-5 which indicates that for a constant pressure p, an increase in width b results in an increase in sinkage z, assuming the length is much longer than the width (see paragraph 4-5.3.1). The reason is that, for the soft layer, the soil is confined by the adhesion at the interfaces; while for firmer soils, the greater width extends the stress levels to greater depths. The length dimension is not included in either formula; therefore, sinkage is independent of length.

Thus, depending on the assumption of the homogeneity of the soil mass, the computation of sinkage and influence of parameters can be quite different. It is recommended that Equation 4-5 be used in general. Sinkage will be of greatest concern when it is large with respect to the size of track or wheel load area. As such, if the depth of the soft layer is slight, then sinkage will not be a serious problem since the vehicle will be mobile on the harder stratum. When the depth of the soft layer is large, then the assumption of a homogeneous continuum is more suitable.

4-6 SOIL THRUST

4-6.1 DEFINITION

A vehicle traveling over the ground surface utilizes the shearing strength of the soil to develop its horizontal propelling force. The total force capable of being developed into propelling force is called "soil thrust."

4-6.1.1 Application of Coulomb's Equation (Ref. 5)

The ultimate soil shearing strength is determined by Coulomb's Equation, 4-2. It has been suggested that maximum thrust H, in pounds, for a wheel or track can be approximated as the product of the shearing strength and the ground-contact area, i.e.,

$$H = sA = Ac + W \tan \varphi \tag{4-21}$$

where

A = area of contact, sq in.
W = total vehicle weight and is equal to pA, lb

The equation indicates that the vehicle weight, ground-contact area, soil cohesion, and internal friction are the important parameters in developing thrust. The thrust computed is the maximum which may be developed, occurring at the moment of impending shear failure in the soil. The thrust developed at any time prior to impending failure is dependent upon slippage occurring at the soil-vehicle interface and the soil stress-strain relationships. Coulomb's equation does not account for these effects.

4-6.1.2 Relationship of Vehicle Weight and Ground-Contact Area in Cohesive Soil

While Coulomb's equation quite evidently is not suitable for general application to the soil-vehicle problem, some interesting observations may be made. In a cohesive soil, φ is approximately equal to zero. Equation 4-21 then becomes

$$H = Ac$$

As such, in cohesive soil the greater the contact area of the vehicle the greater the thrust generated while the vehicle weight contributes nothing to thrust. This assumes, of course, that the shearing resistance at the interface is at least equal to the full cohesion.

4-6.1.3 Relationship of Vehicle Weight and Ground-Contact Area in Frictional Soil

For a frictional soil, cohesion c is approximately zero and Equation 4-21 becomes

$$H = W \tan \varphi$$

As such, in contrast to the cohesive soil, vehicle size adds nothing to the ability to develop thrust. In this soil, thrust developed is directly proportional to vehicle weight; hence, the latter should be as great as possible. There will, of course, be some limitation on minimum size and maximum weight as dictated by such factors as flotation requirements.

4-6.1.4 Illustrative Example

The above relationships between load, vehicular size, and soil properties can be seen in the following illustration. A vehicle weighing 10,000 lb with a ground-contact area of 1000 sq in. is to operate in an intermediate type soil with $c = 1$ psi and $\varphi = 18°$. One might ask, to increase thrust by 30 percent through a change in only one parameter, what changes would be required alternatively in contact area and weight? First of all for the given vehicle

$$H = 1000 \times 1 + 10{,}000 \times 0.325 = 4250 \text{ lb}$$

The desired increase in thrust is $0.3H$ or 1275 lb. To achieve this increase through contact area would require an area increase of 1275 sq in., or an increase of 127 percent. To achieve the increase through change in vehicular weight would require a weight increase of 3920 lb, or an increase of 39 percent. From a design standpoint, the greater than 100 percent increase in ground-contact area would probably be prohibitive. The best solution would be to increase both the vehicle size and weight. Such improvements are achieved with tractors, for example, by mounting larger tires which may be filled with water to substantially increase the vehicle weight.

4-6.2 GENERAL COMPARISON OF LOW PRESSURE TIRE AND TRACK

Thrust is a function of weight in a frictional soil. Therefore, if the vehicle weight does not give a ground pressure greater than the bearing capacity of the soil, a low pressure tire of practical size is preferred to the track since it will give better overall performance than a track. This is illustrated by the fact that pneumatic tires are used almost exclusively in desert operations on dry sand.

Thrust is a function of vehicle contact area in a cohesive soil. Hence, in a soft cohesive soil a track is preferred since a tire of equal area would be unacceptably large.

Performance on low friction surfaces—such as icy roads, slippery mud over hard substrate, or vegetation-covered terrain—is another type of problem. It can be basically viewed in terms of the same equation, however. In most cases a track will develop more thrust for a given weight, and even for a given area, since the grousers will bite through the surface layer. The distinction in this case is due to a greater effective cohesion and friction. More discussion of this situation is given in Chapter 2, Section III.

4-6.3 EFFECT OF GROUSERS OR AGGRESSIVE TREAD (Ref. 5)

Equation 4-21 requires an additional term to account for the increase in shearing force obtained through use of grousers or tire treads. This increase in thrust is defined as H' and the equation is modified as follows:

$$H = Ac + W \tan \varphi + H' \qquad (4\text{-}22)$$

The increase in thrust obtained, H', is a result of several effects which will vary with specific soil conditions; however, it will usually be only a small percentage of the value of $Ac + W \tan \varphi$. Some increase will usually be achieved in all soil types because the addition of the cleat or tread results in side shearing areas which increase the overall soil shearing area. Another advantage is that it prevents a surface of sliding from forming directly at the vehicle-soil interface. Thus the full soil cohesion c and/or friction φ can be mobilized even with a slippery surface.

Because the main effect of grousers or cleats is an effective increase in contact area, it has been suggested that the affects may be evaluated with a satisfactory degree of accuracy by considering the resistance of the lateral shear areas. Equation 4-22 then becomes

$$H = Ac\left(1 + \frac{2h}{b}\right) + W \tan \varphi \left\{ 1 + 0.64 \left[\left(\frac{h}{b}\right) \cot^{-1}\left(\frac{h}{b}\right)\right]\right\} \qquad (4\text{-}23)$$

where

h = depth of sheared area, i.e., depth of grouser or tread penetration, in.

Numerical evaluation of the spud effect in Equation 4-23 is presented in Figure 4-9 for pure frictional and pure cohesive soils. It can be seen that the effect is dependent on soil type and is of greatest importance in cohesive soils.

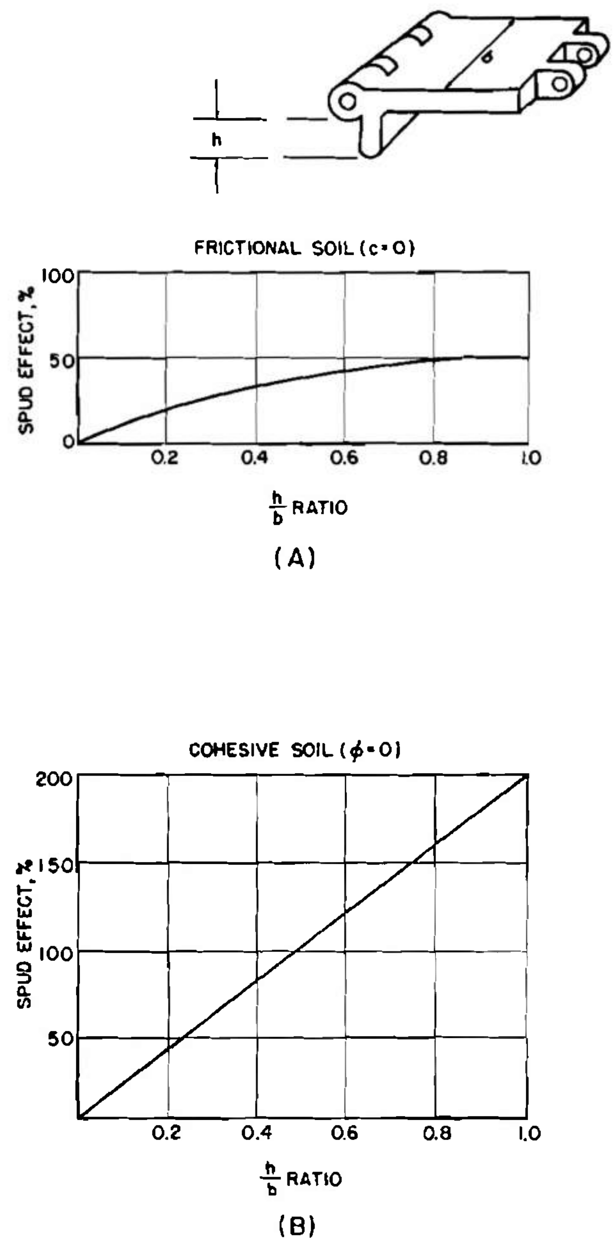

Figure 4-9. Effect of Spuds upon Traction Increase, in Frictional and Cohesive Soils, as Expressed in Terms of the Percentage of Gross Traction: $H = Ac + W \tan \phi$

4-6.4 GENERAL EQUATION OF TOTAL SOIL THRUST

4-6.1.1 Discussion of General Terrain-Vehicle Equation

The soil thrust discussed in the previous paragraphs is the maximum obtainable and occurs at some optimum percent slippage. The actual thrust developed is less than maximum if slip is smaller or greater than optimum. Thus, it is extremely important to know the relationship between thrust and slippage to obtain peak operating performance.

Equation 4-9 evaluates soil shearing resistance in terms of soil properties and slippage. Thus, a general equation for total soil thrust, in pounds, can be obtained upon integration of the shearing resistance over the entire ground-contact area, or

$$H = \iint_A s\,dA = \iint_A \left[(c + p \tan \varphi) \left(1 - e^{-\frac{j}{K}}\right)\right] dA \qquad (4\text{-}24)$$

where

s = shearing stress, psi
A = contact area, sq in.
j = amount of shearing deformation, in.
K = slip parameter, in.

4-6.4.2 Soil Thrust Related to Length of Contact Area

If constant width is assumed, the total thrust developed by a track can be computed by integrating the shear stress along the length of the contact area. Equation 4-24 then becomes

$$H = \left(Ac + W \tan \varphi\right)\left[1 + \frac{K}{i_o l}\left(e^{-\frac{i_o l}{K}} - 1\right)\right] \qquad (4\text{-}25)$$

where

c and φ are assumed constant for a given soil condition
p is assumed to be a uniformly distributed load over the entire contact area
i_o = slippage, % (expressed as a decimal)
W = load on the track, lb
l = length of contact area, in.

It may be deduced from this equation that slippage becomes of less concern as l increases; and, at $l = \infty$, the equation reduces to Equation 4-21.

For a track equipped with grousers, the total soil thrust may be computed approximately as (Ref. 5)

$$H = \left\{ Ac\left(1 + \frac{2h}{b}\right) + W \tan \varphi \left[1 + 0.64 \left(\frac{h}{b}\right) \cot^{-1}\left(\frac{h}{b}\right)\right]\right\} \left[1 + \frac{K}{i_o l}\left(e^{-\frac{i_o l}{K}} - 1\right)\right] \qquad (4\text{-}26)$$

Equation 4-25 defines the relationship between pertinent vehicle and soil parameters, and the ability to generate thrust. Thus it is possible to evaluate various vehicle concepts without building and testing full-sized vehicles. This evaluation is normally accomplished by expressing Equation 4-25 in terms of thrust-slip curves such as shown in Figure 4-10.

4-6.4.3 Illustrative Example

Figure 4-10 compares the computed thrust-slip curves for two tires on a sandy loam. The sizes of the wheels are: (1) $D_1 = 30$ in., $b_1 = 30$ in., and (2) $D_2 = 78$ in., $b_2 = 10$ in. The tires deflect so low wide tire is 10 in., and the high marrow tire is that the length of the ground-contact area of the 30 in. Hence, the ground-contact area of each tire is 300 square inches. The ground pressure under each tire is assumed to be uniform and equal to 3 psi. For the sandy loam, $c = 0.2$ psi, $\varphi = 35°$, and $K = 0.3$ in.; sinkage is considered to be negligible.

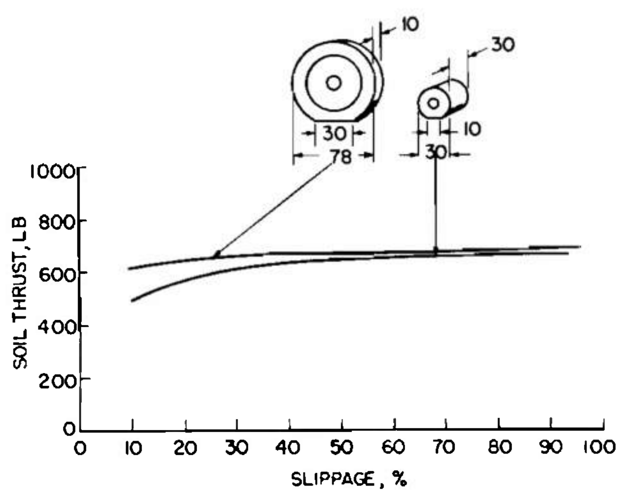

Figure 4-10. Soil Thrust Compared With Slippage for Two Wheel Forms in Sandy Loam

It may be seen in Figure 4-10 that the large diameter narrow tire produces the greater thrust, especially at lower percentages of slip. Thus, the designer would select the greater diameter in this case to obtain greater thrust with less slip.

4-6.4.4 Assumptions and Limitations

Equation 4-25 presents a relationship between soil properties, geometry of contact area, slippage, and soil thrust. It assumes that the maximum soil shear stress available to develop thrust is that described by Coulomb's equation: $s = c + p \tan \varphi$. However, since this shear stress is realized only under prescribed conditions of deformation, Coulomb's equation requires modification to be applicable to the soil-vehicle problem.

The total tractive effort exhibited by a vehicle is obtained by integrating the shear stress over the ground-contact area. Because of the complexities otherwise introduced, it has been assumed that the vehicle's contact area is of constant width b and that the vehicle's weight is uniformly distributed over the contact area. These assumptions are better for a track than a wheel, but the absolute error introduced for either case in all soil conditions is not known.

The effect of grousers is accounted for by modifying the equation to consider the lateral soil shear produced by the grouser. Although this modification increases the computed thrust, it is doubtful that it accurately describes the true effect of the grouser. However, since the contribution of the grouser is generally small compared to the total thrust developed without it, the error in most cases may not be important.

4-7 EXTERNAL RESISTANCE

4-7.1 INTRODUCTORY DISCUSSION

Resistance to motion occurs as a result of forces acting within the vehicle and upon the vehicle. The resistance due to the former is called internal motion resistance. Forces creating internal motion resistance are due to such factors as friction between moving parts, hysteresis, inertia, vibrations, etc.

The resistance due to the action of external forces acting upon the vehicle is called external motion resistance. This resistance is developed in compaction of soil under the vehicle contact area, bulldozing of the soil in front of the contact area, and drag of particles which adhere to the suspension.

To obtain optimum vehicle performance, it is necessary that vehicle elements be designed to keep external resistance as small as possible. Hence, it is necessary to define motion resistance in terms of pertinent vehicle and soil parameters.

4-7.2 RIGID WHEEL

4-7.2.1 Rigid Wheel on Rigid Surface

A perfectly rigid wheel operating on a perfectly rigid surface requires the minimum power for propulsion. This is because no deformations occur; hence, no forces of resistance develop between the medium and the wheel. This case is approached by the steel railroad wheel operating on a steel track; in fact, this type of locomotion requires the minimum propulsion power of all land vehicles. However, aside from generalities, there is little relationship of the rigid wheel on a rigid surface and the problems of off-the-road locomotion.

4-7.2.2 Rigid Wheel on Soft Ground

The resistance to forward movement of a wheel rolling over soft ground is, to a large degree, due to the work done in compacting the underlying soil. For a towed, rigid wheel, this resistance R_c, in pounds, may be estimated by the following equation

$$R_c = \frac{1}{\left(3 - n\right)^{\frac{2n+2}{2n+1}}\left(n + 1\right)\left(k_c + b\,k_\phi\right)^{\frac{1}{2n+1}}}\left[\frac{3W}{\sqrt{D}}\right]^{\frac{2n+2}{2n+1}} \qquad (4\text{-}27)$$

where

k_c, K_ϕ and n = soil values (paragraph 4-4.1)
W = wheel load, lb
D = wheel diameter, in.
b = width of wheel, in.

It may be seen from Equation 4-27 that an increase in the diameter of the wheel reduces the resistance faster than an increase in the width. This is, in effect, analogous to the concept of streamlining.

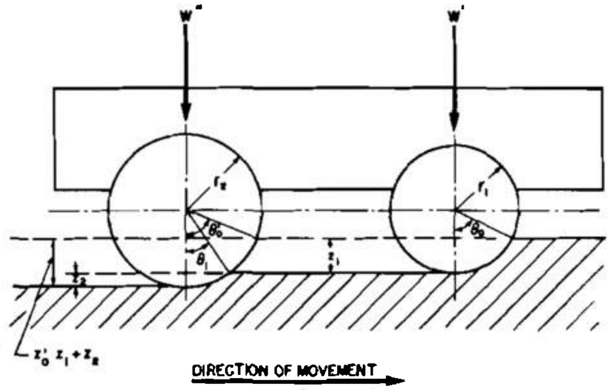

Figure 4-11. Relationships of Tandem Wheels

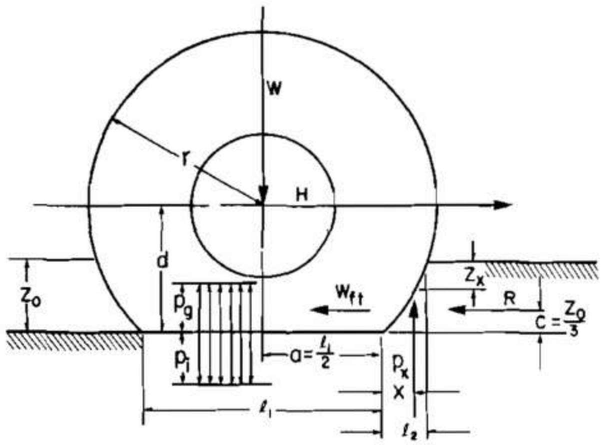

Figure 4-12. Pneumatic Tire Moving Over Soft Ground

One limitation on the validity of these equations is the assumption that the width b of the wheel may be used as the governing (or smallest) dimension in the sinkage relationship taken from Equation 4-5. For a wheel, these two parameters (i.e., width and sinkage) may be quite different. Furthermore, because of the curvature of the wheel contact surface and the nature of the distribution of shearing stresses and normal pressures on this surface, the sinkage behavior of horizontal flat plates—used to obtain k_c, k_ϕ and n—cannot be expected to produce an accurate estimate of the behavior of wheels. As the width of the wheel increases with respect to the diameter, these limitations become increasingly significant.

4-7.2.3 Rigid Wheels in Tandem on Soft Ground (Ref. 5)

Since sinkage affects rolling resistance, the arrangement of the wheels in an undercarriage will affect the total rolling resistance of a vehicle. The movement resistance of two rigid wheels in tandem may be determined using the same analytical approach as for the single wheel, taking into account the fact that the rear wheel resistance will be based upon the sinkage relative to that produced by the front wheel (Figure 4-11), and that the soil conditions will have been modified by the front wheel.

One analytical approach is derived in Ref. 5. Even using a number of simplifying assumptions, the equations are quite complex making it difficult for the designer to use judgment in their application. Since the equations are at best only approximate and their accuracy is uncertain, the complexity is not justified at present; hence, a simpler approach is recommended.

The front wheel, of course, should be analyzed by the methods given in the preceding paragraph (Eq. 4-27). The rolling resistance of the rear wheel can be analyzed with the same equation if a suitable change is made in the soil values to account for the compaction produced by the front wheel. The effect of the front wheel on the soil will be to increase the slope of the pressure-sinkage curve for the rear wheel. This may be accounted for by an increase in any one of the soil values k_c, k_ϕ and n. Because of the lack of information on this problem, it is not possible to state any general rules for estimating the increase in the soil parameters. However, approximate values may be obtained experimentally using a modification of the methods given in paragraph 4-4.2. The modification consists of unloading the plates after a given sinkage and recording the pressure-sinkage curves upon reloading.

4-7.3 PNEUMATIC TIRE ON SOFT GROUND (Refs. 5, 9, 13)

The following discussion is limited to soil conditions strong enough to resist a pressure p_g which is equal to the tire inflation pressure p_i, plus the pressure produced by the stiffness of the tire carcass p_c. In this case, the tire may deflect and a portion of the tire-soil interface will be approximately flat as illustrated in Figure 4-12. Whether

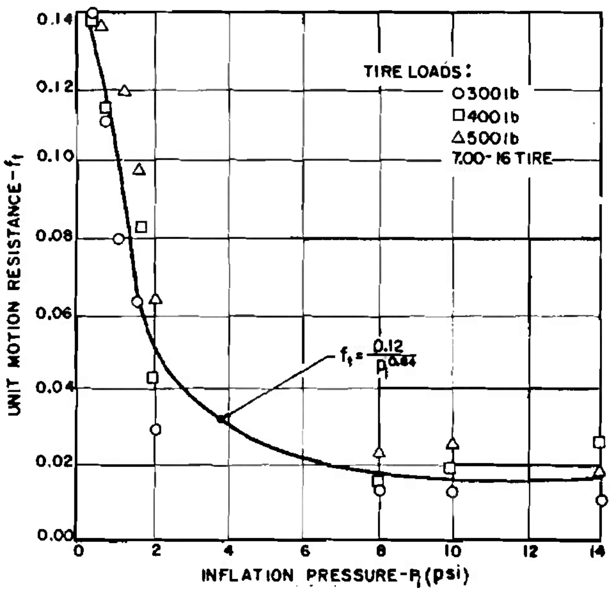

Figure 4-13. Standard Tire Deflection Resistance Curve

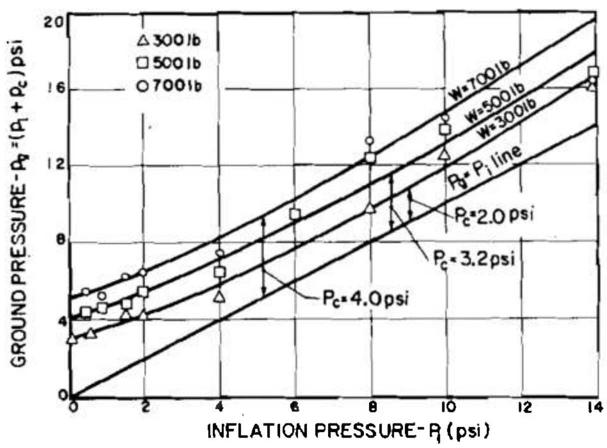

Figure 4-14. Standard Curves of Carcass Stiffness for Different Loads

or not such a condition will occur depends upon the load-deflection characteristics of the tire and the wheel load.

The compaction resistance R_c, in pounds (if pressures are expressed in psi), may be approximated by

$$R_c = \frac{[b(p_i + p_c)]^{\frac{n+1}{n}}}{(k_c + bk_\phi)^{\frac{1}{n}}\ (n+1)} \tag{4-28}$$

where b = width of the tire, in.

Additional resistance due to deflection of the carcass R_d must also be considered. It has been found that this resistance can be expressed with sufficient accuracy by

$$R_d = \frac{Wu}{(p_i)^a} \tag{4-29}$$

where W is the wheel load; u and a are constants to be determined experimentally as described below. The tire must be moved under various loadings and inflation pressures over a hard surface. If the surface is lubricated, the unit motion resistance f_t to towing is caused solely by inelastic work of the tire carcass. The unit motion resistance can be plotted with respect to inflation pressure p_i to produce a curve of tire deflection versus resistance (Figure 4-13).

Experimental results have shown that the curve obtained is hyperbolic in shape and can be represented by the equation

$$f_t = \frac{u}{(p_i)^a}$$

The constants, u and a, are determined as the slope and intercept of the straight line (psi and dimensionless, respectively) obtained from a logarithmic plot of the test results; f_t has units of lb (motion resistance) per lb (wheel load), hence dimensionless.

Thus, the total tire resistance R, neglecting bulldozing and drag, is obtained by combining Equations 4-28 and 4-29 as $R = R_c + R_d$.

The value of p_c for various inflation pressures may also be experimentally determined. On a rigid surface, the area of the tire print can be measured for various loads W, and tire pressures p_i. By dividing the load by the tire print area, the theoretical average ground pressure p_g is determined. Thus, the pressure due to carcass stiffness is determined as $p_c = p_g - p_i$ for various loadings and tire pressures (Figure 4-14).

Since resistance is a function of inflation pressure, an optimum inflation pressure exists for minimum resistance. This may be found by differentiating with respect to p_i and determining the minimum value of the derivative.

$$\frac{dR}{dp_i} = \frac{d(R_c + R_d)}{dp_i}$$

4-7.4 LIMITING CONDITIONS BETWEEN RIGID AND SOFT WHEELS (Refs. 5, 13)

Whether a tire acts as a rigid wheel or a soft tire depends on the relative magnitude of the tire inflation pressure and the maximum strength of the soil. If the maximum soil strength is much greater than the inflation pressure p_i and if the wheel is loaded so that the ground pressure exceeds the inflation pressure, then the tire will deflect and the condition of Figure 4-12 will apply. However, if the tire pressure is greater than the soil strength, the soil-tire contact area will be round and the case of the rigid wheel will be more nearly approximated. To determine the point at which a wheel behavior is more nearly rigid than soft, the following equations are established:

$$p_i + (p_i)_{cr} = \frac{W(n+1)}{\left\{b\left[\frac{3W}{(3-n)(k_c-bk_\phi)\sqrt{D}}\right]^{\frac{1}{2n+1}}\left\{D-\left[\frac{3W}{(3-n)(k_c-bk_\phi)\sqrt{D}}\right]^{\frac{2}{2n+1}}\right\}^{1/2}\right\}} \quad (4\text{-}31)$$

where $(p_i)_{cr}$ is the inflation pressure for the given soil conditions and tire size above which the wheel may be considered rigid with respect to the soil and below which it produces a flat ground-contact area. This equation is an approximation which can be relied upon only to show trends. Since it is quite complex, it should be used only if no other information is available to indicate whether or not the tire behaves rigidly.

4-7.5 COMPACTION RESISTANCE OF A TRACK (Ref. 5)

If it is assumed that a track can be approximated by a uniformly loaded strip of length l and width b; the resistance, in pounds, to forward motion due to soil compaction may be estimated by the following equation:

$$R_c = \frac{1}{(n+1)\ (k_c + bk_\phi)^{\frac{1}{n}}}\left(\frac{W}{l}\right)^{\frac{n+1}{n}} \quad (4\text{-}32)$$

where

k_c, k_ϕ, and n = soil values (paragraphs 4-4.1)
W = load on the track, lb

Since the track length is usually much greater than its width and the contact surface is nearly horizontal, the sinkage equation is probably more applicable to the prediction of track compaction resistance than wheel resistance.

4-7.6 WHEEL AND TRACK BULLDOZING (Refs. 4, 5, 10)

Rolling resistance covered in the preceding paragraphs is based upon compaction of the soil by the wheel or track. This is not the only cause of motion resistance, however. As sinkage increases, a wheel or track may tend to push, or bulldoze, the soil which lies ahead. The resulting resistance to motion developed is approximately computed by means of the soil mechanics equations for passive earth pressures on retaining walls, modified to apply to vehicle bulldozing phenomena. One such equation is given in Ref. 5.

A simpler method to use is based upon a graphical solution of Coulomb's theory for passive earth pressures. From a knowledge of track or wheel shape and sinkage and the shape of the bulldozed soil, the bulldozing forces can be estimated. The theory applies to the two-dimensional case, i.e., neglects contributions of side effects to bulldozing forces. Model studies in cohesionless soils have shown that the three-dimensional effect due to the finite width of track or wheel can be approximately accounted for by using an "effective width" equal to the actual width plus twice the sinkage.

The analysis shows that bulldozing resistance increases with increasing width of tire or track. This indicates a necessity for streamlining the wheels or tracks.

It is difficult to distinguish between compaction resistance and bulldozing. Both occur to some extent in most cases where sinkage is significant. An example is shown in Figure 4-15. The methods of predicting the two resisting forces undoubtedly overlap; i.e., the total resisting force is less than the sum of the two, because the one solution assumes all of the soil is compacted and, hence, none is left to be bulldozed, and vice versa. The proper

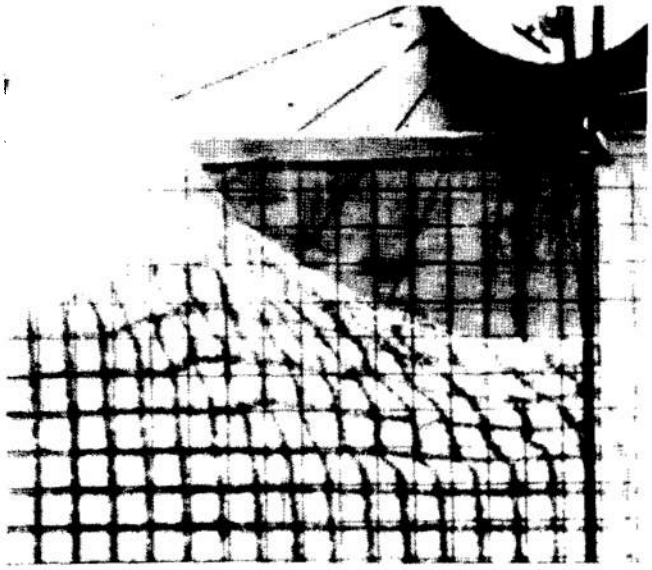

Figure 4-15. Compression and Bulldozing of Soil Beneath Towed Wheel

application of these two approaches still requires considerable judgment because insufficient knowledge is available to indicate how to separate the two effects.

4-7.7 PROCEDURE INVOLVING HALF-FLUID MUDS (Ref. 5)

The previously described procedures are not sufficient to treat the case of half-fluid muds. There is additional external resistance due to viscous drag. It is suggested that procedures available in fluid mechanics be considered to obtain more correct solutions to such problems. The drag R, in pounds, is roughly estimated from the equation

$$R = \frac{C_d \rho V^2 A}{2} \qquad (4\text{-}33)$$

where

φ = mass density of soil, lb-sec^2/ft^4

V = speed of locomotion, fps

A = vehicle ground-contact area, sq ft

C'_f = drag coefficient, expressed in terms of Reynolds number and kinematic viscosity, dimensionless

Thus, to the previous six soil parameters must be added two additional parameters pertaining to loose mud: ρ and C_d. Field measurements of viscosity can possibly be performed using standard techniques with ruggedized portable instruments.

4-8 DRAWBAR PULL

4-8.1 INTRODUCTORY DISCUSSION

The shearing strength of the ground utilized by a vehicle to develop horizontal propelling forces is called *soil thrust*. Part of the absolute thrust available is depleted in overcoming the resistance to motion. The rest—which is a useful force to accelerate the vehicle, climb slopes, or pull loads—is called *net tractive effort* or *drawbar pull*. If the resistance to motion becomes equal to the absolute thrust available, then locomotion will stop.

Drawbar pull is computed, in pounds, as

$$DP = H - R \qquad (4\text{-}34)$$

where

H = total soil thrust available, lb

R = resistance to motion, lb

4-8.2 RELATIONSHIP OF CONTROLLING PARAMETERS

It is generally assumed that the heavier the vehicle, the more it pulls. This is often true, however, exceptions to this rule exist. Neglecting bulldozing and drag for purposes of discussion, for a tracked vehicle, Equations 4-26 and 4-32 give

$$DP = \left\{Ac\left(1 + \frac{2h}{b}\right) + W_T \tan\phi\left[1 + 0.64\left(\frac{h}{b}\right)\cot^{-1}\left(\frac{h}{b}\right)\right]\right\}\left[1 + \frac{K}{i_o}\left(e^{-\frac{i_o l}{K}} - 1\right)\right] - m\left[\frac{1}{(n+1)\,(k_c + bk_\phi)^{1/n}}\left(\frac{W}{l}\right)^{\frac{n+1}{n}}\right] \qquad (4\text{-}35)$$

where W_T is the gross vehicle weight, W is the weight on one track, m is the total number of tracks, and the other symbols are as defined previously.

Here it can be seen that drawbar pull is not directly proportional to vehicle weight, but is also a function of contact area. Contact area size and vehicle weight effects will have varying degrees of influence depending upon the soil type.

The influence of vehicle weight in affecting drawbar pull is related to friction φ. To illustrate, consider a dry sand for which c may be considered zero. The modulus of deformation k_c will be very small and n will be close to 1. For this assumed soil condition, soil thrust H is directly proportional to vehicle weight, and resistance to motion increases by the square of weight, since $\frac{n+1}{n} = 2$. Therefore, for large sinkage, the drawbar pull will diminish with increasing weight. Thus, while drawbar pull is solely a function of weight in a cohesionless soil, it is not directly proportional to weight as is normally assumed.

Size and form of the ground-contact area are of importance in a frictionless soil. In a pure clay, $\varphi = 0$ and soil strength is entirely due to cohesion. For this soil, n might be considered as ½. Soil thrust H is then a function of size of contact area alone while resistance to motion increases by the cube of the weight, since $\frac{n+1}{n} = 3$. Thus, in a clay soil it is invalid to assume that increasing weight will increase drawbar pull. On the contrary, increasing weight rapidly decreases the drawbar pull.

Equation 4-34 assumes that full soil shear strength is mobilized and, as such, maximum thrust is realized. This maximum occurs at an optimum amount of slippage. When the vehicle operates at less than optimum slippage, which may be required by the overall economy of locomotion, soil thrust must be computed by Equation 4-26.

SECTION IV SOIL TRAFFICABILITY

4-9 GENERAL DISCUSSION

4-9.1 IDENTIFYING THE PROBLEM

Soil trafficability involves both the vehicle and the soil, and is defined as the ability of a soil to support a vehicle. Mobility, a companion term, is used to denote the ability of a vehicle to traverse adverse soil conditions. Soil trafficability is concerned with both the improvement of off-road vehicle performance and the prediction of vehicle performance in given adverse soil conditions. A vehicle designed to operate in a specific set of terrain conditions cannot be expected to give optimum performance over the entire range of possible conditions. It is necessary that consideration be given to the relevant parameters of both the soil and vehicle and their interrelationships to arrive at a realistic yardstick for comparison and a basis for determining the degree of compromise acceptable.

4-9.2 SOME METHODS OF APPROACH (Refs. 11, 12)

One suggested method of approach is the use of a cone penetrometer to obtain a single measurement of soil bearing-tractive capacity to predict soil trafficability. This device is a cone-tipped rod pushed into the soil to obtain an indication of soil strength (Chapter 7). The cone penetrometer's value as a tool for measuring bearing and tractive capacity lies primarily in the fact that it has been used in a great number of tests to correlate vehicle performance with soil condition. Any instrument capable of probing the soil to various depths and shearing it in a consistent manner would probably serve equally well.

Measurement is made of the cone index, i.e., the force that must be applied to the penetrometer handle per square inch of cone end area to force it into the ground. Because many soils lose strength under kneading action such as a vehicle applies, a remolding index is also used. This is obtained as the ratio of the cone index of an undisturbed soil sample and the same soil after pounding it in a mold with 100 blows of a 2½ lb tamper falling 12-in. The rating cone index is an overall or final measure of the soil's ability to support sustained traffic, and is computed as the product of the *in situ* cone index and remolding index. Each vehicle must then be tested to determine the minimum cone index value the soil must have to support the vehicle in passage in various soil conditions.

Another suggested single measurement device is a form of vane shear (Chapter 7). This device is

TABLE 4-1
RATING CONE INDEX FOR TYPICAL VEHICLES

Vehicle	Description	Rating Cone Index for 1 Pass	Rating Cone Index for 50 Passes
M29C Weasel	5,500-lb tracked, amphibious cargo carrier	20	25
D7 Engineer tractor	35,000-lb Caterpillar-type construction tractor	30	40
M48 Tank	90,000-lb medium tank	40	50
3/4-ton Weapons carrier	7,400-lb (with load of 1,500-lb) 4 × 4 truck	50	65
2-1/2-ton Cargo truck	16,300-lb (with load of 5,000-lb) 6 × 6 truck	45	60

pressed into the ground and a measurement made of force required to cause rotation. However, this device has not received as great a recognition as the cone penetrometer, partly because it is not very effective in cohesionless soils.

A more complicated method of approach is the use of a device such as the bevameter (Chapter 7) to obtain more direct measurements of both the soil's cohesive and frictional strengths. Bevameter measurements coupled with the mathematical procedures outlined in this handbook permit an analytical means for the prediction of soil trafficability.

4-9.3 SHORTCOMINGS OF EXISTING METHODS

The ability of a soil to support a vehicle statically or in motion and to provide thrust for the moving vehicle can be assessed only by consideration of relevant values pertaining to both the medium and the vehicle. It is difficult, if not impossible, to assess both soil-bearing and tractive capacity in a range of soil types by means of a single load-deformation test such as the cone index. Techniques of this type also require empirical correlation of each vehicle to the load-deformation test. Hence, they are primarily useful for determining the behavior of existing vehicles which have been correlated by means of the cone index and only of limited use in the design of new vehicles.

Use of a more complicated measurement apparatus, however, introduces severe problems for military field measurements. The present state-of-the-art would not justify the additional personnel training, transportation, and logistics problems which a complicated field measurement machine would introduce. On the other hand, such methods are probably necessary for evaluating the behavior of vehicles for which data are not available.

4-10 REQUIREMENTS OF SOIL TRAFFICABILITY EVALUATION TECHNIQUE

4-10.1 RELEVANT SOIL PARAMETERS

The thrust required for locomotion can be developed only if the soil has sufficient shear strength in excess of that required to support the vehicle without intolerable sinkage. Shear strength is a function of the soil's cohesive strength and frictional resistance. However, these properties cannot be simply described by absolute values for any given site. Consideration must be given to the influence of season, moisture, vegetation, and other environmental factors on these basic strength properties. Another important factor influencing

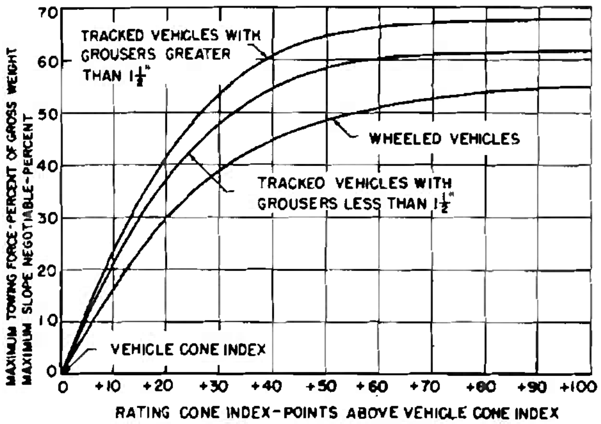

Figure 4-16. Maximum Continuous Tractive Effort on Level Ground, Maximum Slopes and Maximum Towing Force Versus Rating Cone Index Expressed as Points Above Vehicle Cone Index

trafficability is the terrain profile, since it may affect the vehicle's traction requirements.

4-10.2 RELEVANT VEHICLE PARAMETERS

Both size and geometry of the loaded area, as well as vehicle weight and ground pressure, are important vehicle parameters. These have already been discussed at length in previous parts of this chapter. Details of vehicle design also enter into trafficability such as ground clearance, type of tread, and available power.

4-10.3 ESTABLISHMENT OF FUNCTIONAL RELATIONSHIP BETWEEN RELEVANT SOIL-VEHICLE PARAMETERS

Considerable effort has been devoted to determine the relevant soil-vehicle parameters to establish functional relationships between them. This effort has included both empirical and analytical studies.

4-10.3.1 Empirical Relationships (Refs. 11, 12)

Empirically, the vast majority of effort has been undertaken by the Corps of Engineers to relate vehicle performance with cone penetrometer measurements. The minimum cone indexes are determined for which a vehicle can make one pass and fifty passes over a given site. Other tests are performed to measure the drawbar pull available in various soil conditions and this is correlated with the rating cone index for that condition.

From data collected in hundreds of tests with several types of military vehicles on fine-grained soils (loams, silts, clays, etc.) it has become feasible to predict the performance of these vehicles on the basis of the rating cone index. If the rating cone index of a given area is known, one can confidently predict whether a given vehicle will be able to cross it once, whether 50 similar vehicles can cross in the same path, how heavy a load the vehicle can tow through it, or how steep a slope the vehicle can climb.

Table 4-1 shows the rating cone index necessary for completion of one pass and 50 passes by a few typical military vehicles. Cone indexes required for 50 passes of a vehicle up a given slope or for 50 passes towing a given load on level terrain may be determined by using the curves shown in Figure 4-16. Towing forces required for vehicles may be estimated using the curves in Figure 4-17.

By the use of data collected in actual vehicle tests, a system was developed for evaluating the effects of a vehicle's characteristics (weight, contact pressure, etc.) in terms of the cone index required for the vehicle. The system comprises four formulas—one formula for self-propelled tracked vehicles, another for self-propelled wheeled vehicles, and one each for towed tracked and wheeled vehicles, respectively. These formulas are given in Tables 4-2 to 4-5, inclusive.

The *mobility index* is a dimensionless number obtained by applying certain characteristics of a vehicle to the formulas given. The mobility index can then be applied to the curve shown in Figure 4-18 to determine the vehicle cone index.

4-10.3.2 Analytical Relationships (Ref. 5)

The analytical approach to trafficability can he quite involved. Functional relationships between soil and vehicle parameters may be obtained by equations for soil strength (Eq. 4-2) and motion resistance (Eq. 4-27 or 4-32) expressed in terms of force per unit area of contact surface R_{cp}. If the resistance (Eqs. 4-27 or 4-32) expressed in terms of the maximum soil shear strength s, then no locomotion is possible. If it is much less than s, there is a reserve of strength to provide thrust for slope

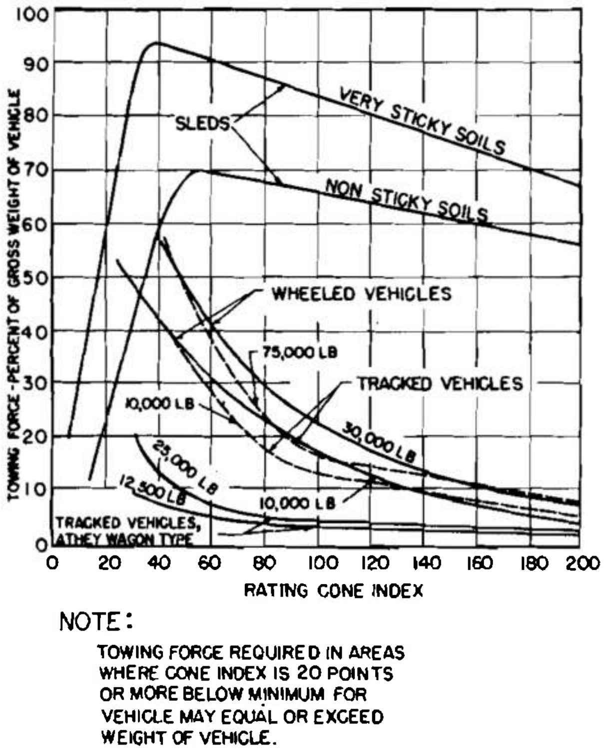

Figure 4-17. Towing Force Required on Level Ground

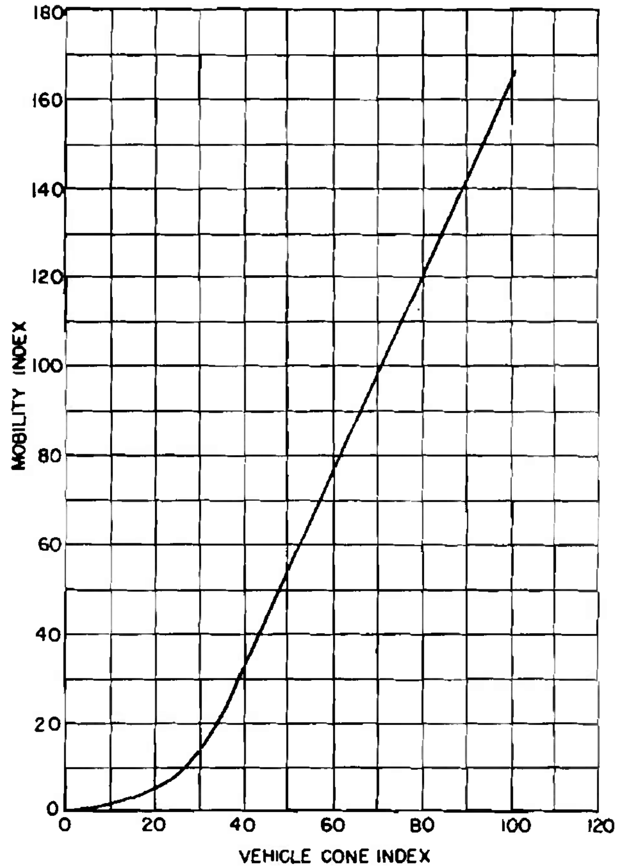

Figure 4-18. Mobility Index Versus Vehicle Cone Index

climbing, acceleration, towing, etc. The difference of maximum shear strength and resistance to motion,

$$\tau = s - R_{cp} \tag{4-36}$$

is called the trafficability function or mobility function of the given terrain. This function produces not only a criterion for locomotion, but also all the intermediate values of a mobility scale.

Geometrically, Equation 4-36 represents simple and well-established relations. A general curve of this type is shown in Figure 4-19(A). The angle formed by the tangent to the curve at the point of intersection with the vertical coordinate axis is the angle of soil friction φ. The ordinate of the point of intersection of the curve with the vertical coordinate axis is the cohesion c. The difference in ordinate between the trafficability curve and the tangent represents the traction loss due to soil compaction to a depth z, as expressed by the integral $\frac{b}{l}\int pdz$.

In sandy soils ($c = 0$), the curve will start at the origin (Figure 4-19(B)). In plastic-saturated clay soils ($\varphi = 0$), the trafficability curve starts almost horizontally, at the point denoting cohesion value c (Figure 4-19(C)). The representation of soil trafficability by means of a simple graph (Figure 4-19) discloses immediately the type of soil considered, whether frictional or cohesive. In a simple reading it gives frictional and cohesive values as well as the compaction resistance to motion in the given soil for a given loading area.

When the moisture content (MC) of soils is changed, different trafficability curves will be obtained which indicate whether a vehicle with the given loading area can negotiate a given ground or whether the ground is trafficable for the given vehicle. Figure 4-20 shows that a soil with $MC_1\%$ moisture content is nonnegotiable for a vehicle with p_1 ground pressure. However, if the latter is lowered to p_2, then such a vehicle may negotiate all

TABLE 4-2
COMPUTATION OF MOBILITY INDEX
FOR SELF-PROPELLED TRACKED VEHICLES

$$\text{Mobility index} = \left(\frac{\text{contact pressure} \times \text{weight factor}}{\text{track factor} \times \text{grouser factor}} + \text{bogie factor} - \text{clearance factor} \right) \times \text{engine factor} \times \text{transmission factor}$$

wherein

$$\text{Contact pressure} = \frac{\text{gross weight, lb}}{\text{area of tracks in contact with ground, sq in.}}$$

Weight factor: less than 50,000 lb = 1.0
50,000 to 69,999 lb = 1.2
70,000 to 99,999 lb = 1.4
100,000 lb or greater = 1.8

$$\text{Track factor} = \frac{\text{track width, in.}}{100}$$

Grouser factor: grousers less than 1.5 in. high = 1.0
grousers more than 1.5 in. high = 1.1

$$\text{Bogie factor} = \frac{\text{gross weight, lb divided by 10}}{(\text{total number of bogies on tracks in contact with ground}) \times (\text{area of 1 track shoe, sq in.})}$$

$$\text{Clearance factor} = \frac{\text{clearance, in.}}{10}$$

Engine factor: 10 or greater hp per ton of vehicle wt = 1.0
less than 10 hp per ton of vehicle wt = 1.05

Transmission factor: hydraulic = 1.0; mechanical = 1.05

soils within the moisture content between $MC_1\%$ and $MC_2\%$. In addition, its thrust will increase in the soft critical soil having $MC_2\%$ from τ_1 to τ_2, which may be considered an advantage.

The disadvantage of this presentation of soil trafficability stems from the fact that all vehicles having different values of b and l must be computed in tables in order to obtain proper soil trafficability data. This may be done quickly on electronic computers once soil values k_c, k_ϕ, n, c, and φ are known. Similar work has been done in aerodynamics with all conceivable airfoil forms which were evaluated and tabulated from the lift-drag viewpoint.

4-11 PROPOSED TECHNIQUES FOR DETERMINING SOIL TRAFFICABILITY

The empirical method for determining trafficability is primarily applicable only to existing vehicle concepts which have been tested. This is, of course, the limitation of empirical methods. For new or untested concepts, or for evaluating the effect of improvements in existing vehicles the analytical approach is necessary. It must be recognized, however, that such an approach, due to the current state-of-the-art, provides only a rough approximation.

Improvement in performance of vehicles operating off-road will most likely result from continued development of analytical methods. That is, equations based on the principles of mechanics must be refined to accurately predict behavior of a wheel or track or an entire vehicle. The phenomena involved are not yet fully understood; therefore, the methods of analysis available are of limited accuracy. They are most certainly useful, however, in predicting trends and give considerable insight into the soil-vehicle interaction problem.

TABLE 4-3
COMPUTATION OF MOBILITY INDEX FOR SELF-PROPELLED WHEELED VEHICLES

$$\text{Mobility index} = 0.6\left[\left(\frac{\text{contact pressure factor} \times \text{weight factor}}{\text{tire factor} \times \text{grouser factor}} + \text{wheel load} - \text{clearance factor}\right) \times \text{engine factor} \times \text{transmission factor}\right] + 20$$

wherein

Contact pressure factor $= \dfrac{\text{gross weight, lb}}{\text{tire width} \times \text{rim diam} \times \text{no. of tires}}$

Weight factor: greater than 35,000 lb = 1.1
15,000 to 35,000 lb = 1.0
less than 15,000 lb = 0.9

Tire factor = 1.25 × tire width, in. divided by 100

Grouser factor: with chains = 1.05
without chains = 1.00

Wheel load = $\dfrac{\text{gross weight, kips}}{\text{no. of wheels}}$ (wheels may be single or dual)

Clearance factor = $\dfrac{\text{clearance, in.}}{10}$

Engine factor: greater than 10 hp per ton = 1.0
less than 10 hp per ton = 1.05

Transmission factor: hydraulic = 1.0; mechanical = 1.05

TABLE 4-4
COMPUTATION OF MOBILITY INDEX FOR TOWED TRACKED VEHICLES

$$\text{Mobility index} = \left(\frac{\text{contact pressure} \times \text{weight factor}}{\text{track factor}} + \text{bogie factor} - \text{clearance}\right) + 30$$

wherein

Contact pressure = $\dfrac{\text{gross weight, lb}}{\text{area of tracks in contact with ground, sq in.}}$

Weight factor: 15,000 lb or greater = 1.0
below 15,000 lb = 0.8

Track factor = $\dfrac{\text{track width, in.}}{100}$

Bogie factor = $\dfrac{\text{gross weight, lb divided by 10}}{\text{(total no. of bogies on track in contact with ground)} \times \text{(area of 1 track shoe, sq in.)}}$

Clearance = clearance, in.

TABLE 4-5
COMPUTATION OF MOBILITY INDEX
FOR TOWED WHEELED VEHICLES

$$\text{Mobility index} = 0.64\left(\frac{\text{contact pressure factor} \times \text{weight factor}}{\text{tire factor}} + \text{axle load} - \text{clearance}\right) + 10$$

wherein

$$\text{Contact pressure factor} = \frac{\text{normal tire pressure, psi}}{2}$$

Weight factor:
15,000 lb per axle or greater = 1.0
12,500 to 14,999 lb = 0.9
10,000 to 12,499 lb = 0.8
7,500 to 9,999 lb = 0.7
less than 7,500 lb = 0.6

Tire factor:

$$\text{single tire} = \frac{\text{width, in.}}{100}$$

$$\text{dual tire} = \frac{1.5 \times \text{width, in.}}{100}$$

$$\text{Axle load} = \frac{\text{axle load, lb}}{1000}$$

Clearance = clearance, in.

The methods outlined in this chapter can be used for design purposes.

Predicting the performance of a specific vehicle in adverse soil conditions can most suitably be accomplished by the empirical method outlined in paragraph 4-10.2. It permits a quick, simple measurement with a lightweight, easily transportable instrument at the spot in question. This is especially valuable in military applications where changing weather conditions can radically influence trafficability, and up-to-date information is constantly required.

A great deal of success has been obtained in predicting vehicle performance on the basis of cone penetrometer measurements. Schemes have been developed for making penetrometer measurements by means of air drop techniques to obtain advance terrain information. A great deal of effort is being expended to correlate cone penetrometer measurements with various soil classification systems to permit an estimate of trafficability in the many areas for which soil classification information is known.

SECTION V GEOMETRIC TERRAIN VALUE SYSTEM

4-12 DISCUSSION OF PROBLEM (Ref. 14)

It is not only soft ground which often causes low operating speeds for off-road vehicles; hard ground, if the surface is rough, will also limit speed due to the vibrations produced in the vehicles. Vibrations are a function of (a) the speed of the vehicle, (b) the ground profile, and (c) the dynamic characteristics of the vehicle. The latter include the suspension system, distribution of mass, damping characteristics, etc. Hence, in order to evaluate rough terrain performance and speed limitations, it is necessary to consider the parameters characterizing both the vehicle and the terrain. This section deals with the latter; other chapters consider the analysis of vehicle dynamics.

There are two basic approaches to describing the terrain roughness—deterministic and statistical.

The deterministic approach represents the contour in terms of a series of periodic functions such as sine waves. There can, of course, be any number of frequency components. This approach is useful for representing a range of "typical" profiles, or where response to certain frequency and or amplitude spectra are desired.

Natural terrain consists of random, irregular contours. A statistical approach, therefore, appears to be the most suitable way of characterizing actual terrain for vehicle response analysis. Analytical tools are provided by the theory of random processes, which is an outgrowth of the probability theory.

4-13 DETERMINISTIC APPROACH (Refs. 14, 15)

The deterministic approach requires only specification of the amplitude of the various frequency components. Each component is usually assumed to be a sine wave which varies in a periodic manner (paragraph 1-19.2). Any number of these components may be combined by means of a sine series of the form

$$y = Y_1 \sin \frac{2\pi}{L_1} x + Y_2 \sin \frac{2\pi}{L_2} x + Y_3 \sin \frac{2\pi}{L_3} x + \ldots \qquad (4\text{-}37)$$

where

y = vertical displacement, in.
Y = amplitude of the component sine waves, in.
L = wavelength, in.
x = distance along the ground from the zero reference point, in.

Once the horizontal velocity of the vehicle is prescribed, Equation 4-37 may be converted into vertical displacement as a function of time by the substitution $x = vt$, where v is the vehicle speed, and t is time. Thus Equation 4-37 becomes

$$y = Y_1 \sin \omega_1 t + Y_2 \sin \omega_2 t + Y_3 \sin \omega_3 t + \ldots \qquad (4\text{-}38)$$

where $\omega = \dfrac{2\pi v}{L}$, the frequency of the wave component. Equation 4-37 may then be introduced into the equation of motion of the vehicle as a time varying boundary condition.

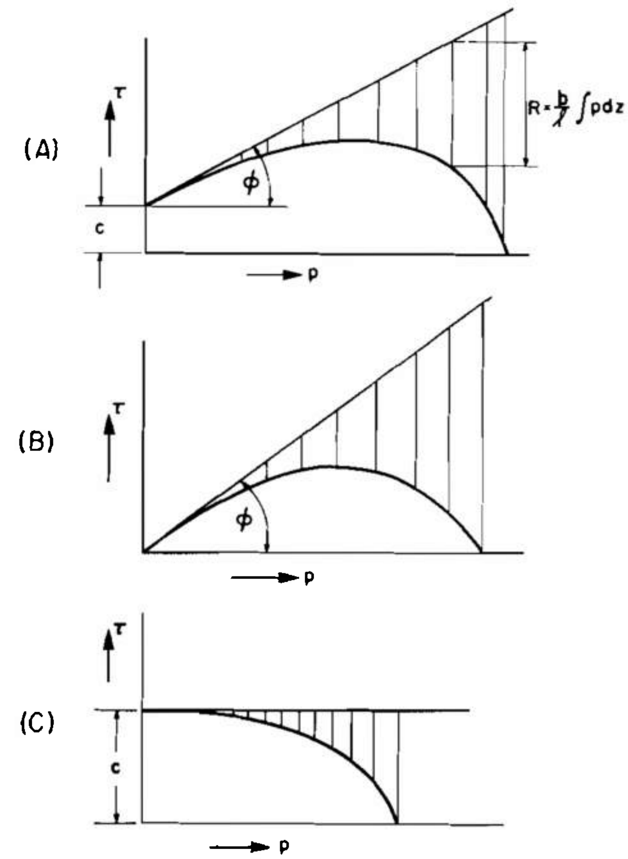

Figure 4-19. Trafficability Curves for Frictional and Cohesive Soils

If a particular terrain profile is to be represented in a deterministic manner, Equation 4-38 is replaced by a Fourier sine series which is of the form

$$y = f(x) = \frac{a_o}{2} + \sum_{n=1}^{\infty} a_n \sin n\,x \qquad (4\text{-}39)$$

where $f(x)$ is the measured terrain profile, and a_o, a_n are the coefficients given by

$$a_n = \frac{2}{\pi} \int_o^{\pi} f(x) \sin n\,x\,dx \qquad (4\text{-}40)$$

Numerical integration may be required using an electronic computer to evaluate the coefficients.

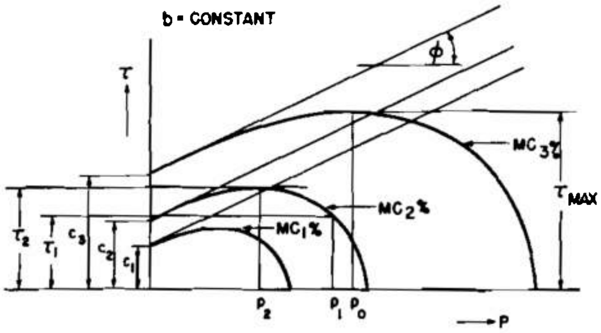

Figure 4-20. Trafficability Curves for Various Moisture Contents, MC%

4-14 STATISTICAL APPROACH (Refs. 14, 16, 17)

The question of the statistical character of an arbitrary ground surface is new and yet to be answered. The problem is becoming of greater importance and increasing attention is being given to it.

If the statistical approach is used, the terrain profile is expressed in terms of power spectral density (PSD) versus frequency curves. To visualize the physical significance of this representation, consider a random surface profile. Subdivide the profile into its basic sinusoidal frequency components. The power spectral density is essentially the mean value of the wave amplitude squared at each frequency. A plot of PSD versus frequency thus describes the variation of amplitude of each frequency component forming the composite profile.

In order to apply the statistical approach to vehicle dynamic response analysis, it is necessary to have the transfer function of the vehicle in addition to the PSD. The transfer function describes the dynamic characteristics of the vehicle. The product of the transfer function and the PSD of the ground profile gives the PSD of the vehicle response, i.e., the mean value of amplitude of vehicle motion (squared) for each frequency component. This analysis indicates, for example, the frequency range for which the vehicle vibrations will be maximum.

The statistical approach to terrain representation is too complex to permit more than a brief discussion here. The references should be consulted for a precise mathematical treatment of the subject.

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CHAPTER 5

APPLICATION OF TERRAIN-VEHICLE RELATIONSHIPS TO VEHICLE DESIGN AND EVALUATION*

SECTION I SOFT SOIL PERFORMANCE OF WHEELED VEHICLES

5-1 WHEEL FORM AND SIZE

5-1.1 RATIONAL WHEEL FORM (Ref. 1)

The design selection of the wheel form to optimize vehicle performance in soft soil conditions should be governed by consideration of the effects of form on rolling resistance and traction. The general effect of form on rolling resistance (for a towed, rigid wheel) can be studied by means of Equation 4-27

$$R_c = \frac{1}{\left(3-n\right)^{\frac{2n+2}{2n+1}}\left(n+1\right)\left(k_c+bk_\phi\right)^{\frac{1}{2n+1}}}\left[\frac{3W}{\sqrt{D}}\right]^{\frac{2n+2}{2n+1}}$$

where

R_c = rolling resistance, lb

k_c, k_ϕ, and n = soil values (paragraph 4-4.1)

W = wheel load, lb

D = wheel diameters, in.

b = width of the wheels, in.

For a discussion of procedures for the determination of k_c, k_ϕ, and n see paragraph 4-4.2.

The factor n generally has a value less than, or equal to, one. Therefore, in order to reduce Equation 4-27 to a form in which the effects of wheel diameter and width can be more easily discerned, we will assume limiting values ($n = 0$, $n = 1$) and a median value $(n = \frac{1}{2})$. Substituting these values into Equation 4-27 we obtain

$$R_c = \frac{W^2}{(k_c + bk_\phi)D} \text{ when } n = 0 \qquad (5\text{-}1)$$

$$R_c = \frac{0.88\, W^{\frac{3}{2}}}{[k_c + bk_\phi]^{\frac{1}{2}}\, D^{\frac{3}{4}}} \text{ when } n = \frac{1}{2} \qquad (5\text{-}2)$$

$$R_c = \frac{0.86\, W^{\frac{4}{3}}}{[k_c + bk_\phi]^{\frac{1}{3}}\, D^{\frac{2}{3}}} \text{ when } n = 1 \qquad (5\text{-}3)$$

In a soil type which has very low frictional resistance, the value of k_ϕ, will be small; and hence, the influence of width b is slight and motion resistance varies with wheel diameter as follows:

$$\frac{1}{D} \text{ when } n = 0$$

$$\frac{1}{D^{\frac{3}{4}}} \text{ when } n = \frac{1}{2}$$

$$\frac{1}{D^{\frac{2}{3}}} \text{ when } n = 1$$

Generally, soils display some frictional resistance, and b is of importance. The greatest influence of b is in a cohesionless soil, such as dry sand. Here k_c may be considered as zero, and motion resistance decreases with an increase in b and/or D in the following manner:

* Written by William B. Truesdale, Dr. Delon Hampton, and Rudolph J. Zastera of the IIT Research Institute, Chicago, Ill.

$$\frac{1}{bD} \text{ when } n = 0$$

$$\frac{1}{\sqrt{b}D^{\frac{3}{4}}} \text{ when } n = \tfrac{1}{2}$$

$$\frac{1}{b^{\frac{1}{3}}D^{\frac{2}{3}}} \text{ when } n = 1$$

From the above discussion it is evident that an increase in wheel diameter results in a decrease in motion resistance in all soils. An increase in wheel width decreases motion resistance in soils which possess some frictional resistance. The magnitude of the effect of variations in wheel diameter and wheel width can be discerned from the equations presented above. However, it should be noted that, in general, wheel diameter has a greater effect on rolling resistance than wheel width.

Equation 4-27 does not consider the increase in rolling resistance due to the effect of bulldozing (paragraph 4-16.2.5). Bulldozing resistance is bound to increase rapidly with increasing wheel width.

When traction is a consideration, e.g., driven wheels, wheel diameter is also found to be more important than wheel width. The soil shearing resistance which can be mobilized to develop traction is defined in Equation 4-9 as follows:

$$s = (c + p \tan \varphi)(1 - e^{-\frac{j}{K}})$$

where

s = soil shearing resistance, psi
φ = angle of internal friction of the soil, deg
c = soil cohesion, psi
p = normal unit load, psi
j = amount of shearing deformation, in.
K = slip parameter, in.

The amount of shearing deformation, j, was computed as $j = i_o x$,
where

i_o = slippage $1 - \frac{v_a}{v_t}$, % (expressed as a decimal)

x = the distance from the beginning of the ground-contact area to the point along the contact area where the shearing resistance is being computed, in.

Hence the longer the ground contact, the greater is the shearing resistance s mobilized for any constant slippage i_o. For two tires of equal ground-contact area, one of small diameter and large width the other of large diameter and small width, the large diameter obviously results in the longer ground-contact area. As a result, it is apparent that wheel diameter is more important than wheel width when traction is a consideration.

Consideration of all factors indicates that the most rational wheel form should be a large diameter, narrow wheel. Thus, the best performing wheel should be the one with the optimum D/b ratio.

5-1.2 TANDEM AND DUAL WHEELS (Ref. 1)

There are several advantages which may be gained by using two smaller wheels in place of a single large wheel. The cost of large pneumatic tires is very high, and substituting two smaller tires can result in considerable economy. Large tires are difficult to handle and also present problems with respect to heat dissipation. Where good design indicates that two wheels should be used rather than one, consideration should be given to the optimum configuration, i.e., whether the wheels should be placed in tandem or in a dual arrangement. With respect to motion resistance in soft soil, the relative merits of tandem and dual arrangements can be examined in a manner similar to that of paragraph 5-1.1.

The dual tire arrangement is similar to a single wheel of a width equal to that of the two tires. The tandem wheel arrangement approaches the case of a tire of quite large diameter. Hence, the dual wheel presents a small D/b ratio while the tandem wheel develops a large D/b ratio. As such, the tandem wheel results in a lower rolling resistance than the dual wheel and is the more desirable wheel arrangement. Additionally, the lead wheel provides some compaction of the soil in passage, thus causing a stiffer soil condition for the passage of the second wheel (paragraph 4-7.2.3).

5-2 NUMBER OF WHEELS

5-2.1 GENERAL DISCUSSION

Determination of the number of wheels required for a vehicle is guided primarily by consideration of ground pressure and traction. Ideally, it is desirable to have low ground pressure and high traction but, generally, both cannot be achieved simultaneously without extremely large diameter tires.

Almost all soils have some frictional strength. It is desirable, therefore, to have high wheel loads for the driving wheels, since the traction developed on soil possessing frictional strength is a function of the load. Sinkage, however, is also a function of load; and, if sinkage becomes appreciable a considerable portion of the traction developed must be utilized to overcome the resulting resistance to motion.

With all wheel drive, the full vehicle weight is utilized in developing traction; however, considerations to slip (paragraphs 4-4.3 and 4-4.4) show that traction is also a function of the length of the ground-contact area. As such, a large number of driving wheels, each with small ground-contact area, may not be as efficient as a few large diameter wheels with relatively long ground-contact areas. Hence, it is necessary that the advantages and disadvantages of low ground pressure with respect to sinkage and traction be weighed against each other in the decision of the optimum number of wheels.

The problem is less complex for a towed vehicle. Here the primary consideration is sinkage and the associated resistance to motion developed. Large diameter, narrow tires, arranged in tandem, provide the minimum motion resistance. The ground pressure desired will dictate the necessary number of wheels.

5-2.2 EFFECT OF WEIGHT DISTRIBUTION OVER SEVERAL WHEELS

The effect of various distributions of weight upon the rolling resistance of the vehicle may be summarized as follows (Ref. 1):

(a) Overloading the front wheels increases the overall wheel motion resistance over that which would be developed with uniform wheel loading.

(b) The effect of the overload in increasing the rolling resistance varies with the relative dimensions of the front and rear wheels.

(c) The influence is least when all of the wheels are of the same size.

(d) The influence is greatest when the front wheels are smaller than the rear. The increase in rolling resistance due to overloading the front wheels increases with an increase in the difference of the wheel diameters.

(e) In general, less rolling resistance is encountered if the greater sinkage occurs at rear of the vehicle rather than at the front.

5-3 TIRE TREAD DESIGN

5-3.1 GENERAL DISCUSSION (Ref. 1)

Off-the-road locomotion in soft soil is not possible without incurring sizable sinkage and slip. Under these conditions, the traction grooves of tire treads become impacted with soil and the tire treads play a secondary, and probably insignificant, role in developing tractive effort. This is true even when design provides for "self-cleaning" treads.

The primary influence of tread design on a tire performance is the effect on the stiffness of the tire carcass. The carcass stiffness influences both the size and the form of the loaded contact area and the character of the load distribution. Thus, it is not the tread itself but rather its effect on the stiffness and geometrical properties of the tire that are responsible for the rather unpredictable behavior of various tread patterns. Lack of investigation in this area prohibits the detailing of tread design criteria to obtain the optimum tire stiffness and geometrical properties for soft soil locomotion.

5-3.2 TREAD DESIGN CONSIDERATIONS

Tread pattern is effective in developing traction when a firm soil is covered with a thin layer of viscous fluid soil. Under this condition, the tread pattern aids in securing a firm grip between the tire and the firm soil surface. By considering the viscous cover layer as an incompressible fluid and the firm soil below as a rigid surface, the rate of sinkage of an infinitely wide tire due to squeezing the fluid out may be computed as (Ref. 1):

$$\frac{dh}{dt} = \frac{Qh^3}{4\eta l^3} \qquad (5\text{-}4)$$

where

h = thickness of the covering layer, in.
Q = load per unit of tread width, lb/in.
η = viscosity of covering layer, lb-sec/sq in.
l = dimension of tread lug, or bar, in the direction of rolling, in.
t = time, sec

Dimension l is seen to be of considerable importance in tread design. The greater this dimension, the greater is the time required to make contact with the firm soil surface. Until this contact is made, a condition of viscous friction exists between the tire and ground and little traction can be developed. Hence, a fine tread pattern is desirable. It is also desirable that the tread pattern provide unobstructed channels for the flow of the fluid being squeezed out between the lug faces and the ground, and that the tread be self-cleaning in order that the channels do not clog. Unobstructed channels are especially important due to the high viscosity of soft soil.

Equation 5-4 also demonstrates the influence of tread width on tire performance in situations involving a viscous, semi-fluid layer overlaying a firm substrate. A narrower tread results in a higher Q term which facilitates more rapid contact with the firm soil base.

Additional discussion of this subject can be found in paragraph 2-10.2.3 of Chapter 2.

5-4 STEERING CHARACTERISTICS

5-4.1 GENERAL DISCUSSION

The behavior of a wheeled vehicle in response to steering is influenced by many factors, the most important of which are: static weight distribution; geometry of the mechanical components; operating characteristics of the steering transmission (in the case of tracked vehicle); vehicle response to centrifugal forces, slope induced forces, and drawbar forces; and vehicle-ground interaction. While considerable efforts have been devoted to the study of the steering of wheeled vehicles, it has been limited almost entirely to firm (hard) surface conditions. Reference 2 presents an excellent list of references on the subject and summarizes much of the available literature with regard to the forces acting on a vehicle in a steering maneuver and the mechanics of various steering systems.

Due to the lack of knowledge of steering characteristics in soft soil, only a general discussion of the subject is warranted. This discussion will be limited to soil conditions wherein sinkage is significant and "go" or "no-go" is the primary consideration. Under these conditions, it is imperative to maintain traction at a maximum and resistance to motion at a minimum.

5-4.2 CHARACTERISTICS OF ACKERMANN STEERING

Ackermann steering is employed almost exclusively on standard wheeled military vehicles. It is a divided axle steering system which requires, for a four-wheeled vehicle, that each wheel travel in a curved path about a point located on an extension of the rear axle center line (Figure 5-1(A)). Thus, all wheels travel about a common point, a necessary condition for a vehicle to negotiate a turn without slipping. Turning without slipping is advantageous in soft soil due to the absence of the frictional forces between the tire and ground, and the absence of motion resistance due to compaction (paragraph 4-7.2.2) and bulldozing (paragraph 4-7.7).

5-4.3 CHARACTERISTICS OF FIFTH-WHEEL (WAGON STEER) STEERING

Fifth-wheel steering is accomplished by pivoting an entire axle about a central pivot (Figure 5-1(B)). The wheels of each axle maintain their initial position with respect to each other and the interconnecting axle during a turn. Theoretically, the center of rotation for a fifth-wheel steering system on a four-wheeled vehicle is also a common point for all wheels and is located at the intersection of extensions of the front and rear axle centerlines.

Fifth-wheel steering is analogous to Ackermann steering with respect to vehicle-soil interaction in negotiating a turn in soft soil. This type of steering is commonly applied to towed vehicles which are simple wagon steered because of the mechanical difficulties of controlled steering and because greater underbody clearance is required for a fifth-wheeled steering system.

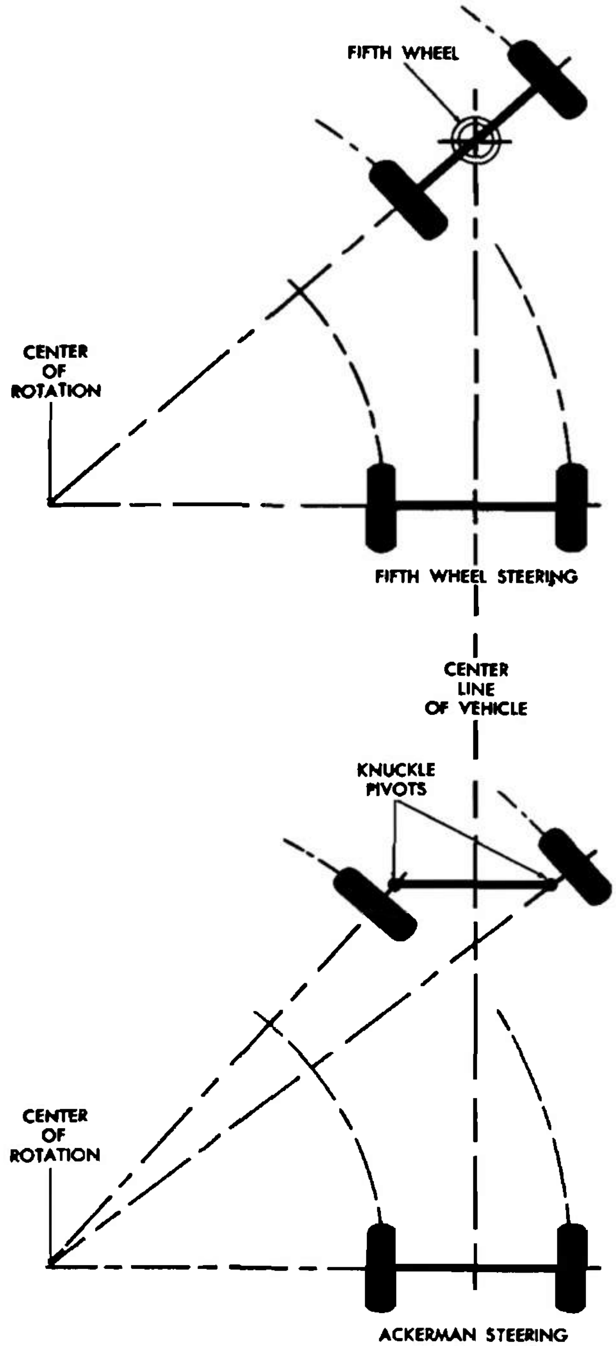


Figure 5-1. Basic Methods of Steering Wheeled Vehicles

5-4.4 CHARACTERISTICS OF SKID STEERING

Skid steering, also referred to as clutch-brake steering, is one of the simplest systems available for controlling the direction of a vehicle. It is applied primarily to tracked vehicles, although it has been applied to simple, lightweight wheeled vehicles. The system makes use of a clutch and a brake in the power train to each track. Steering is accomplished by disconnecting the power being delivered to one track by disengaging the clutch on that side, either partially or fully depending upon the radius of turn desired. As one clutch is permitted to slip, a greater torque is available to the opposite track until, when the clutch is totally disengaged, the opposite track receives the total propulsive effort developed by the engine. The rolling resistance acting on the track and the friction of moving parts cause the disconnected track system to slow down. If this is not a sufficient imbalance to cause the vehicle to slew, the inner track can be further retarded by applying the brake to that side. Pivot turns about one track are possible by locking the track on one side.

Although this system is simple it has several disadvantages. Since there is no regeneration, the power absorbed by the brake in overcoming the kinetic energy of the inner track is dissipated as heat and is lost from the system. When the coefficient of adhesion between the tracks and ground is high, the slewing force necessary to meet the requirements of a sustained turn may be beyond the capacity of the power plant; yet an appreciable amount of power, particularly during high speed operations, is being absorbed by the brake. Furthermore, this lack of regeneration adds a weight and volume penalty in the form of high capacity steering brakes needed to absorb the large amounts of power involved.

A further disadvantage of the clutch-and-brake steering system is the discontinuity of power flow in the steering cycle when the steering clutches are disengaged. This is objectionable from the point of view of steering control, especially so in the case of high speed vehicles, and can give rise to reversed steering. This condition may occur when a turning maneuver is initiated during a deceleration from a high speed or when descending a grade. Under these circumstances, the engine and transmission provide a retarding force to the vehicle. When a clutch is disengaged, as in a normal steering action, the released track will speed up instead of slowing down as it does in a normal steering maneuver, resulting in a change of course in a direction opposite to the disengaged clutch (and opposite to the direction desired). The brake must then be applied to the overrunning track to enforce a turn in the desired direction.

Another undesirable characteristic of clutch-and-brake steering concerns the loss of traction. It occurs when operating in snow, mud, or loose soil where the ground conditions barely permit the tracks to develop sufficient tractive effort to overcome the rolling resistance of the vehicle. Under these conditions, the maximum tractive effort that the two tracks can develop may be required to propel the vehicle on a straight course. When a steering clutch is allowed to slip or is totally disengaged to initiate a turning maneuver, the tractive effort previously developed by that track is reduced or lost entirely. The engine torque previously used by the declutched track is now potentially available to the opposite track for increased tractive effort, but tractive effort is limited by the shear characteristics of the ground. Under the ground conditions described, the ground will fail in shear, the slipping track will dig itself down, and, as traction is lost and resistance is increased, forward movement of the vehicle will cease. If the steering clutch is re-engaged at this time in order to resume forward motion of the vehicle, the track that was previously declutched may initially develop more tractive effort than the slipping track and cause the vehicle to veer in the direction of the slipping track (a direction again opposite to that desired). This is another condition of reversed steering.

A further criticism of the clutch-and-brake system is that the effort required on the steering brake to execute a turn is not constant under all all operating conditions. It increases in downgrade operations and decreases when operating upgrade.

Advantages of the clutch-and-brake system are its relative simplicity, its ability to make sharp skid turns as well as gradual turns by slipping the clutches or brakes, and the fact that heavy drawbar loads, adverse grades, and high rolling resistance assist the steering effort. The incorporation of interlocking and overlapping clutch-brake controls can make this system sufficiently stable for tank operation. It is particularly well suited for use in tractors and low speed vehicles. When used in vehicles having long, narrow track patterns, it is prohibitively wasteful of power.

5-5 CHARACTERISTICS OF ARTICULATED STEERING (Refs. 3, 4, 5)

An articulated vehicle consists of two or more powered units connected in tandem through hinge or pivot joints. Steering control is accomplished by applying torques about the hinge axis in order to yaw the two units with respect to each other. When applied to tracked vehicles, a differential track velocity is not necessary to effect the turn as is the case with common, nonarticulated tracked vehicles.

Investigations in the field of land locomotion (Ref. 4) have shown that the most effective way to maximize the drawbar pull-weight ratio DP/W for a tracked vehicle (assuming that the total weight remains constant) is to reduce the ground pressure p. Since the ground pressure is a direct function of the length of ground-contact area of the track L and the track width b, average ground pressure for a given vehicle weight can be reduced by increasing L, b, or both. Reference 4 indicates that to increase the DP/W ratio, it is much more effective to increase the track length L than the track width b. With the conventional tracked vehicle, however, steering characteristics are influenced by the L/T ratio (paragraph 2-3.4), where T is the distance between the track centers. The following table expresses a qualitative evaluation of the effect of the L/T ratio on steering (Ref. 4):

Range of L/T Ratio	*Steering Characteristics*
1.0 to 1.2	Very good
1.2 to 1.4	Good
1.4 to 1.6	Average
1.6 to 1.8	Poor
1.8 or more	Very poor

The steering limitations imposed by the L/T ratio effects are among the factors that have led to articulated vehicle concepts.

A complete analysis of the turning behavior of articulated track-laying vehicles is presented in Refs. 4 and 5. Some of the conclusions and recommendations resulting from this study are summarized below.

The steering response and behavior of articulated track-laying vehicles depend on many parameters. These include the magnitude of the centrifugal acceleration, the ratio of the weights of the rear section to the front section, the rear to front section length ratio, the angle of articulation, and the torque requirements to achieve or maintain various angles of articulation.

Analytical and experimental studies of vehicle behavior utilizing articulated steering at lateral accelerations to 0.4 g have produced the following conclusions:

(a) Turning response is a linear function of articulation angle and track speed.

(b) Rear to front section length and weight ratios greater than unity result in oversteering behavior, i.e., the hinge moment requirements become negative as the path curvature (radius of turn) per unit articulation angle increases with increasing track speed.

(c) Rear to front section length and weight ratios less than unity result in understeering behavior, i.e., the hinge moment requirements become positive as the path curvature per unit articulation angle decreases with increasing track speed.

(d) Rear to front section length and weight ratios of unity (symmetrical vehicle) result in neutral steering as evidenced by constant steady-state path curvature with increasing track speed.

(e) Symmetrical vehicles require minimum power for turning. (In conventional tracked vehicles, the power required for the turning process exceeds that required for straight-ahead motion.)

(f) Symmetrical vehicles require a minimum of steering system hinge moment.

As the vehicle yawing response increases with increasing articulation angle or track speed with resultant high values of centrifugal acceleration, the equations of motion developed for law values of lateral acceleration become inadequate to predict the vehicle's turning behavior. As the centrifugal acceleration exceeds approximately 0.4 g, the relationship between the yawing velocity (rad/sec) and the articulation angle becomes nonlinear. Furthermore, the functional relation becomes double-valued for a portion of the range. Analytical results indicate vehicle steering becomes unstable at some combination of track speed and articulation angle, i.e., at some yaw rate. If the vehicle can successfully negotiate the unstable region, and if the large power and steering torque requirements of high speed operation are met, a controlled high-speed skid steer with a relatively small turning radius and a large yaw rate is theoretically possible. To bring the vehicle out of the turn, the unstable region must again be traversed.

Since an oversteering vehicle can also encounter a sudden reversal of hinge torque requirements with increasing centrifugal acceleration, the steering system must have sufficient stiffness to maintain a constant articulation angle while passing through the point of discontinuity.

For low values of centrifugal acceleration, the following simplified equations may be used to describe the vehicle behavior:

$$r \approx \frac{1}{l_1}\left[2\left(\frac{1+\frac{\Gamma}{\gamma}}{1+\Gamma\gamma}\right)\right]^{\frac{1}{2}} V_t \tan\frac{\delta}{2} \tag{5-5}$$

$$V \approx V_t \sec\frac{\delta}{2} \tag{5-6}$$

$$\beta_o \approx \frac{\delta}{2} \tag{5-7}$$

where

r = vehicle yaw rate, rad/sec

l_1 = length of ground contact of the front section, ft

Γ = weight ratio of the rear section to the front section

γ = ratio of the ground-contact lengths, rear section to the front section

V_t = longitudinal velocity vector of the entire vehicle, fps

δ = input articulation angle, rad

V = velocity of the hinge point relative to the ground, fps

β_o = side slip angle of the hinge point, rad

Experimental studies, to date, indicate that articulated steering improves the cross-country mobility of vehicles, particularly in soft soil conditions. It is especially beneficial when applied to tracked vehicles because it permits large overall L/T ratios with their attendant advantages and without the steering difficulties these ratios impose on nonarticulated vehicles.

SECTION II SOFT SOIL PERFORMANCE OF TRACKED VEHICLES

5-6 TRACK SIZE AND SHAPE

5-6.1 THE RATIONAL TRACK FORM

5-6.1.1 Track Form Based Upon External Rolling Resistance Due to Compaction

For a uniformly loaded track of length l and width b the resistance to forward motion due to soil compaction may be estimated by Equation 4-32

$$R_c = \frac{1}{(n+1)(k_c + bk_\phi)^{\frac{1}{n}}}\left[\frac{W}{l}\right]^{\frac{n+1}{n}}$$

where

k_c, k_ϕ and $n =$ soil values
$W =$ total track load, lb

The value of n is always a positive number and may be considered to have an upper bound of 1. Thus, for all values of $0 < n < 1$

$$\frac{n+1}{n} > n.$$

The loading area of the track is rectangular in shape and, if this area is maintained constant, the form of the area can only be changed by varying the width and length simultaneously so that their product remains constant. By assuming a constant area, Equation 4-32 shows that for soft soil conditions (k_ϕ small), an increase in track length l and a corresponding decrease in width b will result in a decrease in motion resistance. Motion resistance decreases because the length factor occurs in the denominator to the $\frac{n+1}{n}$ power, while the width factor occurs in the denominator only to the $\frac{1}{n}$ power. It is also seen that in a purely cohesive soil in which $k_\phi \approx 0$, the width b has negligible influence on motion resistance, while increasing length decreases resistance. Hence, it may be concluded that a long, narrow track is the most rational form based upon external rolling resistance due to compaction.

5-6.1.2 Track Form Based Upon Flotation

Flotation denotes the ability of a vehicle to traverse soft soil without undue sinkage. Equation 4-5

$$p = \left(\frac{k_c}{b} + k_\phi\right) z^n$$

relates the contact pressure p, the width of the loaded area b, and the soil values k_c, k_ϕ, and n to sinkage, z. Equation 4-5 can be rewritten as

$$z = \left[\frac{p}{\left(\frac{k_c}{b} + k_\phi\right)}\right]^{\frac{1}{n}} \qquad (5\text{-}8)$$

If the loaded area of a track is kept constant, p remaining constant and the length and width varied, it can be seen that increasing the width will result in an increase in sinkage. The influence of a change in width is greatest for cohesive soils and of negligible effect in frictional soils ($k_c \approx 0$). Hence, based on flotation, the rational track form is also that of a long, narrow track.

5-6.1.3 Track Form Based Upon Slippage

Equation 4-9, $s = (c + p \tan \varphi)(1 - e^{-\frac{j}{K}}$, relates the soil shearing resistance that is developed as slippage occurs to the soil shearing deformation j. It is apparent that increasing j results in an increase in mobilized shearing resistance. It was shown (paragraph 5-1.1) that $j = i_o x$, where i_c is the slippage $1 - \frac{v_a}{v_t}$ and x is the distance from the beginning of the ground-contact area to the point along the track where shearing resistance s is being computed. From this it follows that the longer the loaded area, the greater is the shearing resistance mobilized. Hence, based on slippage, the most desirable track form is also that of the long, narrow track.

5-6.1.4 Summary

The influence of the track form on vehicle performance varies somewhat with soil type. Changing the width of the track has negligible effect on rolling resistance due to compaction in cohesive soils and on flotation in frictional soils. Increasing the width of the track decreases rolling resistance in frictional soils, but to a lesser extent than does increasing the track length. Increasing the width of the track while maintaining constant ground pressure increases sinkage (which is tantamount to decreasing flotation) in cohesive soils. Increasing track length is effective in all soils in decreasing rolling resistance, increasing flotation (for the case when area is held constant, since an increase in length results in a decrease in width), and increasing soil shearing resistance. Thus, the most rational track form for all conditions is that of the long, narrow track.

5-6.2 TRACKS AND WHEELS COMPARED

A significant advantage of a track over a wheel is the large percentage of total surface area utilized as ground-contact area. The total track surface area A_T consists of the area in contact with the ground A_1, an equal area (approximately parallel) above the road wheels A_2, and the portion which passes around the front and rear wheels A_3. Area A_3 is approximately equal to A_1; hence, about ⅓ of the total track surface area is in contact with the ground.

The total tread area of a wheel of radius r and width b is

$$A_w = 2\pi rb \qquad (5\text{-}9)$$

To obtain the same ratio of ground-contact area A_1 to total surface area A_w for a wheel as for a track at zero sinkage, a wheel must sink a considerable distance. In Figure 5-2, the angle θ must be $120°\left(\frac{120°}{360°} = \frac{1}{3}\right)$, and the wheel sinkage

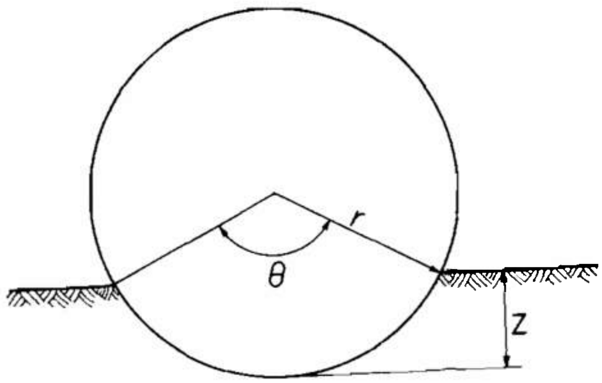


Figure 5-2. Wheel Sinkage-Ground Contact Area Relationships

$$z = r \sin\frac{\theta}{2} \tan\frac{\theta}{4} = 0.50\, r \qquad (5\text{-}10)$$

The above computation is made assuming equal areas of ground contact ahead and behind the wheel. It is more reasonable to expect that a rut will be formed behind the wheel; therefore, even greater sinkage z will be required to obtain

$$\frac{A_1}{A_w} = \frac{1}{3}$$

Hence, when a wheel has a ratio of ground-contact area to total surface area which approaches that of a track, it will have undergone significant sinkage and considerable resistance to motion will be encountered.

The track is also superior to the wheel with respect to traction. This superiority is due to the fact that the length of the ground-contact area determines the amount of slip which will occur—wheels having shorter contact areas will slip to a considerably greater extent than tracks.

The track does, however, have several disadvantages:

(a) Resistance to sliding, especially with grousers, makes maneuvering difficult and requires a great deal of power. Heavy duty clutches are required to make operation of the vehicle possible.

(b) The track consists of metal plates that are hinged together. The hinges are constructed with metal pins on which the terminal eyes of the plates and bushings rotate. This method of connection wears quickly and absorbs a large quantity of the power available.

(c) Due to the tracks and the necessary accessories required to enable the tracks to function effectively, the weight and cost of the tracked vehicle is greater than a wheeled vehicle.
(d) Tracks are less suited to high speed vehicle operations.
(e) Tracks are noisy in operation and introduce undesirable vibrations into the vehicle.
(f) Tracks account for considerable power loss due to internal friction. For this reason they offer a great deal of resistance towing and are, therefore, less suitable for towed vehicles.

In summary, it may be said that the significant difference of the track and wheel is the difference in the length of the respective ground-contact areas. Sufficient flotation can be obtained by providing enough wheels, however, the tractive effort of a track cannot be matched without slippage unless the diameter of the wheel is large enough to produce a sufficiently long ground-contact area. It is advisable to use wheels on towed vehicles where flotation and cost of the vehicle are prime considerations. But the track is superior to the wheel for use in propelling vehicles.

5-7 GROUSER DESIGN

The primary role of the grouser on conventional tracked vehicles is to prevent slippage from occurring along the soil-track interface thus assuring mobilization of the full shearing strength of the soil. The grousers are plates normal to the track surface which dig into the soil, forcing slippage to take place within the soil mass. The surface of slippage may, for conventional vehicles, be assumed to be tangent to the lower edge of the grousers.

Some shearing resistance is mobilized along surfaces tangent to the grouser edges. In cohesionless soils, where shearing resistance is a function of load, this will be of negligible magnitude as compared to the shearing resistance mobilized along the slip surface below the track. However, in soils which possess little or no frictional resistance, e.g., cohesive soils or fully saturated soils, the shearing resistance is independent of the normal load and the additional slip surface area along the side edge of the grouser can significantly increase the total mobilized shear resistance. Hence, the greater the size of the grouser, within practical limits, the greater the amount of soil shearing strength mobilized. As such, the limiting grouser dimensions should be governed by vehicle considerations such as stress concentrations induced in the track, clearance required, etc.

5-8 TRACK SINKAGE AND BOGGING DUE TO SLIP

Track sinkage due to slip occurs as a result of localized shear failure in the soil permitting material to be carried along under the track from the front to the rear of the vehicle. This "digging-in" phenomenon results in sinkage of an entirely different nature than that resulting from settlement due to vertical load.

For tracked vehicles, greater sinkage occurs at the rear of the vehicle than at the front because the removal of soil increases from zero, at the lead edge of the track, to maximum removal at the rear end of the track. The amount of soil which is removed from under the track due to slip varies with the smoothness of the track and the degree of slip incurred. With a grousered track, the grousers act somewhat like blades and scoop a considerable amount of soil when slippage occurs. The scooping action of the grousers is reduced in a soil which possesses cohesion as a result of the greater mass of soil effective in resisting the shearing action of the grousers. Consequently, less slippage will result in the case of a grousered track.

5-9 ROAD WHEEL NUMBER AND SIZE

5-9.1 SINKAGE AND PRESSURE DISTRIBUTION UNDER TRACK

The vehicle load is transmitted from the road wheels to the ground through the track. A concentration of load occurs beneath the road wheels because the track is not a rigid platform but consist of a series of hinged plates. These hinges provide flexibility, partially permitting the track to follow the ground contour as sinkage occurs. Hence, while the assumption of a rigid loading surface may be satisfactory to compare performance on hard ground, it is erroneous when dealing with soft soil conditions where track flexibility must be considered.

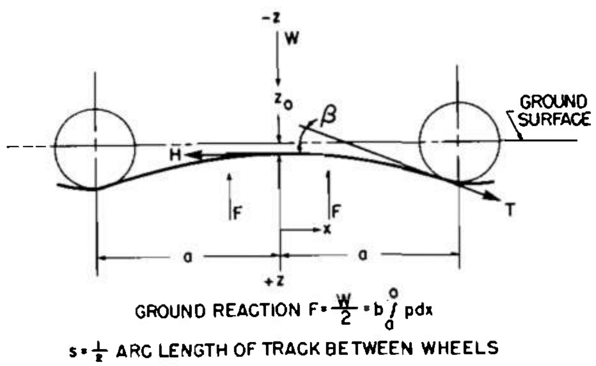


Figure 5-3. Flexible Track-Soft Ground Force Diagram

The problem of determining the pressure distribution under the track between two adjacent bogie wheels is complex and simplifying assumptions are required to permit even a general understanding. For a vehicle at rest, only the vertical forces acting on the ground need be considered. Assume that the track is a perfectly flexible, continuous strip and that the wheels are of sufficiently small diameter not to interfere with the natural deflection of the track. Figure 5-3 is a force diagram for a track section between two adjacent bogie wheels.

The pressure p at any point under the track is assumed to be proportional to the sinkage z at that point in accordance with Equation 4-5 (which is repeated here for convenience).

$$p = \left(\frac{b}{k_c} + k_\phi\right) z^n$$

Consider the case where $n = 1$. The equation of track sinkage for the central portion of the track may be expressed as a series expansion:

$$z = z_o\left[1 + \frac{x^2}{2!}\sqrt{\frac{6(s-a)}{z_o^2\, a^3}} + \frac{x^4}{4!}\left(\frac{6(s-a)}{z_o^2\, a^3}\right) + \ldots\right] \tag{5-11}$$

where

$$s = a + \frac{z_o^2 a^3}{6}\left(\frac{k_c + bk_\phi}{H}\right) \tag{5-12}$$

and

$$z_o = \frac{W}{2a\,(k_c + bk_\phi)} - 0.41\sqrt{a\,(s+a)} \tag{5-13}$$

Thus, the ground pressure at any point x distance from the z-axis may now be found from the equation $p = \left(\frac{k_c}{b} + k_\phi\right) z^n$.

Equation 5-11 indicates that sinkage z depends on the wheel spacing $2a$ and the slack in the track, a measure of which is $(s-a)$. For a perfectly tight track, $s - a = 0$, and $z = z_o$. The track would sink to a uniform depth along its length and approach the case of a rigid contact surface and uniform pressure distribution. For $(s - a) > 0$, the sinkage of the central portion of the track between the wheels would be less than that incurred with a rigid contact surface. But, as indicated by Equation 5-11, the track sinkage is not uniform and increases rapidly with the distance x from the center point. As such, from Equation 4-5, much higher pressures will be realized with increasing distance x, and the central portion of the track will not be used effectively. This illustrates the inadequacy of the "mean ground pressure" concept (vehicle weight divided by the ground-contact area) for evaluating vehicle performance in soft soil.

The development of the equations presented here is based on a number of simplifying assumptions. The equations appear to indicate correct trends but can only be considered as crude approximations.

5-9.2 SINKAGE AND NUMBER OF SUPPORTING WHEELS

Equation 5-11 expresses sinkage as a function of track length s and wheel spacing $2a$. To examine the influence of increasing the number of wheels for a given track, Equations 5-11 (considering only the first two terms) and 5-13 are combined as

$$z = \frac{W}{2a\,(k_c + bk_\phi)} - 0.41a\sqrt{\frac{s}{a} - 1} + 1.23a\left(\frac{x}{a}\right)^2\sqrt{\frac{s}{a} - 1} \tag{5-14}$$

The distance x in Equation 5-14 is expressed as a dimensionless ratio $(\frac{x}{a})$ of the half wheel spacing.

With reference to Figure 5-3, consider the addition of a third wheel placed at the z-axis. The equation for track sinkage for the track portion between the added wheel and either of the other wheels is

$$z_1 = \frac{\frac{W}{2}}{2\frac{a}{2}(k_c + bk_\phi)} - 0.41\,\frac{a}{2}\sqrt{\frac{\frac{s}{2}}{\frac{a}{2}} - 1} + 1.23\,\frac{a}{2}\left(\frac{x}{a}\right)^2\sqrt{\frac{\frac{s}{2}}{\frac{a}{2}} - 1}$$

or

$$z_1 = \frac{W}{2a\,(k_c + bk_\phi)} - 0.205a\sqrt{\frac{s}{a} - 1} + 0.615a\left(\frac{x}{a}\right)^2\sqrt{\frac{s}{a} - 1} \qquad (5\text{-}15)$$

A comparison of Equations 5-14 and 5-15 shows that when $\frac{x}{a} = 0$ (at the z-axis), z_1 is greater than z; but when $\frac{x}{a} > 0$, z is greater than z_1. This illustrates that the addition of a road wheel midway between two wheels produces a greater sinkage at the midpoint than occurred before the additional wheel was introduced, but it results in a reduction of sinkage at all other points. The net result is a more uniform track sinkage and a more uniform pressure distribution under the track. Equations 5-14 and 5-15 are based upon the assumption that the track is perfectly flexible. Actual tracks possess considerable rigidity which helps distribute the load more uniformly and reduces the necessity for a large number of road wheels.

5-9.3 SINKAGE AND LOAD DISTRIBUTION UNDER WHEELS

Consider again the case of the vehicle at rest (only vertical forces need be considered), and assume that the track is flexible and loose enough to wrap freely around the wheel as it sinks into the ground (Figure 5-4). The ground pressure at any point is, by Equation 4-5,

$$p = \left(\frac{k_c}{b} + k_\phi\right) z^n$$

From Figure 5-4, neglecting the thickness of the track,

$$z = z_o - r(1 - \cos\theta) \qquad (5\text{-}16)$$

Substituting into Equation 4-5,

$$p = \left(\frac{k_c}{b} + k_\phi\right)[z_o - r(1 - \cos\varphi)]^n \qquad (5\text{-}17)$$

The resulting pressure distribution is of the form shown on Figure 5-4. This, of course, represents an extreme case. In practical cases, the track will have some rigidity and tension due to sinkage of adjacent wheels which will limit the ability of the track to conform to the wheel shape as it sinks into the ground. Both of these effects will lead to more uniform pressure distribution under the wheel and a reduction in the maximum wheel sinkage.

In general, however, the distribution of load under a track in the longitudinal direction is not uniform but varies with the design of the track support, the track flexibility, and with the soil parameters k_c, k_ϕ, and n.

5-10 STEERING OF TRACKED VEHICLES

5-10.1 GENERAL DISCUSSION

All current standard military tracked vehicles use laterally rigid tracks. The most serious basic limitation on the performance of these vehicles is the almost universally used skid-steering method of maneuvering. Steering is accomplished by controlling the relative speed of the tracks to reduce the net longitudinal thrust of one track and, if possible, increase the longitudinal thrust of the other track. This causes the vehicle to pivot about the slower track. Both transverse and longitudinal sliding of the slower track occurs, and a large amount of power must be utilized to overcome shear forces along the ground-contact area. In soft soil

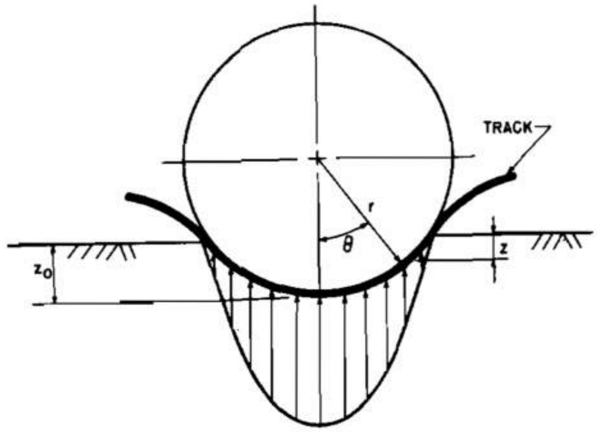


Figure 5-4. Wheel Ground Pressure Distribution

where sinkage occurs and operation is marginal, attempting to steer by this method often causes complete immobilization.

An alternate method of steering which has been the subject of recent research study is articulated steering. In an articulated steering system, two powered track-laying units are joined together at a hinge or pivot point. Steering control is exercised by applying torques about the hinge in order to yaw the units with respect to each other. Unlike the usual tracked vehicle, a differential track velocity is not used to effect the turn. Within the past few years, vehicles from 1 to 45 tons in gross weight have successfully demonstrated the high mobility capabilities of articulated steering.

5-10.2 EFFECT OF DISTRIBUTION OF GROUND PRESSURE ON STEERING CHARACTERISTICS

The steering characteristics of a tracked vehicle depend to a large extent on the load distribution on the track ground-contact area. A particular case of uniform turning when no acceleration of motion occurs may be represented by the equations (Ref. 1)

$$F_1 = f_o \frac{W}{2} - \frac{M_o}{w}, \text{ and}$$

$$F_2 = f_o \frac{W}{2} + \frac{M_o}{w} \qquad (5\text{-}18)$$

where

F_1 and F_2 = forces propelling each track, lb

f_o = unit resistance to motion (dimensionless) and equal to $\frac{R}{W}$, where R, in pounds, is the total motion resistance experienced by the vehicle (see Chapter 4)

W = gross vehicle weight, lb

w = center-to-center distance between the two tracks, in.

M_o = total moment of the resistance to rotation about the vehicle's center of gravity, in.-lb

The problem is to evaluate M_o. The moment of resistance is developed due to the frictional resistance to sliding between the track and soil, and the load distribution characteristics. A load which is concentrated near the point about which the track is turning will result in the lowest value of M_o. This may be demonstrated by considering the cases of uniform and triangular load distribution under a track (Figure 5-5(A) and (B)).

For uniform load distribution, the pressure in psi at any point under the track is

$$p = \frac{W}{2bl}$$

where

W = vehicle weight, lb

b = track width, in.

l = track length, in.

The force per unit area resisting the track turn is assumed to be μp, where μ is the coefficient of lateral friction of the track and ground. The resisting moment may then be computed by integration as

$$M_o = 4 \int_o^{\frac{l}{2}} \mu p \, bx \, dx = \frac{\mu W l}{4}$$

or from Figure 5-5(A) using resultant forces

$$M_o = 4\mu \frac{W}{4} \frac{l}{4} = \frac{\mu W l}{4}$$

Equations 5-18 may then be written as

$$F_1 = f_o \left(\frac{W}{2}\right) - \frac{\mu W l}{4w}, \text{ and}$$

$$F_2 = f_o \left(\frac{W}{2}\right) + \frac{\mu W l}{4w} \qquad (5\text{-}19)$$

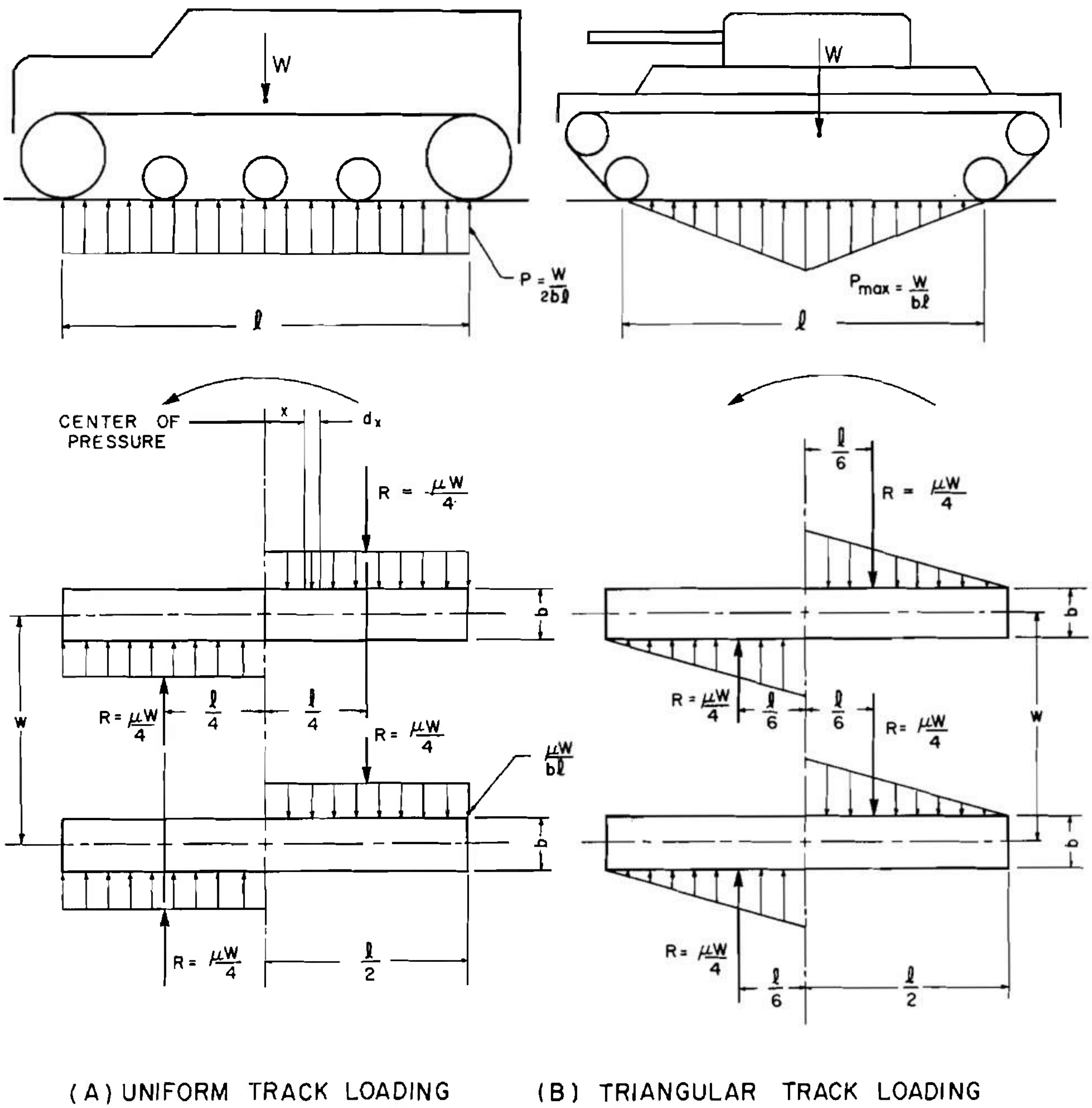


Figure 5-5. Forces Developing Moment of Resistance to Turning for Uniform and Triangular Track Loading

For a triangular load distribution as shown in Figure 5-5(B), the problem may be handled in the manner which follows. The maximum pressure under the center of the track is

$$p_{max} = \frac{W}{bl}$$

The pressure at distance x from the center is

$$p = \frac{l - 2x}{l}\, p_{max} = \frac{(l - 2x)\, W}{bl^2}$$

The moment of resistance against turning is then

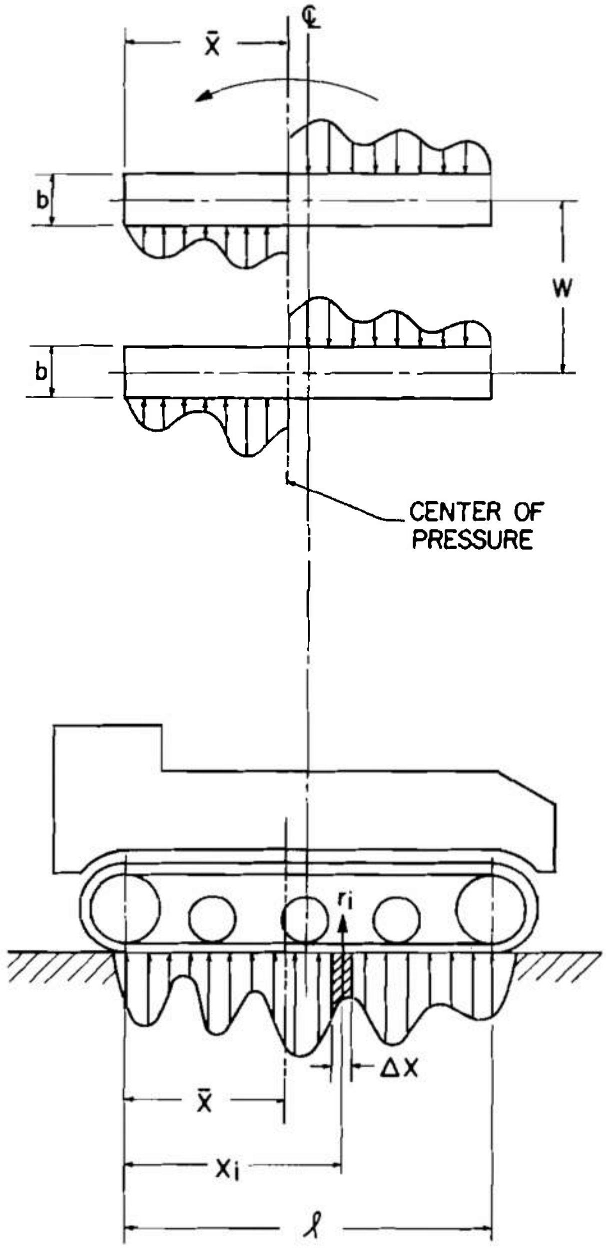


Figure 5-6. Forces Developing Moment of Resistance to Turning for Arbitrary Track Loading

$$M_o = 4 \int_o^{\frac{l}{2}} \mu p \; bx \; dx = \frac{\mu W l}{6}, \text{ or}$$

from Figure 5-5(B) using resultant forces

$$M_o = 4\mu \frac{W}{4} \frac{l}{6} = \frac{\mu W l}{6}$$

Equations 5-18 then may be expressed as

$$F_1 = f_o \left(\frac{W}{2}\right) - \frac{\mu W l}{6w}, \text{ and}$$

$$F_2 = f_o \left(\frac{W}{2}\right) + \frac{\mu W l}{6w} \qquad (5\text{-}20)$$

It is seen that the smaller moment of resistance to rotation occurs with the triangular load distribution where there is a greater concentration of the resisting forces about the turning point.

The resisting moment of any arbitrary load distribution such as Figure 5-6 may be computed using a numerical integration procedure. When only those forces considered in this discussion are acting, the axis of rotation passes through the center of the ground pressure. The track length may be divided into several increments n each of length Δx. The resultant load r_i on each increment may then be determined graphically from the load distribution diagram. The location of the center of the ground pressure from the end of the track, $\bar{x}$, may then be determined as

$$\bar{x} = \frac{2 \sum_{i=1}^{n} r_i x_i}{W}. \qquad (5\text{-}21)$$

The resisting moment to rotation may then be computed by the summation of the moments of the incremental resisting forces μr_i about the center of the ground pressure.

5-10.3 EFFECT OF OVERALL DIMENSIONS ON STEERABILITY

It was seen in the previous paragraph that the moment of resistance to rotation was reduced if the load distribution under the track resulted in a concentration of load near the point about which rotation occurred. This reduction in the resisting moment occurs even though the total ground load remains constant because the moment arm of the resisting forces is reduced. Obviously, this same effect is achieved if the length of the track is reduced. Hence, the long, narrow track form which is advantageous with respect to motion resistance, flotation, and slippage is detrimental with respect to steerability.

Vehicle width was also seen to enter into the turning Equations 5-18. It may be seen by inspection that the wider the vehicle, the smaller will be the influence of the resisting moment. Here again, the concept of the long, narrow vehicle, which is advantageous in most other respects, is detrimental to vehicle steerability.

The interrelationship of the vehicle overall dimensions on the steering characteristics has been qualitatively evaluated in terms of the length-to-width ratio and can be found in paragraph 5-4.5.

5-10.4 EFFECTS OF SLOPE OPERATION ON STEERING

5-10.4.1 Negotiating an Uphill Turn

When a vehicle turns on a slope, lateral forces must be taken into account in the equations describing the turning phenomenon. In negotiating an uphill turn, the track loadings orthogonal to the soil surface (Figure 5-7(A)) are

$$Q_1 = \frac{W}{2}\left(1 - \frac{2h}{w}\tan\alpha\right)\cos\alpha, \text{ and}$$

$$Q_2 = \frac{W}{2}\left(1 + \frac{2h}{w}\tan\alpha\right)\cos\alpha \qquad (5\text{-}22)$$

where W is the gross vehicle weight, w is the center-to-center distance between the two tracks, h is the height to vehicle center of gravity (measured perpendicular to the ground), and α is the angle of the slope. For the vehicle to be in equilibrium as it turns, the distribution of the frictional forces between the track and the ground must be such that they will balance the lateral force $W\sin\alpha$ which tends to move the vehicle down hill. Hence, the location of the center of rotation will no longer be at the center of the ground pressure. For the case of uniform track loading, the center of rotation will be forward of the center of ground pressure a distance u as shown in Figure 5-7. Equations 5-18 hold, but the limits of integration must be changed to accommodate the new center of rotation. The resisting moment may then be computed as:

$$M_o = \frac{(Q_1 + Q_2)\mu}{l}\left[\int_o^{\left(\frac{l}{2}+u\right)} x\,dx + \int_o^{-\left(\frac{l}{2}-u\right)} x\,dx\right] - Wu\sin\alpha$$

Integration yields

$$M_o = \frac{W\mu\cos\alpha}{4l}\,[l^2 + 4u^2] - W\,u\sin\alpha \qquad (5\text{-}23)$$

The center of rotation will be displaced from the center of ground pressure by the distance u which yields the minimum M_o. Hence, u may be evaluated by differentiating M_o with respect to u.

$$\frac{dM_o}{du} = \frac{2\,u\,W\mu\cos\alpha}{l} - W\sin\alpha = 0, \text{ and}$$

$$u = \frac{l\tan\alpha}{2\mu} \qquad (5\text{-}24)$$

Substituting Equation 5-24 into Equation 5-23,

$$M_o = \frac{Wl\mu\cos\alpha}{4}\left[1 - \left(\frac{\tan\alpha}{\mu}\right)^2\right] \qquad (5\text{-}25)$$

Thus it is seen that the slope decreases the resisting moment when the vehicle negotiates an uphill turn. The required propelling forces F_1 and F_2 (Eqs. 5-18) are

$$F_1 = \frac{f_o W\cos\alpha}{2}\left(1 - \frac{2h}{w}\tan\alpha\right) - \frac{Wl\mu\cos\alpha}{4w}\left[1 - \left(\frac{\tan\alpha}{\mu}\right)^2\right]$$

$$F_2 = \frac{f_o W\cos\alpha}{2}\left(1 + \frac{2h}{w}\tan\alpha\right) + \frac{Wl\mu\cos\alpha}{4w}\left[1 - \left(\frac{\tan\alpha}{\mu}\right)^2\right] \qquad (5\text{-}26)$$

Effects of load distributions other than uniform can be determined as outlined in paragraph 5-10.2 once the center of rotation is located.

5-10.4.2 Negotiating a Downhill Turn

The force analysis of a tracked vehicle negotiating a downhill turn is similar to that of the

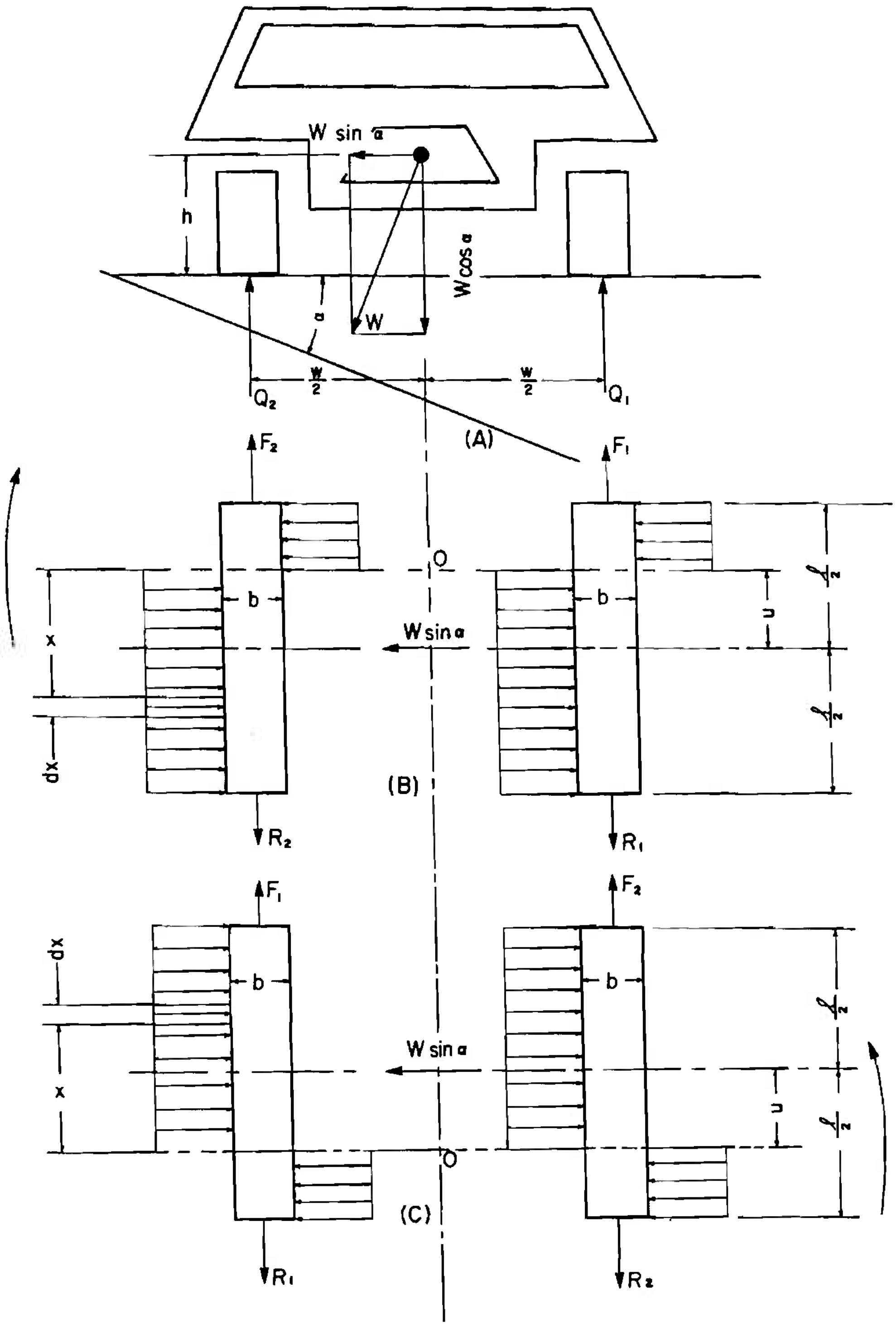


Figure 5-7. Effects of Slope Operation on Steering (A) Track Loading (B) Negotiating Uphill Turn (C) Negotiating Downhill Turn

vehicle negotiating an uphill turn discussed in the preceding paragraph. In this case, however, the lateral forces cause the center of rotation to move toward the rear of the vehicle, and the moment of resistance is greater for the downhill turn than for the uphill turn (Figure 5-7(C)). The required propelling forces are then:

$$F_1 = \frac{f_o W \cos\alpha}{2}\left(1 + \frac{2h}{w}\tan\alpha\right) - \frac{Wl\mu\cos\alpha}{4w}\left[1 + \left(\frac{\tan\alpha}{\mu}\right)^2\right]$$

$$F_2 = \frac{f_o W \cos\alpha}{2}\left(1 - \frac{2h}{w}\tan\alpha\right) + \frac{Wl\mu\cos\alpha}{4w}\left[1 + \left(\frac{\tan\alpha}{\mu}\right)^2\right] \quad (5\text{-}27)$$

5-10.5 EFFECTS OF CENTRIFUGAL FORCE ON STEERING

The primary effect of centrifugal force is the introduction of a lateral force F_c away from the direction of turn (Figure 5-8). For a turning radius r_o which is large compared to the length l of the track ground-contact area

$$\cos\beta \approx 1$$
$$\sin\beta \approx 0$$
$$r_o \approx \text{constant}$$

Then

$$F_c\cos\beta \approx F_c = \frac{Wv^2}{gr_o}$$

where, if F_c is in pounds

W = total vehicle weight, lb
v = vehicle speed, fps
g = gravitational constant, fps^2
r_o = turning radius, ft

The track loadings Q_1 and Q_2 are

$$Q_1 = \frac{W}{2} - F_c\frac{h}{w}, \text{ and}$$
$$Q_2 = \frac{W}{2} + F_c\frac{h}{w} \quad (5\text{-}28)$$

As was the case for the uphill turn, the center of rotation shifts forward of the center of ground pressure. If the ground pressure is uniform beneath the tracks, the moment of resistance to rotation is determined as

$$M_o = \frac{(Q_1+Q_2)\mu}{l}\left[\int_0^{\frac{l}{2}+u} x\,dx + \int_0^{-\left(\frac{l}{2}-u\right)} x\,dx\right] - F_c u, \text{ and}$$
$$M_o = \frac{W\mu}{4l}(l^2 + 4u^2) - \frac{Wv^2u}{gr_o} \quad (5\text{-}29)$$

The displacement of the center of rotation is determined by the same procedure as previously given

$$u = \frac{lv^2}{2\mu g r_o} \quad (5\text{-}30)$$

The required track propelling forces may then be determined as

$$F_1 = \frac{f_oW}{2} - \frac{hWv^2}{wgr_o} - \frac{W\mu l}{4w}\left[1 - \left(\frac{v^2}{g\mu r_o}\right)^2\right], \text{ and}$$
$$F_2 = \frac{f_oW}{2} + \frac{hWv^2}{wgr_o} + \frac{W\mu l}{4w}\left[1 - \left(\frac{v^2}{g\mu r_o}\right)^2\right] \quad (5\text{-}31)$$

When the turning radius r_o is not large compared to the track length l, the value of $\sin\beta$ becomes significant and the longitudinal component of the force cannot be neglected. This force disturbs the pressure distribution under the track. It may be assumed that the effect is linear along the length of the track, decreasing the ground pressure in front of the center of rotation and increasing the ground pressure behind the point of rotation. Once the ground pressure distribution is determined, the resisting moment can be determined by the previously outlined procedures for nonuniform pressure distribution. In general, however, it may be considered that the Equations 5-31 give an adequate approximation and the degree of exactness of the analysis does not warrant the additional labor involved in considering the longitudinal component $F_c \sin\beta$.

5-10.6 EFFECTS OF A TOWED LOAD ON STEERING (Ref. 1)

A towed load introduces both longitudinal and lateral forces which affect the steering of the vehicle (Figure 5-9). If it is assumed that the towing connection is so located that it enables the tow to be made without disturbing the pressure distribution under the tracks, the moment of resistance (in in.-lb) against turning for uniform load distribution is computed as

$$M_o = \frac{W\mu}{l}\left[\int_0^{\left(\frac{l}{2}+u\right)} x\,dx + \int_0^{-\left(\frac{l}{2}-u\right)} x\,dx\right] + F_h\,(L-u)\sin\alpha$$

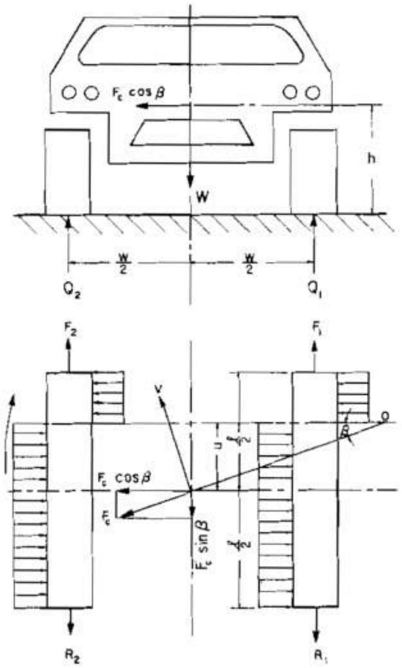


Figure 5-8. Effects of Centrifugal Force on Steering

from which

$$M_o = \frac{W\mu}{4l}(l^2 + 4u^2) + F_h\,(L-u)\sin\alpha \quad (5\text{-}32)$$

where

W = total vehicle weight, lb
L = distance from vehicle c.g. to tow connection, in.
F_h = towed load, lb
u = displacement of center of rotation, in.
α = angle between longitudinal centerlines of towed and towing vehicle, deg

Differentiating M_o with respect to u and equating the resulting equation to zero yields

$$u = \frac{lF_h \sin\alpha}{2W\mu} \quad (5\text{-}33)$$

Both M_o and u increase with an increase in α. Hence, a gradual turn with a towed load is more desirable. If α increases to such an extent that $u > \frac{l}{2}$, the turning point is behind the track and the vehicle will skid.

When Equations 5-32 and 5-33 are substituted into Equations 5-18 and the component $F_h \cos\alpha$ is added, the driving forces of the tracks are:

$$F_1 = \frac{f_oW}{2} + \frac{F_h\cos\alpha}{2} - \frac{W\mu l}{4w} - \frac{F_h\sin\alpha}{w}\left[L - \frac{lF_h\sin\alpha}{4W\mu}\right], \text{ and}$$

$$F_2 = \frac{f_oW}{2} + \frac{F_h\cos\alpha}{2} + \frac{W\mu l}{4w} + \frac{F_h\sin\alpha}{w}\left[L - \frac{lF_h\sin\alpha}{4W\mu}\right] \quad (5\text{-}34)$$

When the tow connection is such that the load distribution on the track is disturbed, the problem can be handled using the procedures previously outlined if the new load distribution can be determined.

5-10.7 EFFECTS OF LONGITUDINAL TRACK SLIPPAGE ON STEERING

In the previous analysis the forces F_1 and F_2 were determined assuming no longitudinal slip. The more general case is that in which both tracks slip longitudinally—this condition of turning is discussed in Ref. 6. Slip is defined as rotation around the instantaneous centers O_1 and O_2 (Figure 5-10) which are located on a line perpendicular to the track axis, generally to the side of the track on

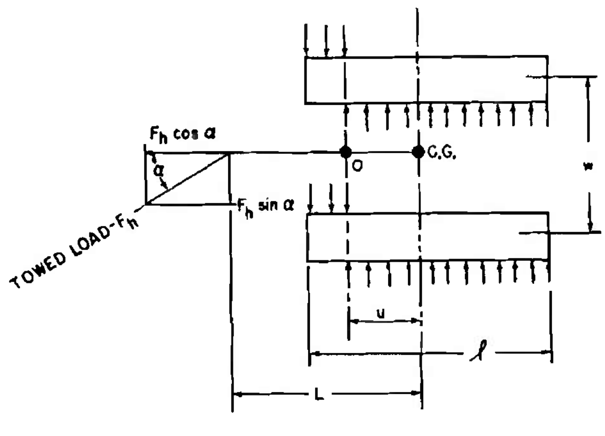


Figure 5-9. Effects of Towed Load on Steering

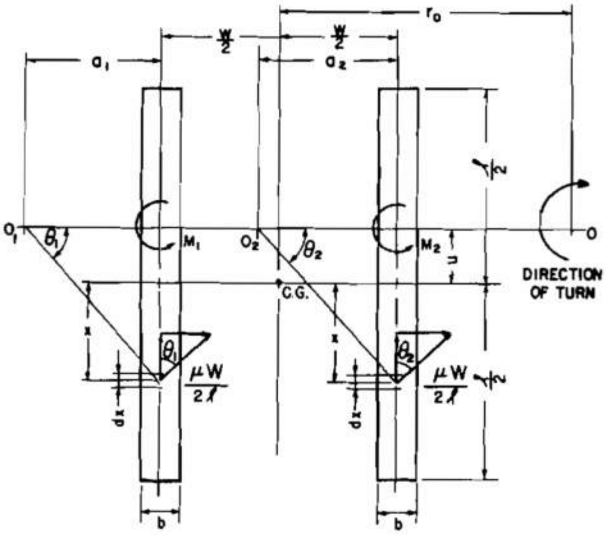


Figure 5-10. Effects of Longitudinal Track Slippage on Steering

the outside of the turn. The radii of track slip a_1 and a_2 are positive when O_1 and O_2 are on the outside of the turn, and negative when on the inside. The radius of curvature is determined as

$$r_o = \frac{\frac{w}{2}\left(\frac{\omega_1}{\omega_2} + 1\right) + a_1 - a_2\frac{\omega_1}{\omega_2}}{\frac{\omega_1}{\omega_2} - 1} \tag{5-35}$$

where

ω_1 and ω_2 = angular velocities of the sprockets driving the tracks, rad/sec

w = center-to-center distance between the two tracks, in. (if a_1, a_2, and r_o are in inches)

The forces F_1 and F_2 acting on the tracks due to slip are determined by integration. Figure 5-10 illustrates the case of a uniformly loaded track. The track load per unit of length is $W/2l$; and, for a smooth track, it is assumed that the coefficient of track ground friction μ is the same in the longitudinal direction as in the lateral direction. For this case,

$$F_i = \frac{\mu W}{2l}\int_{\left(u-\frac{l}{2}\right)}^{\left(u+\frac{l}{2}\right)} \cos\theta_i \, dx \text{ and}$$

$$T_i = \frac{\mu W}{2x}\int_{\left(u-\frac{l}{2}\right)}^{\left(u+\frac{l}{2}\right)} \sin\theta_i \, dx$$

where $\cos\theta_i = \frac{a_i}{\sqrt{a_i^2 + x^2}}$

$$\sin\theta_i = \frac{x}{\sqrt{a_i^2 + x^2}} \text{ and}$$

$$i = 1, 2, \text{etc.}$$

Integration yields

$$F_i = \frac{\mu W a_i}{2l}\, ln\left[\frac{2u + l + \sqrt{(2u+l)^2 + 4a_i^2}}{2u - l + \sqrt{(2u-l)^2 + 4a_i^2}}\right] \tag{5-36}$$

$$T_i = \frac{\mu W}{4l}\left[\sqrt{(2u+l)^2 + 4a^2} - \sqrt{(2u-l)^2 + 4a_i^2}\right] \tag{5-37}$$

The moments of forces about the point Q are

$$M_i = \frac{\mu W}{2l}\int_{\left(u-\frac{l}{2}\right)}^{\left(u+\frac{l}{2}\right)} x \sin\theta_i \, dx.$$

Integration yields

$$M_i = \frac{\mu W}{16l}\left[(2u+l)\sqrt{(2u+l)^2 + 4a_i^2} - (2u-l)\sqrt{(2u-l)^2 + 4a_i^2}\right] - \frac{a_i}{2}F_i \tag{5-38}$$

Equations 5-36 to 5-38 were solved in Ref. 6 for various values of $2a_i/l$ and $2u/l$ (Figures 5-11, 5-12, and 5-13). The solution to any particular problem is obtained with the aid of the graphs by trial and error selection of the parameter a_i to satisfy these equations.

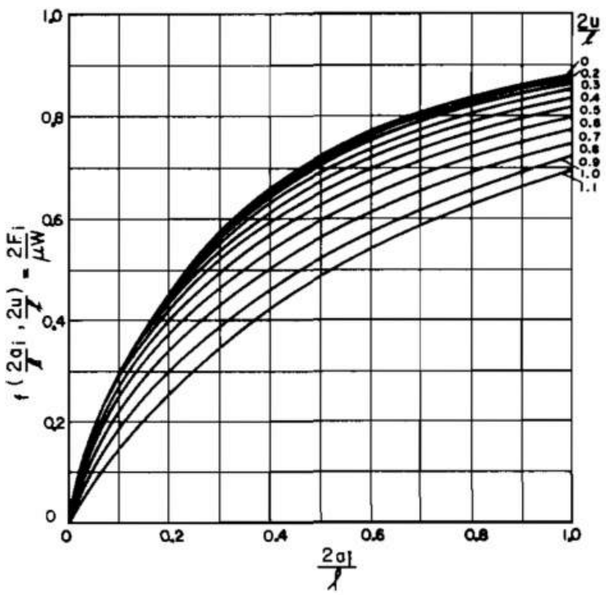


Figure 5-11. Longitudinal Force Function (Ref. 6)

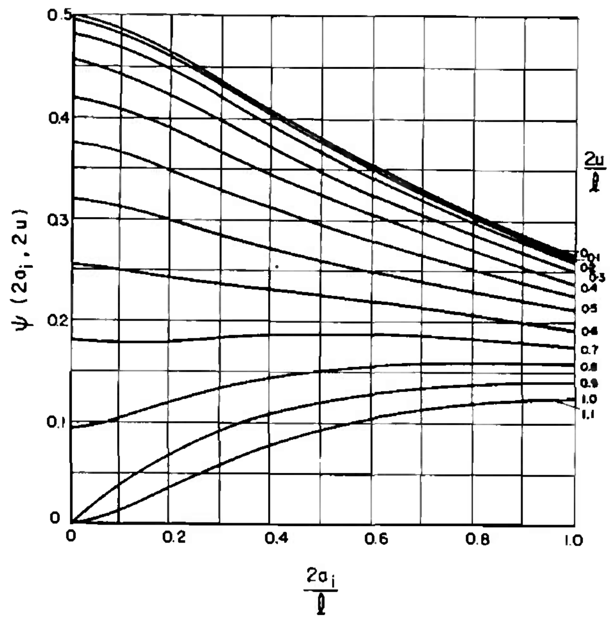


Figure 5-13. Moment Function (Ref. 6)

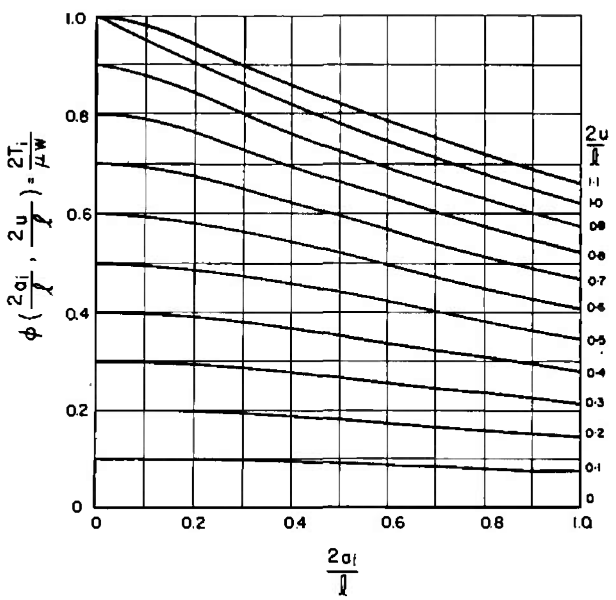


Figure 5-12. Lateral Force Function (Ref. 6)

SECTION III HARD SOIL PERFORMANCE OF VEHICLES

5-11 EFFECTS OF VEHICLE SIZE AND WEIGHT

On hard soil, vehicle sinkage is negligible and the primary problems related to vehicle size and weight are their effects on vehicle steering. The discussion of these problems presented in Sections I and II of this chapter, with respect to soft soils, is applicable here—with the exception of the resistance to motion caused by vehicle sinkage. In general, the greater the vehicle length-to-width ratio, the less the steerability of the vehicle. Similarly, the heavier the vehicle, the greater will be the frictional sliding forces and the greater the resistance to turning.

5-12 LIMITS TO VEHICLE SPEED

The majority of contemporary cross-country vehicles are equipped with power trains capable of developing maximum speeds in excess of 35 miles per hour. Yet these vehicles are able to operate off the road on typical hard rough terrain at speeds of only 5 to 10 miles per hour. This apparent contradiction results because the sustainable speed of off-road locomotion is not governed by power alone, but also by the geometry and dynamic properties of the vehicle. As a vehicle travels over rough tertain the interaction between the vehicle and terrain induces pitching, bouncing, and vibrations in the vehicle. With current vehicles, these induced responses may become of such magnitude that they prohibit control and even surpass the endurance limit of the operator and the fatigue strength of some mechanical components.

Although the disadvantages which stem from such a situation are serious and have been apparent since the inception of the conception of off-road locomotion, study of the problem of rough-terrain vehicle interaction has only recently been undertaken. Research in this area comprises principles of the mechanics of vehicle vibrations and mathematical formulation of the typical unevenness of terrain over which vehicles are expected to operate. Only by combining the results of studies in both these areas can the optimum solution to the problem of locomotion over rough ground be obtained.

Human factors considerations which limit maximum speeds of vehicles are thoroughly discussed in Section II of Chapter 2.

Environmental factors and their effects upon vehicle speed and control are discussed in Section III of Chapter 2.

5-13 EFFECTS OF SUSPENSION CHARACTERISTICS

5-13.1 SPRING RATE AND DAMPING

To determine the effects of suspension characteristics on vehicle speed over hard ground, it is necessary to study the interaction of the vehicle and the terrain surface as a function of vehicle speed. The problem comprises the mechanics of vehicle vibrations and mathematical analysis of terrain geometry. Methods of analysis of vibrations problems have been under study for a considerable time and the analytical techniques most applicable to the problems of vehicle vibrations are discussed in Chapters 9, 10, and 11. The development of analytical methods to describe, mathematically, typical geometry of the terrain surface is of recent origin. Discussion of some of the promising

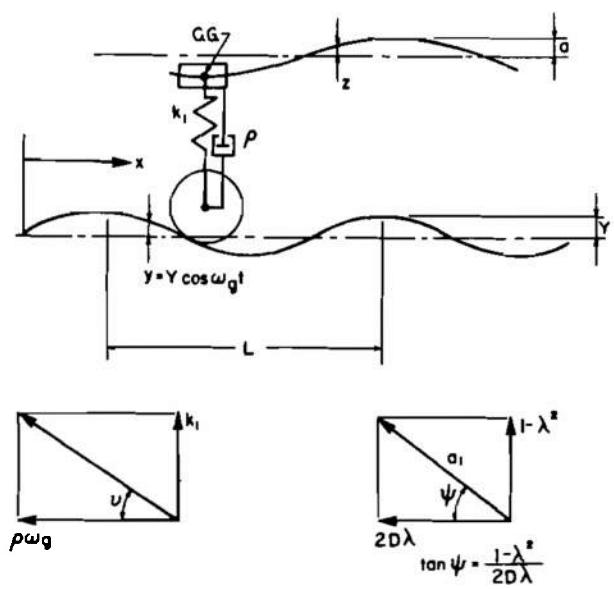


Figure 5-14. Forced, Damped Vibrations

methods of approach is presented in Chapters 4 and 7.

To illustrate how an interaction study might proceed, consider a wavy ground profile (Figure 5-14) described by the formula

$$y = Y \cos \frac{2\pi x}{L} \tag{5-39}$$

where

$y =$ vertical displacement of the ground, in.
$Y =$ amplitude of the vertical displacement, in.
$L =$ wavelength, in.
$x =$ distance along the ground from the zero reference point, in.

If a vehicle travels over this profile at a constant horizontal velocity, vertical displacement may be expressed as a function of time by the substitution $x = vt$, where v is the vehicle speed. Equation 5-39 may then be written as

$$y = Y \cos \omega_g t \tag{5-40}$$

where

$\omega_g = \frac{2\pi v}{L}$, the frequency of the wave component.

On the assumption that a vehicle body may be represented by the system shown on Figure 5-14, the equation of motion of the vehicle body is

$$m\left(\frac{d^2z}{dt^2}\right) + \rho\left(\frac{d}{dt}\right)(z - Y\cos\omega_g t) + k_1(z - Y\cos\omega_g t) = 0$$

$$m\left(\frac{d^2z}{dt^2}\right) + \rho\left(\frac{dz}{dt}\right) + k_1 z = Y(k_1\cos\omega_g t - \rho\omega_g \sin\omega_g t) \tag{5-41}$$

where

$m =$ mass of the vehicle body, in lb-sec^2/in.
$\rho =$ damping factor, lb-sec/in.
$k_1 =$ spring rate, lb/in.
$z =$ vertical displacement of C.G. of the mass, in.

The particular solution of Equation 5-41 is (Ref. 1)

$$z = a\cos(\omega_g t - \psi), \tag{5-42}$$

where a is the amplitude of the forced vibration of mass m and ψ is the phase angle.

Differentiating Equation 5-42 twice yields:

$$\frac{dz}{dt} = -a\,\omega_g \sin(\omega_g t - \psi)$$

and

$$\frac{d^2z}{dt^2} = -a\omega_g^2 \cos(\omega_g t - \psi). \tag{5-43}$$

By addition of the vectors k_1 and $\rho\omega_g$ (using phase angle ν, Figure 5-14) the difference $(k_1 \cos \omega_g t - \rho\omega_g \sin \omega_g t)$ in Equation 5-41 may be expressed as

$$(k_1\cos\omega_g t - \rho\omega_g\sin\omega_g t) = -\sqrt{k_1^2 + \rho^2\omega_g^2}\,\sin(\omega_g t - \nu). \tag{5-44}$$

When Equations 5-41, 5-42, and 5-43 are combined, Equation 5-44 yields

$$\left[(1 - \lambda^2)\cos(\omega_g t - \psi) - 2D\lambda\sin(\omega_g t - \psi)\right] + \left[\frac{Y}{a}\sqrt{1 + (2D\lambda)^2}\,\sin(\omega_g t - \nu)\right] = 0 \tag{5-45}$$

where

$$\omega_n^2 = \frac{k_1}{m}$$

and

$\omega_n =$ the natural frequency of the system, rad/sec

$$\left.\begin{aligned} \lambda &= \frac{\omega_g}{\omega_n} \\ D &= \frac{\rho}{2m\omega_m} \end{aligned}\right\}, \text{ dimensionless parameters}$$

The first term of Equation 5-45 describes the free damped vibrations of the system, while the last term describes the forced damped vibrations. The amplitude a_1 of free vibrations is (Ref. 1)

$$a_1 = \sqrt{(1 - \lambda^2)^2 + (2D\lambda)^2} \tag{5-46}$$

and Equation 5-45 can be rewritten as

$$-\sqrt{(1-\lambda^2)^2 + (2D\lambda)^2}\,\sin(\omega_g t - 2\psi)$$
$$= -\frac{Y}{a}\sqrt{1 + (2D\lambda)^2}\,\sin(\omega_g t - \nu). \tag{5-47}$$

The response factor of forced vibrations is then

$$\frac{a}{Y} = \frac{\sqrt{1 + (2D\lambda)^2}}{\sqrt{(1-\lambda^2)^2 + (2D\lambda)^2}} \qquad (5\text{-}48)$$

defined in terms of the natural frequency of the system, ω_n; the ratio of the frequency of the ground wave to the natural frequency λ and the system damping parameter D.

Example:

Consider a wheeled vehicle traveling at 10 miles per hour over the cross-ties of a railroad track spaced on 2-ft centers and protruding 1½ in. above the ballast. The weight of the sprung mass supported by the front axle is 5,600 lb; the combined spring rate of the two front springs is 1600 lb/inch; and the combined damping factor is 300 lb-sec/inch. From Figure 5-14 and the preceding discussion, the amplitude a of the front end vibration can be calculated as follows:

$$Y = 1.5 \text{ inch}$$

$$v = 10 \text{ mph} \times 5280 \times \frac{12}{3600} = 176 \text{ in./sec}$$

$$L = 2 \text{ ft} \times 12 = 24 \text{ in.}$$

$$\omega_g = \frac{2\pi v}{L} = \frac{2\pi \times 176}{24} = 46.1 \text{ rad/sec}$$

$$m = \frac{5600}{32.2 \times 12} \times 12 = 14.5 \text{ lb-sec}^2\text{/in.}$$

$$k_1 = 1600 \text{ lb/in.}$$

$$\omega_n = \sqrt{\frac{k_1}{m}} = \sqrt{\frac{1600}{14.5}} = 10.5 \text{ rad/sec}$$

$$\lambda = \frac{\omega_g}{\omega_n} = \frac{46.1}{10.5} = 4.4$$

$$D = \frac{\rho}{2m\omega_n} = \frac{300}{2 \times 14.5 \times 10.5} = 0.985$$

$$\frac{a}{Y} = \frac{\sqrt{1 + (2D\lambda)^2}}{\sqrt{(1-\lambda^2)^2 + (2D\lambda)^2}}$$

$$= \frac{\sqrt{1 + (2 \times 0.985 \times 4.4)^2}}{\sqrt{(1 - 4.4^2)^2 + (2 \times 0.985 \times 4.4)^2}}$$

$$= 0.43$$

$$a = 0.43Y = 0.43 \times 1.5 = 0.65 \text{ in.}$$

5-13.2 WHEEL TRAVEL

In the analysis presented in the previous paragraph, the relative movement between the wheel and the mass representing the vehicle body was described by the term $(z - Y \cos \omega_g t)$ in Equation 5-41. No consideration was given to the existence of a limiting magnitude of this relative movement. In actuality, the relative motion will be restricted by stop limits on wheel travel. When the wheel engages a stop, the movement of the mass abruptly ceases to be described by the function

$$z = a \cos(\omega_g t - \psi)$$

and the mass is subjected to an abrupt change in acceleration. As a result, a jolt is delivered to the driver of the vehicle and some vehicle components are subjected to high stress concentrations.

Thus, it is desirable to consider the interrelationships of vehicle speed, damping, natural frequency, and the geometry of the ground surface to arrive at a design which restricts the relative movement between the sprung and unsprung masses within the bounds of the wheel travel.

5-13.3 ACTIVE SUSPENSION SYSTEMS

In the discussion of the suspension system thus far, spring rate and damping have been characterized by values k_1 and ρ. For a particular speed over a particular terrain geometry, values can be assigned to k_1 and ρ to result in an optimum suspension design. However, if the vehicle speed is varied without a change in the terrain geometry, or if the terrain geometry varies without a change in speed, the assigned k_1 and ρ will no longer result in optimum suspension characteristics. A system designed with spring rate and damping characterized by constant values of k_1 and ρ is a passive suspension system. It is obviously desirable to have k_1 and ρ vary to optimize the suspension system to the operational environment of the vehicle.

One method of varying k_1 and ρ to optimize the suspension system to the operational environment is to design them to vary as the suspension system responds to disturbances of the ground surface. For example, as the system of Figure 5-14 suddenly encounters a downward sloping contour, the spring rate tends to cause the sprung mass and the wheel to separate. The velocity of this separation depends upon the natural frequency of the spring system which is reflected in the spring rate (or stiffness) of the system. A system with greater spring

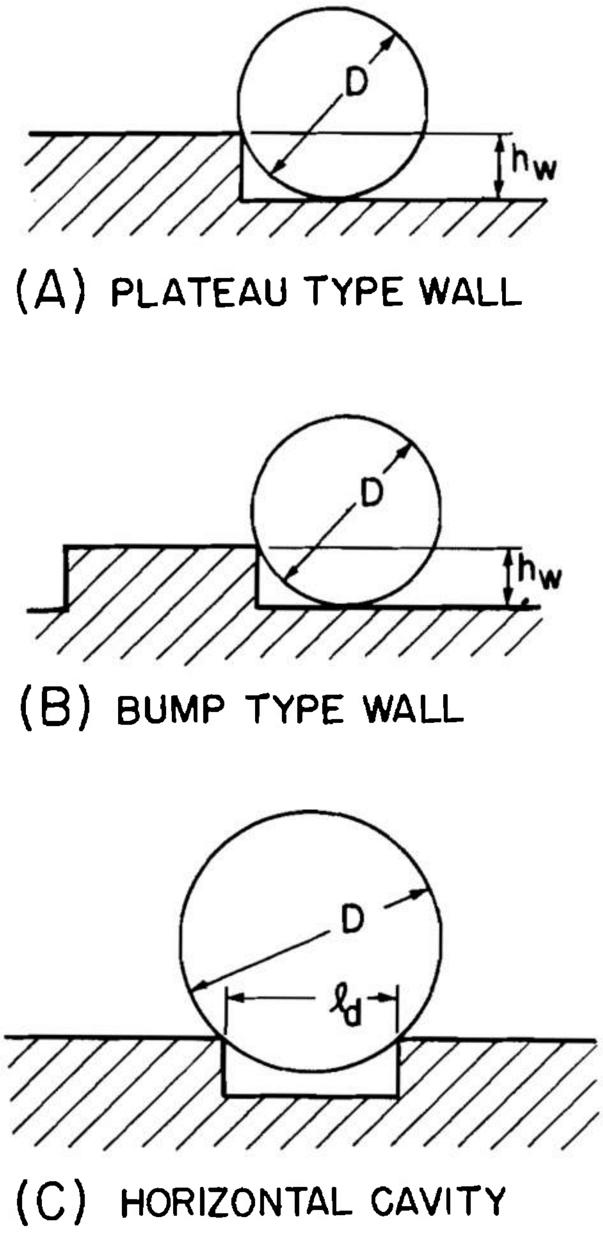


Figure 5-15. Typical Terrain Obstacles

rate has a higher natural frequency and will, therefore, exhibit a quicker response than a system of low spring rate, where the separating velocity will be relatively slow, and the sprung mass will tend to follow the ground profile. Since, in a high spring rate system the separating velocity will be high and the response of the sprung mass will lag the unsprung mass, the former will experience reduced vertical accelerations and displacements. Thus, when terrain-vehicle conditions are such as to cause the sprung and unsprung masses to separate, it is more desirable to have a stiff suspension.

Initially, it is also desirable that damping be small so that the relative motion between the two mass systems is not impeded. However, as the magnitude of this displacement approaches the structural limits of the design, it is necessary to reduce the velocity of the displacement by increased damping and by reducing the rate of the springs.

As the ground slope suddenly changes from a horizontal to an upward slope, the two mass systems will move toward each other. Under these conditions it is more desirable to have a soft spring and low damping so that the wheel may readily follow up the slope with minimum disturbance to the sprung mass. A suspension system with the capability of varying its stiffness and damping as the system responds to terrain irregularities is a semi-active system. Because the system characteristics change as the terrain irregularities are encountered, a lag in optimizing the system to the environment is inherent.

Systems which precondition the suspension system to the terrain disturbances to be encountered are active suspension systems. Preconditioning may be accomplished by means of feelers or sensors ahead of the wheel, which sense the disturbance to be encountered and adjust pertinent suspension elements accordingly. Another method of preconditioning is to program desired suspension characteristics for typical terrain contours into the suspension system. The response history can then be used to predict the probable nature of the ground surface to be encountered next, and suspension characteristics can be adjusted for the expected condition. Studies of typical terrain contours, discussed in Chapter 7, are encouraging to this latter method in that results indicate ground geometry conforms to general patterns.

5-14 OBSTACLE PERFORMANCE

5-14.1 GENERAL DISCUSSION

Overcoming obstacles is mainly a "go" or "no-go" consideration, and speed of traverse and vibrations encountered are of secondary importance.

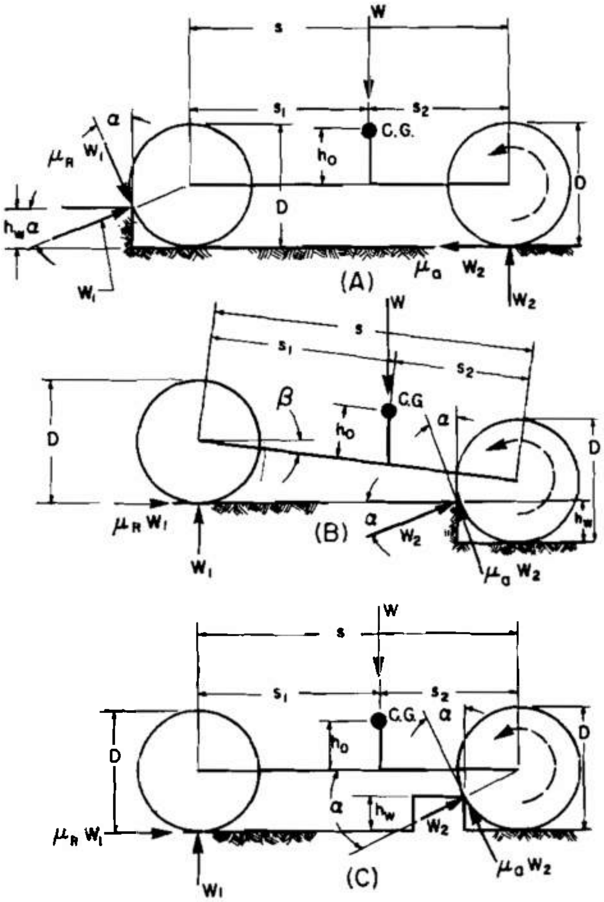


Figure 5-16. Vehicle With Rear Axle Drive Encountering Vertical Obstacles (A) Front wheel encountering vertical obstacle (B) Rear wheel encountering plateau-type wall (C) Rear wheel encountering a vertical bump type wall

This section is concerned only with obstacles presented by the geometry of the terrain; namely, vertical walls and horizontal cavities (Figure 5-15). These two types of obstacles, considered as formed by the right angle intersection of plane surfaces, are generally accepted as the measure of obstacle performance and automatically postulate the condition of undeformable hard ground. It has been shown (Ref. 7) that the obstacle performance of a two-axle vehicle is determined almost exclusively by two factors of its geometry; namely, the ratio of the wheelbase length to the diameter of the wheels, s/D, and the weight distribution over the two axles, i.e., the s_1/s ratio, where s_1 is the longitudinal distance from the front axle to the vehicle center of gravity. A summary of this study is given in the paragraphs which follow.

5-14.2 VERTICAL WALLS

5-14.2.1 Rear Axle Drive Only

Figure 5-16(A) shows a free body diagram of a two-axle, rear-axle driven vehicle encountering a vertical wall obstacle with its front wheels. A rigid suspension is assumed; kinetic forces are neglected since the velocity of the vehicle will be very low. A summation of forces in the horizontal and vertical directions, and moments about the front axle yield the following equations:

$$W_1 \cos\alpha + \mu_R W_1 \sin\alpha - \mu_a W_2 = 0$$
$$W_1 \sin\alpha + W_2 - \mu_R W_1 \cos\alpha - W = 0 \quad (5\text{-}49)$$
$$\mu_R W_1 \frac{D}{2} + W_2 s - W s_1 - \mu_a W_2 \frac{D}{2} = 0$$

where

μ_R = coefficient of rolling resistance such that the total rolling resistance, $R = \mu_R W$

μ_a = traction coefficient such that

$$\mu_a = \frac{\text{gross tractive effort, } H\text{-total rolling resistance, } R}{\text{gross vehicle weight, } W}$$

and the other symbols are as indicated in Figure 5-16.

The solution of Equation 5-49 results in the following dimensionless relation:

$$\left[\frac{\mu_a + \mu_R}{\mu_a}\left(\frac{s_1}{s}\right) - \frac{\mu_R}{\mu_a} + \frac{\mu_R D}{2s}\right]\sin\alpha - \left[\frac{1}{\mu_a} - \frac{1-\mu_R\mu_a}{\mu_a}\left(\frac{s_1}{s}\right) - \frac{D}{2s}\right]\cos\alpha = \frac{\mu_R D}{2s} \quad (5\text{-}50)$$

From the geometry of the problem

$$\sin\alpha = 1 - \frac{2h_w}{D} \text{ and}$$
$$\cos\alpha = \frac{2\sqrt{Dh_w - h_w^2}}{D} \quad (5\text{-}51)$$

Thus, Equation 5-50 gives the relationship between the ratio of vertical obstacle height to wheel diameter, h_w/D, and the geometrical parameters of the vehicle: s_1/s and D/s. The explicit solution of this equation for h_w/D is extremely unwieldy; however, if it is assumed that μ_R is very small* the equation can be readily solved to give

*μ_R varies from about 0.01 to 0.15 (concrete to medium-hard soils) to about 0.15 to 0.35 (for sands).

$$\left(\frac{h_w}{D}\right)_{\substack{front\\wheel}} = \frac{1}{2}\left\{ 1 - \frac{1}{\sqrt{1+\mu_a^2\left[\dfrac{\frac{s_1}{s}}{1-\frac{s_1}{s}-\frac{\mu_a D}{2s}}\right]^2}} \right\} \tag{5-52}$$

A few observations of interest that can be made from Equation 5-52 are that, for a given value μ_a, a relatively small increase in s_1/s will bring about a substantial increase of h_w/D; whereas increasing the D/s ratio will have a comparatively minor effect. Thus, the front wheel will be able to surmount higher obstacles as the vehicle center of gravity is shifted toward the rear (s_1/s made larger) and as the s/D ratio is decreased. In reducing the s/D ratio, it is much more effective to do so by decreasing the wheel base rather than by increasing the wheel diameter.

When considering the conditions relating to the rear wheels encountering the vertical obstacle (only rear axle driven), two cases exist as shown in Figure 5-16(B) and (C). Since the center of gravity of most vehicles is low (h_o small), the corresponding forces in each case are substantially the same; but in climbing a plateau-type wall (Figure 5-16(B)), the moment arm of the $\mu_R W_1$ force with respect to the driving axle is $D - h_w$. In the case of the bump-type wall (Figure 5-16(C)), the corresponding moment arm is D. Thus, the resisting moment acting on the front wheels is greater in the case of the bump-type wall, Figure 5-16(C), making this the critical case. The following equations of equilibrium may be written:

$$\left.\begin{aligned} &\mu_R W_1 + W_2\cos\alpha - \mu_a W_2 \sin\alpha = 0\\ &W_1 + W_2\sin\alpha + \mu_a W_2\cos\alpha - W = 0\\ &\mu_a W_2\frac{D}{2} + W_1 s - W s_2 - \mu_R W_1\frac{D}{2} = 0\end{aligned}\right\} \tag{5-53}$$

The solution of Equations 5-53 results in the following dimensionless relationships:

$$\left\{\begin{aligned} &\left[\frac{1-\mu_a\mu_R}{\mu_a}\left(\frac{s_1}{s}\right)+\mu_R-\frac{\mu_R}{\mu_a}\right]\cos\alpha\\ &-\left[\frac{\mu_a+\mu_R}{\mu_a}\left(\frac{s_1}{s}\right)-\frac{\mu_R}{\mu_a}-\frac{\mu_R D}{2s}\right]\sin\alpha\end{aligned}\right\} \tag{5-54}$$

If it is again assumed that μ_R is negligibly small, Equation 5-54 reduces to

$$\sin\alpha = \frac{\cos\alpha}{\mu_a}$$

Again applying the geometric relationships of Equations 5-51 results in the following solution for Equation 5-54:

$$\left(\frac{h_w}{D}\right)_{rear\ wheel} = \frac{1}{2}\left(1 - \frac{1}{\sqrt{1+\mu_a^2}}\right) \tag{5-55}$$

The absence of s and s_1 terms in Equation 5-55 indicates that the performance of the rear wheel is independent of the structural geometry of the vehicle. A plot of Equations 5-52 and 5-55 for all values of μ_a shows that the rear wheel limits the obstacle performance of a vehicle.

5-14.2.2 Front and Rear Axle Drive

The case of both axles driving (the so-called 4×4 vehicle) is shown in Figure 5-17. A comparison of the forces shown in Figures 5-17(A) and 5-16(A) shows that the only significant difference is that the resisting force $\mu_R W_1$ of Figure 5-16(A) is replaced by a propelling force $\mu_a W_1$. Thus, by replacing μ_R in Equation 5-50 with $-\mu_a$, the following dimensionless relationship is obtained for the front wheel:

$$\left[\frac{1}{\mu_a} - \frac{1+\mu_a^2}{\mu_a}\left(\frac{s_1}{s}\right) - \frac{D}{2s}\right]\cos\alpha - \left[1 - \frac{\mu_a D}{2s}\right]\sin\alpha = \frac{\mu_a D}{2s} \tag{5-56}$$

where, again, the relationships of Equations 5-51 apply. When Equation 5-56 is plotted in terms of h_w/D, it is seen that h_w/D decreases somewhat as the s/D ratio increases, and increases considerably as s_1/s increases (center of gravity shifts toward rear axle). Under certain conditions of μ_a and weight distribution it is possible for the front wheel to climb vertical obstacles that are higher than the radius of the wheel. Thus, on a two-axle, all wheel drive vehicle, the front wheels perform best with a

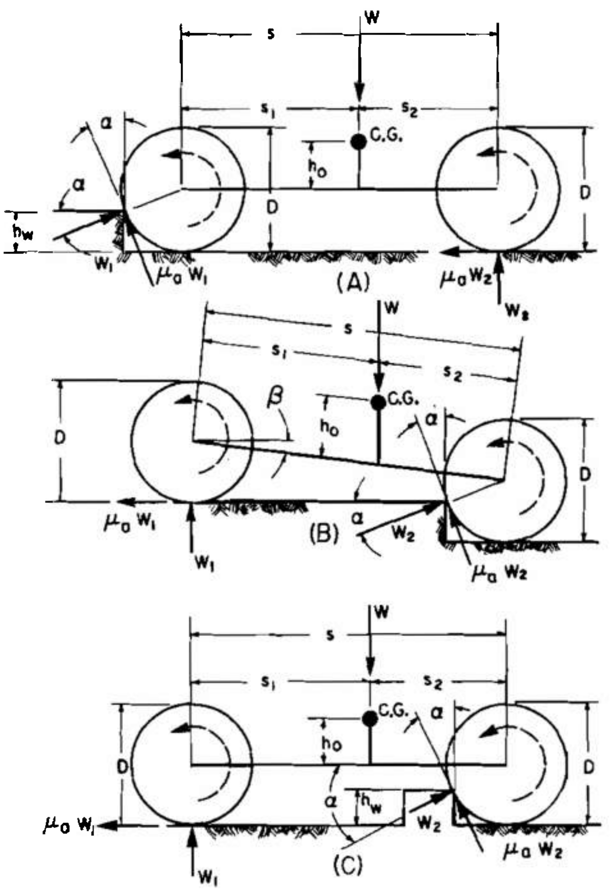


Figure 5-17. Vehicle with Two Axle Drive Encountering Vertical Obstacles (A) Front wheels encountering vertical obstacle (B) Rear wheels encountering plateau-type wall (C) Rear wheels encountering a vertical bump type wall

short wheelbase and the center of gravity toward the rear.

Consideration of the rear wheels again reveals two conditions as shown in Figure 5-17(B) and (C) but, unlike the rear-wheel drive, the $\mu_a W_1$ force is a propelling force. The moment arm of this force about the rear axle is less by the distance h_w in the case involving the plateau-type wall of Figure 5-17(B). This, then, is the limiting condition for the rear wheels. Application of the same procedures used in obtaining Equations 5-50 and 5-54 yields tbe following:

$$W_1 + W_2 \sin\alpha + \mu_a W_2 \cos\alpha - W = 0$$

$$-\mu_a W_1 + W_2 \cos\alpha - \mu_a W_2 \sin\alpha = 0 \qquad (5\text{-}57)$$

$$-W_1 s\cos\beta - \mu_a W_1\left(\frac{D}{2} - h_w\right) - \mu_a W_2\left(\frac{D}{2}\right) + W(s_2\cos\beta - h_o\sin\beta) = 0$$

where sin α and cos α are as before (Eq. 5-51) and

$$\left.\begin{aligned}\sin\beta &= \frac{h_w}{s}\\ \cos\beta &= \sqrt{1-\left(\frac{h_w}{s}\right)^2}\end{aligned}\right\} \qquad (5\text{-}58)$$

Combining Equations 5-57 yields the following dimensionless relationship:

$$\left[\left[(\cos\beta - \mu_a\sin\beta) + \frac{\mu_a D}{2s}\right]\sin\alpha - \left\{\left(\frac{1+\mu_a^2}{\mu_a}\frac{s_1}{s} - \mu_a\right)\cos\beta - \left[\frac{1+\mu_a^2}{\mu a}\left(\frac{h_o}{s}\right) + 1\right]\sin\beta - \frac{D}{2s}\right\}\cos\alpha\right] = \frac{\mu_a D}{2s}\left[(s - s_1)\cos\beta + h_o\sin\beta\right] \qquad (5\text{-}59)$$

A study of Equation 5-59 indicates that the effect of weight distribution s_1/s on the obstacle climbing ability of the rear wheels (of an all-wheel drive vehicle) is exactly opposite to its effect upon the climbing abilities of the front wheels. The rear wheels are, again, the limiting factors. A nose-heavy vehicle with a long wheelbase will perform better when crossing a vertical obstacle with its rear wheels than when crossing it with its front wheels. Large s/D ratios improve the performance of the rear wheels regardless of the weight distribution. This is apparent from the free body diagram (Figure 5-17(B)). Increasing the wheelbase s increases the moment of the normal force W_1 acting on the front wheel while the remaining moments undergo little change.

Because the obstacle performance h_w/D of the front wheels responds oppositely from that of the rear wheels to changes in s/D and the weight dis-

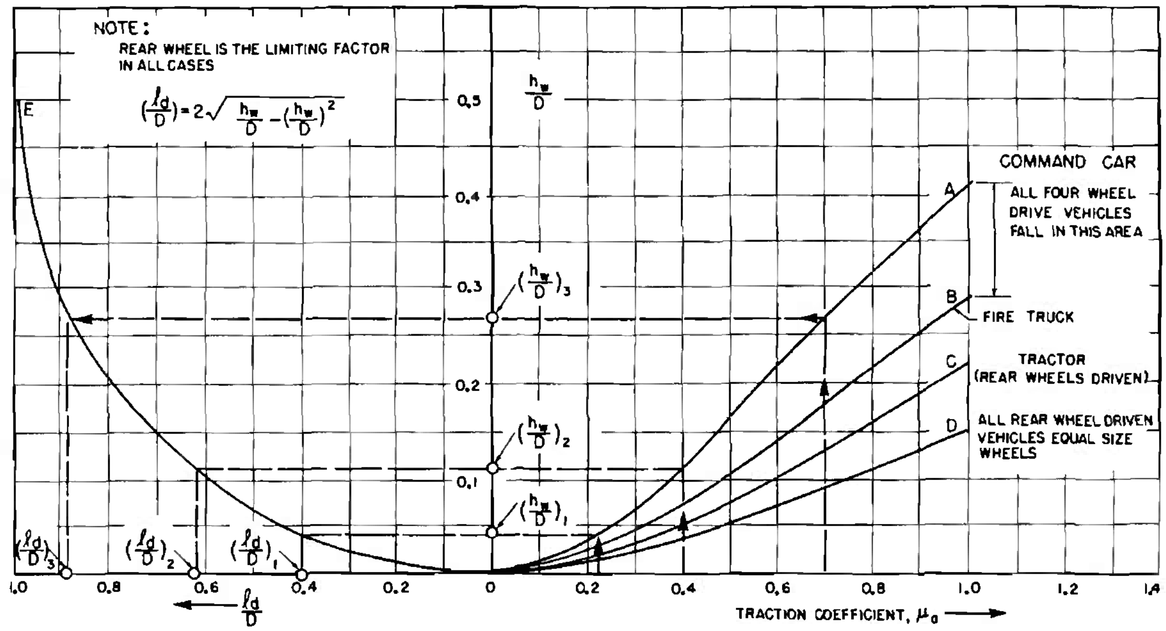


Figure 5-18. Maximum Obstacle Performances of Military Four-Wheel Vehicles (Ref. 7)

tribution s_1/s, optimum values of these parameters exist which result in equal performance of front and rear wheels, and neither limits the performance of the other. These optimum values can be determined by plotting the $h_w/D = f(\mu_a)$ lines for varying values of s_1/s for the front and rear wheels. The point at which these curves intersect is the required optimum. Optimum points may be determined for various values of μ_a and plotted on s/D and s_1/s axes. Lines of constant h_w/D may also be drawn on the same axis. The chart can then be used as a design tool to deter-

TABLE 5-1
OBSTACLE PERFORMANCE OF STANDARD U.S. ARMY FOUR-WHEELED VEHICLES*

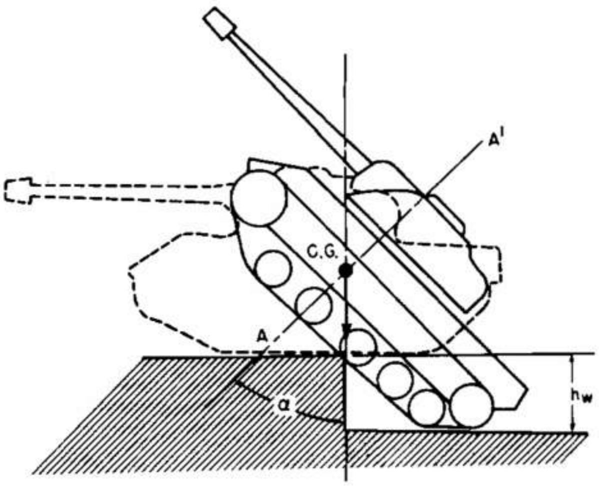
Pay-load, tons	Drive	Type or Purpose	Wheel-base Ratio, S/D	Weight Distri-bution, S_1/S	Tire Size	For $\mu_a = 0.8$ h/D	h/D (opt)	h, in.	Opt h, in.	h/h opt
1/4	4x4	Command	2.85	.608	600×16	.219	.400	6.14	11.20	.548
1/4	4x4	Command	2.85	.558	600×16	.253	.400	7.10	11.20	.633
1/2	4x2	Light Maint.	4.12	.536	600×16	.110	.400	3.08	11.20	.275
1/2	4x2	Pickup	3.83	.579	750×15	.110	.400	3.30	12.00	.275
1/2	4x2	Pickup	3.93	.581	650×16	.110	.400	3.19	11.60	.275
1/2	4x4	Ambulance	3.96	.613	750×16	.235	.400	7.60	12.4	.588
1/2	4x4	Carryall	3.73	.575	750×16	.260	.400	8.07	12.4	.650
1/2	4x4	Command	4.65	.512	750×10	.320	.400	8.00	10.0	.800
1/2	4x4	Emerg'y Repair	3.97	.640	750×16	.210	.400	6.52	12.4	.525
1/2	4x4	Panel Delivery	3.75	.556	750×16	.275	.400	8.51	12.4	.688
1/2	4x4	Pickup	3.87	.576	700×16	.263	.400	7.89	12.0	.658
1/2	4x4	Pickup	3.75	.575	750×16	.259	.400	8.05	12.4	.648
1/2	4x4	Weapons Carrier	3.75	.505	750×16	.322	.400	10.00	12.4	.805
3/4	4x4	Ambulance	3.56	.615	900×16	.225	.400	7.66	13.6	.563
3/4	4x4	Ambulance	3.56	.613	900×16	.226	.400	7.68	13.6	.565
	4x4	Scout	3.59	.594	825×20	.240	.400	8.76	14.6	.600
3/4	4x4	Carryall	3.35	.585	900×16	.245	.400	8.31	13.6	.613
3/4	4x4	Command	2.86	.625	900×16	.206	.400	7.00	13.6	.515
3/4	4x4	Command	2.86	.590	900×16	.226	.400	7.68	13.6	.565
3/4	4x4	Emerg'y Repair	3.56	.533	900×16	.290	.400	9.86	13.6	.725
3/4	4x4	Light Maint.	3.56	.556	900×16	.272	.400	9.25	13.6	.680
3/4	4x4	Phone Repair	3.56	.561	900×16	.270	.400	9.18	13.6	.675
3/4	4x4	Weapons Carrier	2.86	.620	900×16	.210	.400	7.14	13.6	.525
3/4	4x4	Weapons Carrier	2.86	.590	900×16	.227	.400	7.72	13.6	.568
1-1/2	4x2	Stake & Platform	4.09	.715	650×20	.110	.400	3.63	13.2	.275
1-1/2	4x2	Dump	4.09	.715	650×20	.110	.400	3.63	13.2	.275
1-1/2	4x4	Cargo	4.23	.650	700×20	.213	.400	7.24	13.6	.533
1-1/2	4x4	Cargo	4.15	.638	750×20	.220	.400	7.70	14.0	.550
1-1/2	4x4	Cargo	4.51	.605	750×20	.250	.400	8.05	14.0	.625
1-1/2	4x4	Dump	4.15	.645	750×20	.216	.400	7.56	14.0	.540
	4x2	Fire	4.72	.671	700×20	.110	.400	3.73	13.6	.275
	4x2	Fire	4.65	.713	700×20	.110	.400	3.73	13.6	.275
	4x2	Fire	4.85	.640	650×20	.110	.400	3.63	13.2	.275
	4x4	Fire	4.15	.670	750×20	.198	.400	6.94	14.0	.495
	4x4	Fire	4.15	.648	750×20	.220	.400	7.70	14.0	.550
	4x2	Tractor	2.16 1.39	.610	1200×24 750×16	.160	.400	7.68	19.2	.400

* Compiled from TM 9-2800. *Military Vehicles.*

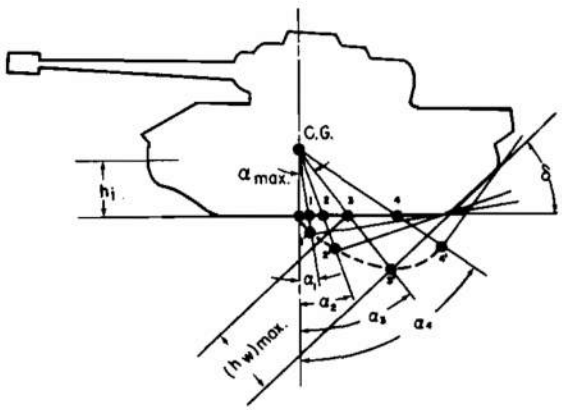
mine what s/D and weight distribution s_1/s a proposed vehicle should have for optimum design, or to evaluate an existing vehicle.

5-14.2.3 Comparison of Two-Axle and Single-Axle Drives

Studies have shown that the obstacle-climbing abilities of four-wheel-drive vehicles are far superior to comparable vehicles with only rear wheel drive. In general, rear-wheel-drive vehicles do about one-half as well as the poorest four-wheel-drive vehicle and about one-third as well as the best. The performance of rear-wheel-drive vehicles can be improved considerably by using much larger



(A) POSITION OF EQUILIBRIUM WHEN VEHICLE LEVELS



(B) METHOD OF GRAPHICAL SOLUTION

Figure 5-19. Vertical Wall Obstacle Crossing of Tracked Vehicle

wheels in the rear and shifting the center of gravity well toward the rear. Even under these conditions, however, the maximum performance will be about 75 percent of that of the poorest standard comparable four-wheel-drive vehicle.

Figure 5-18 presents a plot of maximum vertical wall obstacle performance of the military vehicles listed in Table 5-1. The superiority of vehicles with all wheel drive is readily apparent.

5-14.2.4 Track-laying Vehicles

The vertical obstacle crossing performance of tracked vehicles may also be analyzed by establishing equations of equilibrium of forces. This method of approach is described in Ref. 8. However, a more simple approach is a graphical solution described in Ref. 1. A tracked vehicle climbing a wall will level when its center of gravity passes over the edge of the obstacle (Figure 5-19). The graphical solution is obtained in the following manner.

The line A-A' (Figure 5-19(A)) is defined as a line perpendicular to the longitudinal axis of the vehicle passing through the center of gravity. For any particular wall height h_w the vehicle must rotate through an angle α for the center of gravity to pass over the edge of the wall. On a scale drawing of the vehicle, as Figure 5-19(B), a line is constructed passing through the center of gravity at some arbitrary angle α_1 with the vertical. A perpendicular to this line is then constructed tangent to the rear envelope of the track. Similar constructions are performed at different angles, as α_2, α_3, and α_4. The distances 1-1′, 2-2′, 3-3′, and 4-4′ are the respective obstacle heights which would produce the corresponding values of obstacle height h_w which will cause angles of rotation α_1, α_2, α_3, etc. The maximum h_w may be thus determined along with the corresponding angle of tilt α_{max}.

In design, the angle of departure δ should be equal to α_{max}; and it is generally accepted that the height h_i of the front wheel should be approximately equal to $(h_w)_{max}$. Obstacle heights greater than $(h_w)_{max}$ will overturn the vehicle.

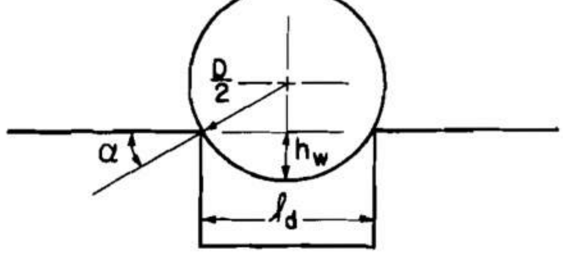
5-14.3 HORIZONTAL CAVITIES

Due to the geometrical similarity between l_d/D (Figure 5-20) and h_w/D (Figure 5-16), Equations 5-52, 5-55, 5-56, and 5-59 can be used to analyze the ditch-crossing problem. This relationship is shown in Figure 5-20 and can be obtained by applying the following conversion factor

$$\frac{l_d}{D} = 2\sqrt{\frac{h_w}{D} - \left(\frac{h_w}{D}\right)^2} \qquad (5\text{-}60)$$

With the previously listed equations solved for h_w, corresponding cavity widths l_d can be obtained from Equation 5-60.

A logical means of proceeding with the analysis



$$\cos \alpha = \frac{l_d}{D} = 2\sqrt{\frac{h_w}{D} - \left(\frac{h_w}{D}\right)^2}$$

Figure 5-20. Geometrical Relationship Between Ditch and Wall Obstacles

of ditch-crossing performance is to construct a conversion curve on the graph on which wall performance has been plotted (Figure 5-18). On the figure, curve E is the conversion of curve A; the method of construction is evident.

The ditch-crossing performance of tracked vehicles has been analyzed by Kuhner (Ref. 8). With reference to Figure 5-21, it is suggested that if the center of gravity is located half way on the distance, $s + 0.7\,(r_f + r_r)$, the vehicle can cross a ditch of width $l_d = \frac{4}{9}\,[s + 0.7\,(r_f + r_r)]$, where r_f and r_r are the radii of the front and rear wheels, respectively.

5-15 EFFECT OF VEHICLE MORPHOLOGY

5-15.1 CONVENTIONAL VEHICLE FORMS

The origin of the vehicle form and the details of form geometry and dimensions were discussed in Chapter 3. It was pointed out that off-road vehicle form developed independent of its operational environment. Wheeled vehicle form developed as an extension of the horse-drawn carriage; and tracked vehicle form, from considerations of steerability. The net effect is that the vehicle is often too wide to pass between obstacles, or too long (rigid length) to thread its way between them.

Vehicle form is also important in overcoming obstacles. Protuberances on the underbelly can be snagged on bumps which the wheels or tracks straddle. Protuberances on the vehicle in front of

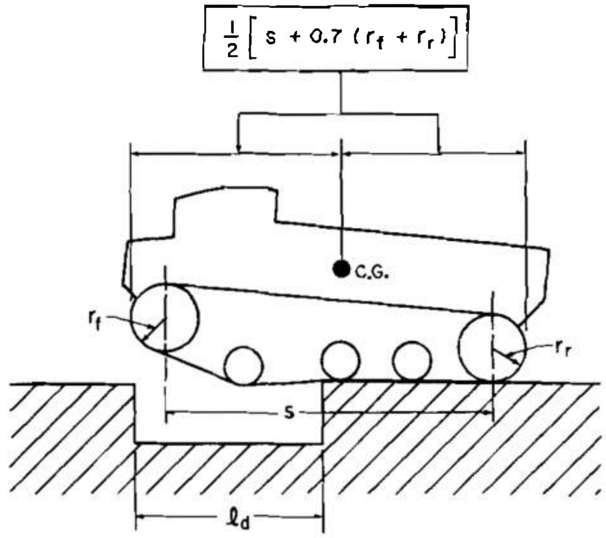


Figure 5-21. Tracked Vehicle Ditch Crossing (Ref. 8)

the wheels or tracks can dig into the ground in ditch crossing. This was dramatically illustrated in World War I when the St. Chammond tank failed on a French battlefield because a protruding portion of armor prevented it from climbing out of trenches (Ref. 1).

5-15.2 MULTI-UNIT VEHICLES

Multi-unit vehicles are discussed at greater length in the next section. For the present it is sufficient to state that they consist of a number of vehicles coupled together. Hard soil performance of multi-unit vehicles is principally a function of the damping and mechanical steering provided in the coupling design.

For any multi-unit vehicle there is some speed at which lateral oscillations, snaking or fishtailing, will occur. The speed at which lateral oscillations become critical depends on interrelationships between vehicle characteristics, coupling characteristics, and the nature of the ground surface, e.g., the amplitude and frequency of ground protuberances and cavities. For example, consider a multi-unit vehicle with no damping at the couplings between units. The first unit encounters a bump which causes side sway. Lateral motion will also be produced in the trailing units through the couplings. If the speed of the vehicle is such that the side sway caused when successive units encounter the bump is additive, the sway will increase in magnitude with the passage of each successive unit over the bump. This process can cause violent oscillations to occur.

Currently, research is being conducted to study this problem area (Refs. 9 and 10). Results indicate that the multi-unit vehicle is feasible, that the coupling control system need not be complex, and that significant improvements in ride and speed can be obtained with multi-unit vehicles with respect to single unit vehicles.

SECTION IV MULTI-UNIT VEHICLES

5-16 THE TRAIN CONCEPT

The train concept is based on the principle which railroads use to increase their cargo-carrying capacity. Although there are several basic railroad car designs, tankers, freight cars, coal hoppers, etc., the overall size of the cars does not vary greatly. To obtain any desired payload capacity a number of cars are coupled together to form a long articulated vehicle. In this manner, the individual unit or car ground pressure is maintained constant and there is no sacrifice in the overall vehicle maneuverability.

Adaption of this principle appears to be the only rational method of increasing the payload capacity of off-the-road vehicles. Continuation of current design trends can only lead to large vehicles with inadequate maneuverability, or vehicles or undesirably high ground pressure. An initial start toward the train concept is seen in the design of vehicles of approximately 40 or more feet in length, e.g., semitrailers. However, in developing the train concept for off-the-road locomotion the semitrailer, or two unit vehicle, is not considered to be in the vehicle train category. Vehicle trains are comprised of three or more units, and generally, each unit will be a driving unit and may even be capable of operating independently. This concept is presently in the research and development stage and some of its advantages and limitations are discussed in the paragraphs which follow.

5-17 ADVANTAGES OF THE TRAIN CONCEPT

5-17.1 GROUND PRESSURE INDEPENDENT OF VEHICLE SIZE

Under the train concept, vehicle size is increased by increasing the number of units which comprise the vehicle. The individual unit size need not be varied, hence each unit can be designed to produce the same ground pressure. This permits the vehicle train to vary in size and weight while unit ground pressure and vehicle ground pressure remain constant.

This is of great advantage where soft soil conditions will permit movement only of low ground pressure vehicles. The required cargo-carrying capability can be achieved by coupling together a sufficient number of low ground pressure units.

5-17.2 MANEUVERABILITY INDEPENDENT OF SIZE

The maneuverability of a railroad train is independent of size, because the train follows the course of the tracks it travels over. The train itself has no ability to maneuver. For off-road locomotion, however, the vehicle must be maneuverable.

The train vehicle does not have the pivot turn capability of a skid-steered vehicle. However, the train vehicle, by means of articulated steering, has maneuverability independent of the size of the train. The articulated train maneuverability has significant advantages over that of large single vehicle units. The moment of resistance to turning created by the sliding frictional forces is much less for the smaller train units than the long single unit. This is because the moment arm of the frictional forces is reduced, and because frequently only a few units are simultaneously involved in the turn.

5-17.3 IMPROVED TRACTION THROUGH ALL WHEEL DRIVE

As with any vehicle, traction is most efficient with all wheel drive because the total vehicle weight is utilized in mobilizing the frictional portion of the soil shearing strength. With a train vehicle, an additional advantage is realized due to the vehicle length. In traversing soft soils, the units which have the firmest soil conditions serve to push or pull those units encountering the poorest soil conditions.

For highway travel and when off-road conditions permit, it is most efficient to supply power only to one or two lead units as necessary to tow the remainder of the units in the fashion of a railroad train.

5-17.4 MORE EFFICIENT SHAPE

Because the train vehicle grows only in the longitudinal direction, it achieves the most efficient shape for off-road locomotion. It maintains constant width regardless of payload and can pick and choose the most advantageous path across the terrain, passing between obstacles that a wide vehicle would be required to either circumvent or go over. The train has also the long narrow form desirable in soft soil conditions to achieve the minimum resistance to motion.

5-17.5 JUSTIFICATION AND BACKGROUND

5-17.5.1 U.S. Army Overland Train

The vehicle designed by R. G. Le Tourneau, Inc., has been developed to the use stage. The prototype was put into service in 1956 and tested in the Arctic. The vehicle is 572 ft long and is composed of 13 cars. It has the cargo-carrying capacity equivalent of a fleet of sixty 2½ ton Army trucks but requires a crew of only six men.

Ten of the train's cars are for cargo; two carry the power plants; and one contains the controls, crew quarters and an auxiliary engine. The control car is a six-wheeled vehicle capable of independent operation; it is the lead car in the train and serves as a switch engine for making up the train. The remaining cars are four-wheeled vehicles not capable of operating alone. All wheels are 4 ft wide, 10 ft in diameter, and are independently driven by electric motors. The vehicle has a top speed of 20 miles per hour, and is capable of operating in snow and over a wide range of soil conditions.

5-17.5.2 The Cobra

The Cobra is an experimental, three-unit, tracked vehicle developed by Wilson-Nuttal-Raimond-Engineers for the U. S. Army Tank-Automotive Center (Ref. 11). A single engine mounted in the lead unit is capable of delivering tractive power to each of the other units. The vehicle, which grosses twelve tons, has proven to be highly mobile in a variety of terrains. In addition, it was found that the three-unit configuration permitted higher loads on the track and suspension than were possible with only two units of the same size.

5-18 DISADVANTAGES OF THE TRAIN CONCEPT

As there are two sides to everything, so there are disadvantages as well as advantages to the train concept when applied to cross-country vehicles. It is, therefore, only fair that the designer be made aware of some of the disadvantages.

Since the train is composed of many similar units, the problems of maintenance and stockage of spare parts takes on increased proportions. The number of times each maintenance task must be performed is multiplied by the number of units that comprise the land train. Consider a relatively simple four-unit train-type vehicle. If each unit has four wheels, the entire vehicle will have a total of 16 wheels. The routine inspection and adjustment of inflation pressures, for example, becomes four times as great a chore as it is on a conventional four-wheeled vehicle and becomes even greater as more units are added. In many respects, the maintenance and logistics required by a multiple-unit vehicle are comparable to those required by a fleet of an equivalent number of conventional vehicles as units comprising the land-train vehicle.

Furthermore, if the train vehicle is of a type that uses a central power generating car from which power is distributed to driving motors at each axle or wheel, any malfunction or scheduled maintenance that deadlines the power unit deadlines the entire train. A similar deadlining of a power unit in a fleet of conventional vehicles deadlines only one vehicle while the remainder are free to operate. Providing each unit with its own power generating system will circumvent this difficulty but this concept becomes not too dissimilar from a fleet of conventional vehicles coupled together. A standby power unit is another solution; and, to reduce the economic burden, one standby power unit can serve several train-type vehicles.

Another disadvantage of the train concept relates to the frequency with which an obstacle is encountered. A large, single-unit vehicle encounters an obstacle in its path (bump or hole) with its front and rear wheels, and is done with it. A multiple-unit vehicle, however, must endure each of its many units negotiating the obstacle individually, and it cannot resume its normal speed until the last axle of the last unit has cleared the obstacle. By this time, depending upon the length of the vehicle and the frequency of the obstacles, the leading unit may be encountering another obstacle. Thus, the overall speed of the train-type vehicle will be slower than that of a single-unit vehicle since the latter can increase its speed between obstacles.

Steering, too, presents problems not encountered with single-unit vehicles. Successive units of a land train do not track in the exact path of the preceding units when negotiating a curve—each successive unit takes a path along a larger radius than that taken by its preceding unit. Steering linkages between the units (not required with conventional vehicles) can do much to improve this but they cannot reduce the tracking error to zero. Furthermore, the passing of each unit of the train over the same ground alters the soil characteristics so that each successive unit is encountering a different soil environment. This difference may be quite significant when the route is over marginal soil to begin with.

Thus, each successive unit will negotiate the curve by following ever increasing arcs making it necessary for the train-type vehicle to have more lateral space (road width) to negotiate a turn than is required by conventional vehicles. Furthermore, the driver is unable to observe the clearances around the trailing units of his train when rounding a curve, especially if the train has any appreciable length or tall vegetation or other obstructions are present. Should the driver of a long train-type vehicle begin a turn and discover, when part way through, that there is insufficient lateral clearance to complete it, there is little he can do to correct this situation; backing a long, multi-unit vehicle around a curve is virtually impossible and can be accomplished only with a considerable amount of luck.

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CHAPTER 6

UNUSUAL SUSPENSION CONCEPTS*

SECTION I UNUSUAL TRACK TYPES

6-1 SPACED-LINK TRACK

The spaced-link track concept resulted from analytical studies of the behavior of soils under vehicle action that were conducted by M. G. Bekker, then of the Directorate of Vehicle Development of the Canadian Army and later Director of the Land Locomotion Research Laboratory of the U. S. Army Tank-Automotive Center, Detroit Arsenal. Recognizing that the most critical obstacle to vehicle movement is the soil (which includes mud, sand, clay, snow, etc.) upon which the vehicle has to operate and which it has to utilize to develop sufficient horizontal thrust or tractive effort for locomotion, he applied known principles of soil mechanics to the design of a more effective track.

A particular terrain can be satisfactorily traversed by a vehicle only if (a) the soil will support the vehicle at some maximum sinkage where its major nondriving elements are clear of contact with the soil, and (b) the soil has sufficient shear strength to allow the vehicle to develop a greater tractive effort than its resistance to motion. Soils resist deformation through their shear strength. By shear strength is meant the ability of the soil to resist the slippage of a loaded portion of the soil with respect to an adjacent nonloaded soil mass with which it is in contact, and it is derived from the internal friction, cohesion, and density of the soil. These properties, and consequently the shear strength, vary greatly with the soil's moisture content.

The nature of shearing forces in soil is illustrated in Figure 6-1. A simple masonry footing is shown upon a supporting soil. As the footing is loaded, it tends to sink into the soil; but in order for it to do so, it must displace an equivalent volume of soil outward and upward. The heavy curved lines in the figure indicate the surfaces along which shear failure will occur if the soil is overloaded. A certain amount of elastic and plastic deformation of the soil may take place, depending upon the type of soil involved; however, a predominantly cohesionless soil will exhibit a shear pattern approximately as shown.

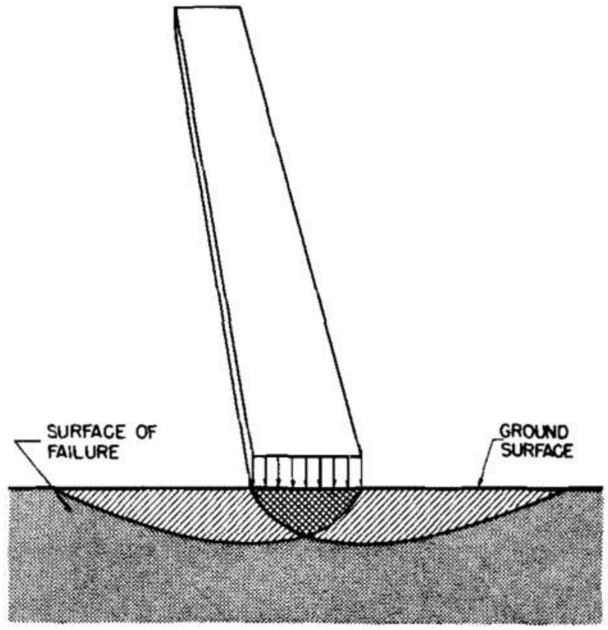

Figure 6-1. Surfaces of Soil Failure Under a Statically Loaded Rectangular Footing

If a plain surfaced track shoe is considered instead of a statically loaded footing, the conditions will be similar to those shown in Figure 6-2. The track shoe is subjected to horizontal load H due to the thrust of the driving mechanism as well as to a vertical load V. The resultant of these two forces is transmitted to the soil at an incline to the vertical, and the shear pattern will be approximately as indicated by the heavy lines in Figure

*Written by Rudolph J. Zastera of the IIT Research Institute, Chicago, Ill.

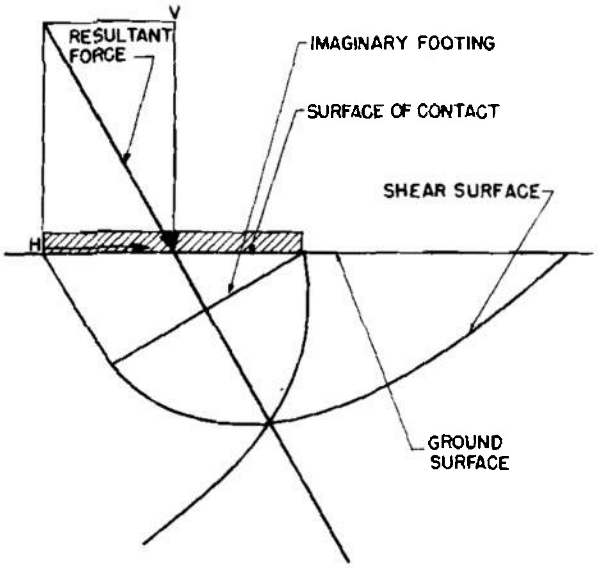

Figure 6-2. Surfaces of Soil Failure Under a Plain Track Shoe

6-2. Although this pattern is symmetrical about the axis of the resultant force, it will produce a nonsymmetrical failure, since failure to the left cannot take place to the same extent as to the right; failure along the shear curve to the right will always relieve the stress. The magnitude of the horizontal forces is limited by the coefficient of friction, the track shoe material, and the surface of the soil.

The same condition exists in the case of a plain tire tread in contact with soil. If the track shoe, or tire, is provided with a tread pattern to improve its grip of the ground, the horizontal force will be limited by the shear strength of the soil, generally a higher value.

Figure 6-3 shows a similar diagram for a track shoe provided with a vertical grouser to preclude sliding of the plate over the surface of the ground. This permits larger values of horizontal thrust.

Figure 6-4 shows two similar track shoes that are linked together closely in Figure 6-4(A) and widely spaced in Figure 6-4(B). A fundamental difference in soil behavior is evident. The soil between the closely spaced grousers of Figure 6-4(A) shears along a horizontal plane passing through the bottom edges of the grousers, and the full shear curve cannot be developed. This type of soil failure is commonly known as *grip failure* while the

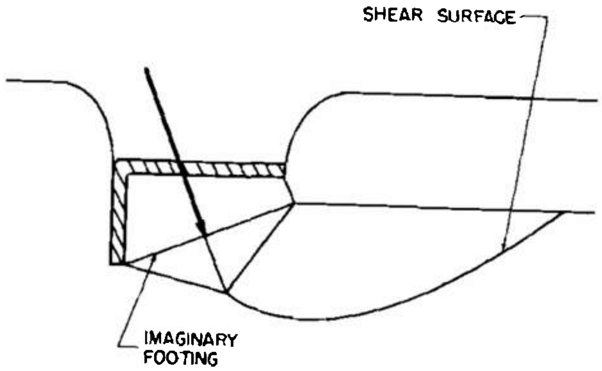

Figure 6-3. Soil Failure Pattern Under a Grousered Track Shoe

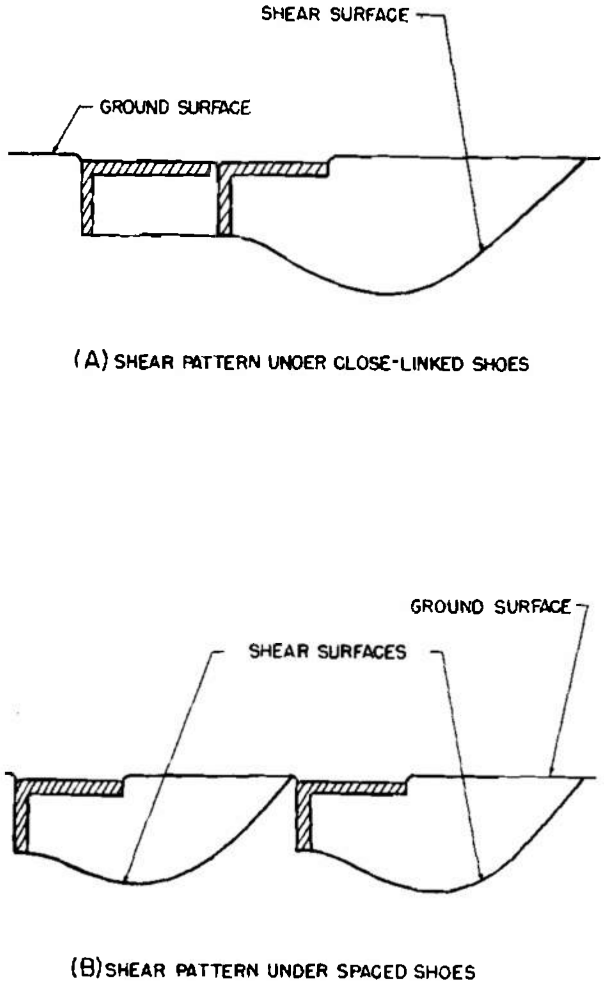

Figure 6-4. Soil Shear Patterns Under Close-Linked and Spaced Track Shoes

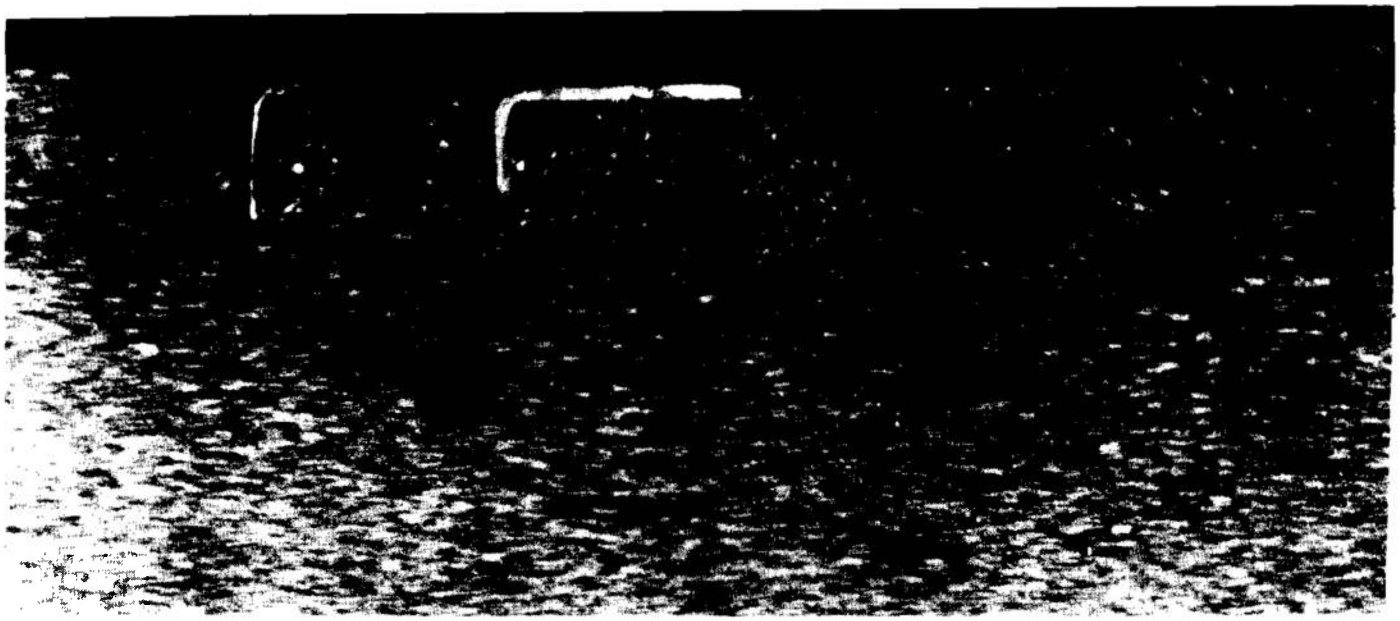

Figure 6-5. Photograph of Soil Shear Pattern Under Action of Grousered Track Shoes

type shown in Figures 6-3 and 6-4(B), where the full shear curve is developed, is known as *ground failure* or *soil failure*. The track shoes of Figure 6-4(B) are spaced sufficiently far apart to prevent interference of the grousers with the shear curve generated by the preceding grouser. By spacing the grousers along the length of the track at points where the preceding shear curves just meet the surface of the ground, a maximum soil mass can be subjected to shear and a maximum tractive effort can be developed.

Since the actual size and shape of the shear curve varies with soil characteristics and loading, the spacing of the grousers must be optimized for the conditions to be encountered.

Figure 6-5 is a photograph of an experiment conducted to demonstrate the shear pattern in a soil subjected to a horizontal thrust by two grousered track shoes. Note the typical grip failure of the soil between the two closely spaced shoes and the fully developed shear pattern to the right of the right-hand shoe where true soil failure took place. This photograph substantiates the theory of which an over-simplified synopsis has just been given here. The complete analytical treatment of the subject is quite involved and can be found in Refs. 1 and 2.

The spaced-link track, then, is a track design resulting from these analytical studies. Following laboratory testing on scale models, an experimental vehicle was constructed which has since been designated as The Ground Hog (Figure 6-6). This vehicle is merely a test rig whose purpose was to compare the tractive ability of the spaced-link track with conventional close-linked tracks and to demonstrate the accuracy with which performance can be predicted based upon laboratory tests.

The Ground Hog is a bellyless vehicle that utilized the suspension components and power train of the M29 Cargo Carrier (Weasel) with the addition of an extra bogie, to increase the track length, and an additional gear reduction between the differential axle and the driving sprockets to provide sufficient torque for maximum tractive effort. In mud or deep snow, the vehicle sinks deeply until soil of sufficient strength for support and traction is encountered. The bellyless design and openness of the track makes this large sinkage possible. Furthermore, the open design of the track presented a minimum frontal area to the mud or snow pack and allowed the mud to flow freely around the track components, thus minimizing the vehicle's resistance to movement. Figure 6-7 shows a close-up of the spaced-link track of the Ground Hog ve-

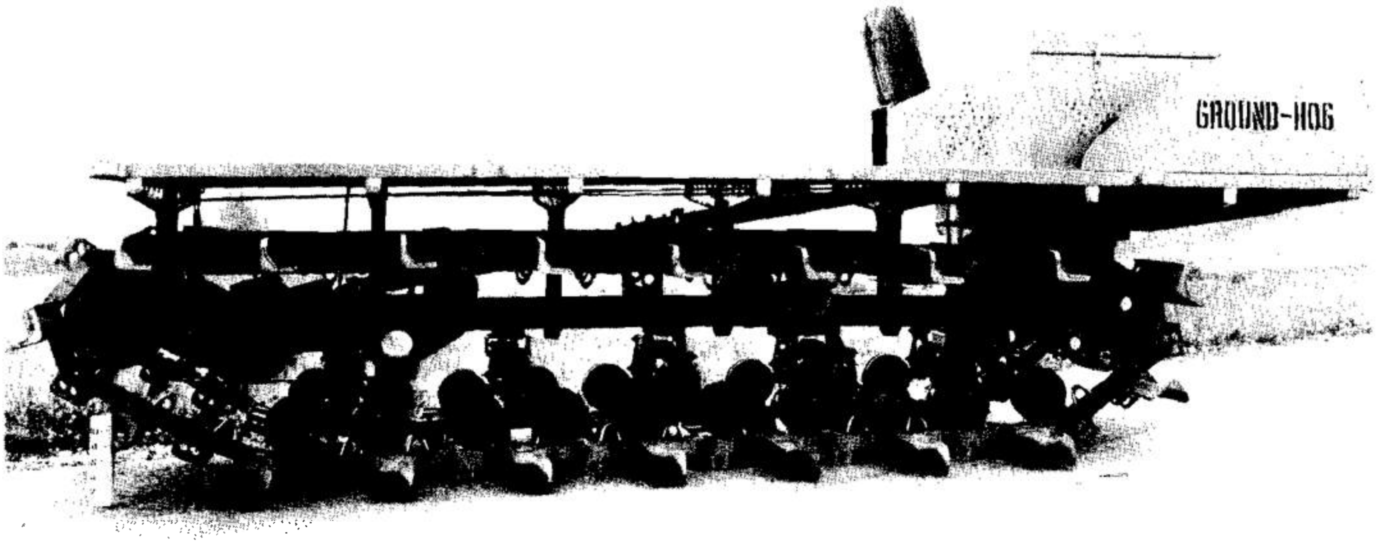

Figure 6-6. Spaced-Link Track on Experimental Test Bed; Ground Hog Vehicle

hicle and the extent to which the soil between the track cleats is being sheared.

When tested in deep snow, the Ground Hog developed a maximum tractive effort equal to 84 percent of the gross vehicle weight, which was a greater tractive coefficient than was developed by any other of 18 different vehicles tested under the same conditions, and 2.4 times greater than was developed by the M29 Weasel with which it was specifically compared because of its equivalent weight and similar suspension and power train. The slope-climbing ability of the Ground Hog in snow was also far superior to the other vehicles tested (Ref. 3). When tested in sand, the Ground Hog developed 50 percent more tractive effort than did the similarly weighted M29. In negotiating wet, soupy mud in a 4 to 5 ft deep basin, it was compared with three other vehicles: the standard M29 with 20-in. wide tracks fitted with 1-in. grousers, a modified M29 with the tracks widened by 7 in. and grousers extended to 3 in., and the M76 Otter. All vehicles, except the Ground Hog, became immobilized. This

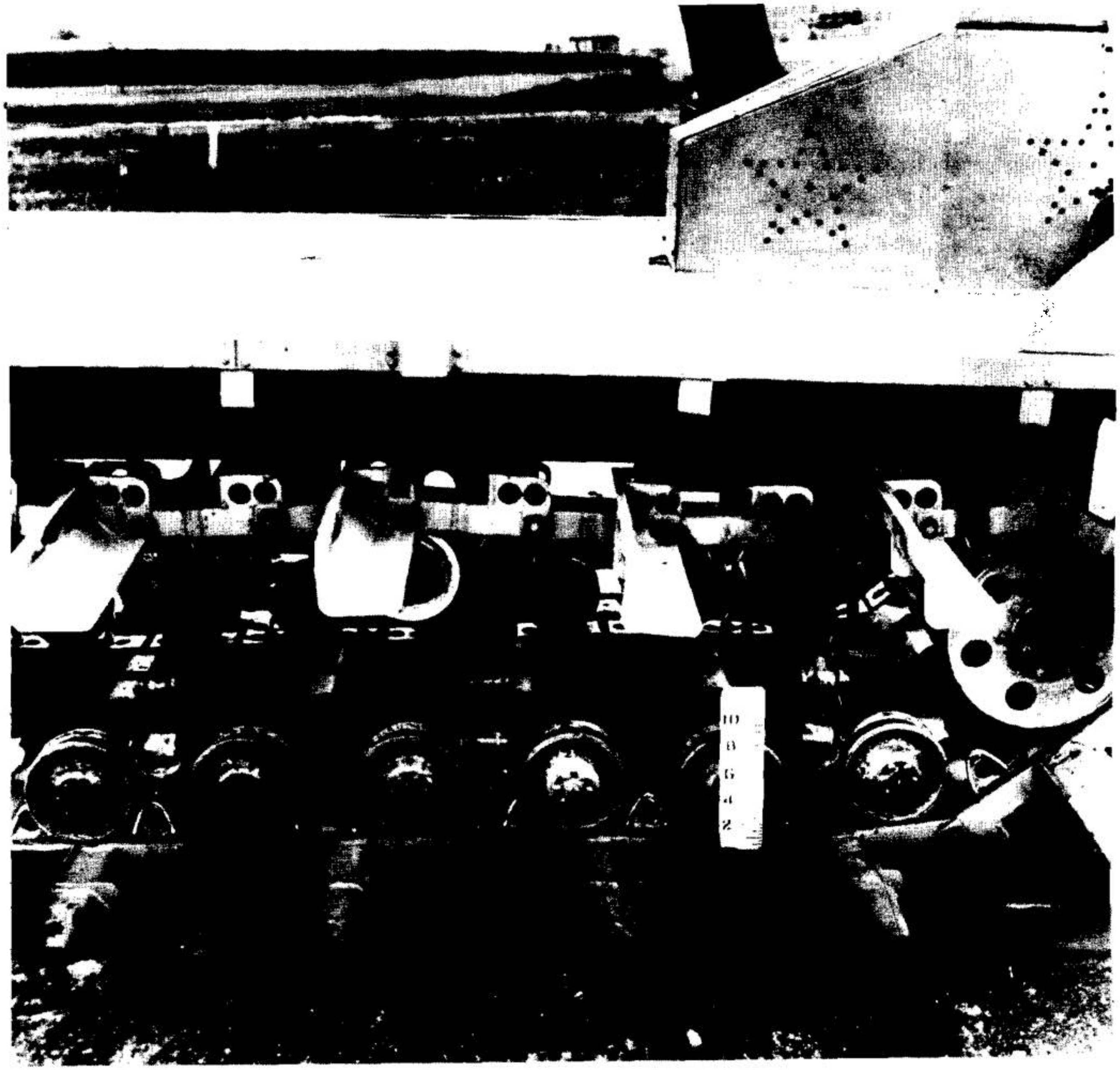

Figure 6-7. Soil Failure Under Spaced-Link Track

vehicle crossed the swamp with ease, practically without slip, with the engine at only part throttle and the transmission in high gear.

The successful demonstrations of the Ground Hog led to the development of the T60 test vehicle to proof-test the practicality of the spaced-link track for lightweight, cross-country vehicles. This test vehicle was intended to establish the basis of design for prototypes of the T60 amphibious cargo carrier, a projected ½-ton cargo carrier for operating over water, soft marshy terrain, muskeg, sand, snow, ice, and tundra. When first tested by the manufacturer, the vehicle successfully scaled ridges of loose "gumbo" of 72.6 percent maximum slope, negotiated a pool of water about 3 ft deep, and towed a 13,625 lb tracked bulldozer with its tracks locked and its dozer blade down, through approximately 18-in. deep mud (Ref. 4). In tests conducted at Aberdeen Proving Ground (Ref. 5), the vehicle operated successfully up a 60 percent slope with the transmission in high gear and the transfer case in low. In sand and mud, the vehicle successfully towed a 105 mm howitzer weighing 3,900 lb; it also pulled the M76 amphibious cargo carrier, with the 105 mm howitzer in tow, from the mud course when the M76 was unable to continue maneuvering. Operating in soft sand, the T60 test vehicle developed a drawbar pull equal to 73 percent of its gross weight. This exceeded the percentage drawbar pull of the M29 Weasel by 14 percent and that of the M76 Otter by 27 percent.

After successfully negotiating a test course on the edge of a tidal marsh, it was decided to install a set of flotation belts designed to improve the flotation by masking the track spacing. These belts did not improve the flotation and caused vast amounts of muck to be retained within the track periphery, which had an adverse effect upon vehicle performance. It became necessary to shift the transmission to low range (which increased the overall ratio by more than 3.0) and to use full throttle instead of the part throttle which had previously sufficed. This incident furnished eloquent testimony to the importance of movement resistance and the advantage, in this regard, of the open track design on this vehicle. It also emphasized that a reduction in motion resistance is as effective in improving mobility as is an increase in traction.

The major defects of the spaced-link vehicles tested were poor maneuvering in deep snow and an appreciably greater resistance to towing than comparable closed-track vehicles. The steering difficulties encountered when operating in deep snow were a major handicap which required further study and development.

6-2 PNEUMATIC SKID SUSPENSION

There is some question whether the pneumatic skid suspension should be classified as a track or a wheel. It was intended to be a replacement for a wheel; but then, so was a track. A track is really a flattened wheel, anyway. Since the pneumatic skid suspension resembles a track more than it does a wheel, it is so classified here.

It has been pointed out in the preceding chapters that the larger the wheel diameter, the better the performance. There is obviously a limit to this trend, however, since sooner or later the size, weight, and cost of the wheel become prohibitive. The pneumatic skid suspension concept is an attempt to produce a long ground-contact area (long in the direction of travel) without incurring the undesirable height of a large wheel. Its principles are illustrated in Figure 6-8. The suspension unit consists of a pneumatic cushion *a*, interposed between a supporting frame *b* and a rubber track *d*. The supporting frame *b* also provides support for the two pulleys *c* which may be driving sprockets, idlers, or one of each whichever suits the vehicle design. In operation, the track *d* slides under the cushion. The rubbing surfaces are well lubricated with either solid or liquid lubricant and result in very low friction losses. The sides of the unit are completely sealed from the outside by diaphragms *e* which prevent dirt and water from penetrating between the rubber track and the pneumatic cushion and also keep the lubricant from escaping.

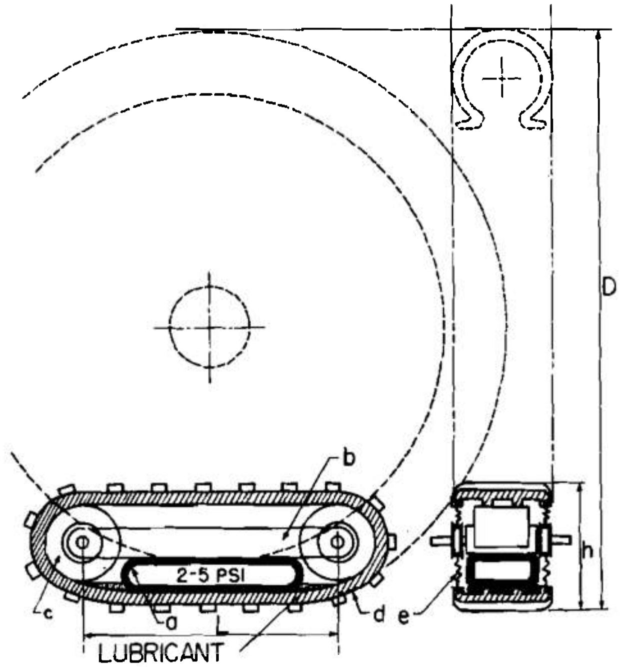

Figure 6-8. Pneumatic Skid Suspension

A suspension unit of this type produces a ground-contact area of a length L shown in Figure 6-8. In order to produce the same length of ground contact with a pneumatic tired wheel, a wheel with an approximate diameter D superimposed over the skid suspension in Figure 6-8, would

Figure 6-9. Skid Suspension Compared to a Corresponding Tire

be required. The height of the pneumatic skid suspension is shown by dimension h in the figure.

Laboratory tests have shown that this type of suspension is entirely feasible. Figure 6-9 shows a comparison between an experimental skid suspension and a corresponding pneumatic-tired wheel that produces a similar ground-contact area. The wheel shown has a 14.00 $\times$ 20 tire.

6-3 ROLLER TRACK

The roller track is an experimental track concept intended to give a vehicle improved mobility in extremely soft mud. It consists of a series of inflated rubber rollers (rolligons) placed parallel to each other and with their axes interconnected at each end by a roller chain to form an endless track configuration. The pneumatic rollers are free to rotate about their axes. One such track is mounted

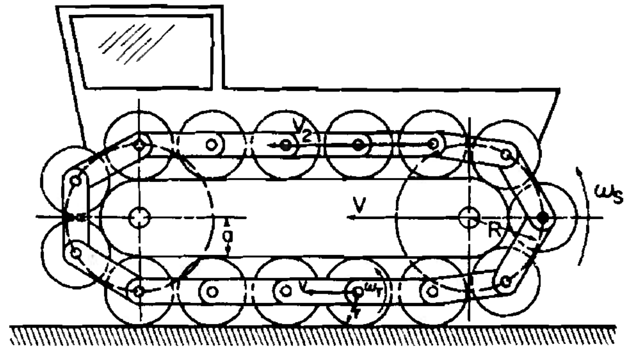

Figure 6-10. Velocities in Roller Track Vehicle

on each side of the vehicle and is driven by a sprocket which engages the roller chain at one end of the vehicle. Track tension is maintained by an idler sprocket at the other end of the vehicle. Thus it is driven in much the same manner as is a track on a conventional tracked vehicle. The vehicle hull rests directly upon the inflated rollers which provide the only elastic support between the sprung mass and the ground. In this manner, the rollers perform the multiple functions of springs, shock absorbers, and propulsion elements.

The action of the rollers is quite different from that of track shoes of a conventional tracked vehicle. In a tracked vehicle, the track shoes in contact with the ground remain stationary (except for slippage) while the road wheels and sprung mass pass over them. As the end of the vehicle passes, they are picked up and carried forward, at twice the speed of the vehicle, to be placed beneath the front of the vehicle. Since the rollers of the roller track are free to rotate while in contact with the ground, they have a translational velocity in the direction of vehicle motion due to their rolling over the ground. In addition, they impart an equal translational velocity to the vehicle hull due to their rolling contact with the hull. This causes the hull to move forward with a velocity twice that of the supporting track rollers. As the rollers are overtaken by the hull, they are picked up and carried forward at three times the forward speed of the supporting rollers. This is illustrated in Figure 6-10 where v, V, and V_2 represent the velocities, with respect to the ground, of the supporting rollers, the hull, and the returning rollers; R and r are the radii of the driving sprocket and the rollers, respectively; and ω_s and ω_r are their angular velocities. The following relationships are, therefore, evident

$$V = R\omega_s + v = R\omega_s + r\omega_r \qquad (6\text{-}1)$$

$$\omega_r = \frac{R\omega_s}{r} \qquad (6\text{-}2)$$

$$\omega_s = \frac{r\omega_r}{R} \qquad (6\text{-}3)$$

Substituting alternately the expression for ω_r and ω_s given in Equations 6-2 and 6-3 into Equation 6-1 results in

$$V = 2R\omega_s = 2r\omega_r = 2v \qquad (6\text{-}4)$$

from which it is evident that

$$R\omega_s = r\omega_r = v \qquad (6\text{-}5)$$

The velocity V_2 of the returning rollers is obviously equal to $R\omega_s$ plus the velocity of the vehicle or:

$$V_2 = R\omega_s + V = v + 2v \qquad (6\text{-}6)$$

$$V_2 = 3v \qquad (6\text{-}7)$$

The experimental test bed which utilized this unique track principle suffered considerable mechanical malfunctions during its testing and, as a result, this track principle was not thoroughly evaluated. The main difficulties were repeated breakage of the roller chain through which the track was driven and excessive track throwing. The rolligon rollers developed considerable resistance to lateral movement which made steering of the vehicle very difficult and required a large amount of power to slew the vehicle around.

At speeds in excess of approximately 15 miles per hour on relatively smooth flat terrain, the vehicle developed a pitch oscillation which increased in amplitude as the speed increased until the vehicle became unstable. The ride quality became increasingly unpleasant as the pitch oscillation increased until the vehicle had to be slowed down. This condition could perhaps be remedied by relocating the vehicle center of gravity with respect to the pitch center and by introducing more damping into the elastic support system.

An important design consideration peculiar to this type of track is the shape of the curved path followed by the rolligons as they approach the ground and receive the weight of the vehicle. For the sake of simplicity, Figure 6-10 shows this curve as an arc of a circle, but Figure 6-10 is only a schematic representation and no attempt was made to illustrate the true curve. Since the rolligons are inflated to a rather low pressure and deflect considerably when loaded, the transition from their unloaded (undeflected) state to their fully loaded (fully deflected) state must be gradual. Otherwise, each rolligon will strike the ground with a considerable impact, setting up an unacceptable vibration (and noise) within the vehicle. Obviously, this transition is too abrupt with a circular arc. The desired curve is of the form of a hyperbolic spiral, $R\varphi = a$, with its origin on the horizontal centerline of the sprocket (not necessarily coincident with its center), and a equal to the difference between the height of the origin above the ground and the loaded height of the rolligon as indicated in Figure 6-10. Both ends should be made alike to permit satisfactory operation in reverse as well as in the forward direction.

6-4 PADDLE TRACK

The paddle track was an experimental track concept developed as an auxiliary mechanism for use with amphibious vehicles to produce a dynamic lift and thrust to improve amphibian mobility, particularly when operating on the water-shore borderline. It was also intended to improve the mobility of land vehicles when operating in very loose, half-fluid soils. The concept is illustrated schematically in Figure 6-11. Its method of operation follows. Sprocket A is driven in the direction indicated. Blades D, mounted on chain C by means of pins E are free to rotate on these pins. The pitch of the blades is controlled by means of cams G fixed to the blade arms F. When the chain C causes the blades D to move, cams G follow path H. The amount of pitch variation is determined by the relative location of the chain C, cam G, and the blade pins E which follow the chain and cam path. A complete analysis of this system together with detailed test results are given in Ref. 6. Some pertinent specifications of a full size application of this concept are shown in Figures 6-12 and 6-13.

Experimental work with this auxiliary lift-

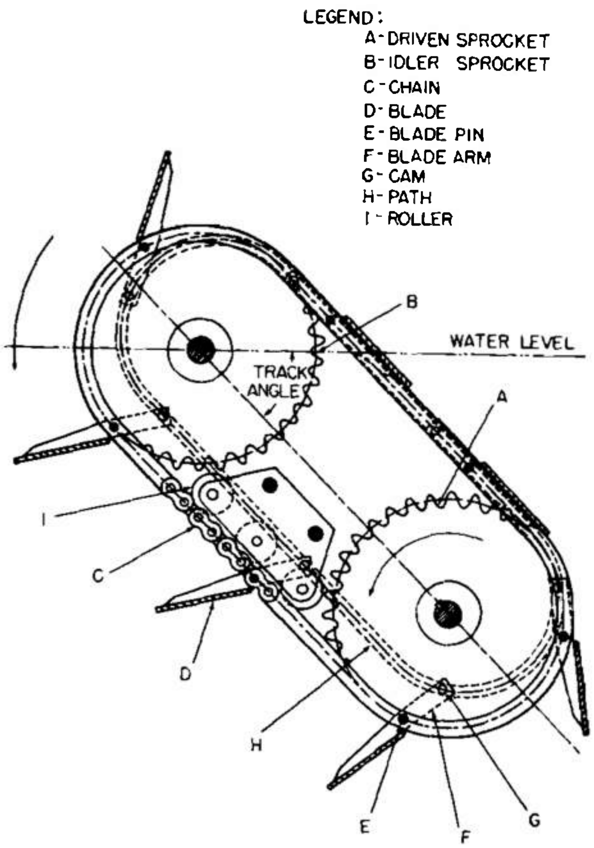

Figure 6-11. Schematic Drawing of a Full Size Paddle Track

thrust concept has revealed the following relationships:

(a) Both lift and thrust increase with increases in blade speed; however, the efficiency of the system decreases rapidly with increases in blade speed. This is quite understandable since, at higher blade speeds, losses increase due to turbulence. It was concluded (Ref. 6) that the efficient upper range of blade speed for this type of system is about 20 to 25 fps.

(b) Lift-thrust experiments performed with different sizes of blades showed that wide and short blades are more efficient than narrow and long ones.

(c) The ratio of lift to thrust can be varied by changing the track angle (Figure 6-11). The maximum lift-thrust was obtained at a track angle of 37 deg where the lift and thrust were equal. At increased track angles, lift is increased while thrust is decreased. At about 90 to 100 deg thrust will be zero while lift will be at its maximum.

(d) When incorporated into the design of new vehicles by using one unit at each end, as indicated in Figure 6-12, considerable saving of space and weight is anticipated along with improved performance over lift-thrust boosters making use of propellers or hydrojets. In addition, this system is expected to improve mobility over swamps and marshes where soils resemble a fluid more than they do a granular mass.

Figure 6-13 shows a design sketch of a full size track, based upon the rubber-water-lubrication principle, composed of rubber parts moving on a lightweight metal structure. The blades are mounted on a rubber belt which slides in a slot and is driven by a sprocket. The blades are free to rotate on their pins; rubber skids are mounted on the ends of the blade arms and the blade pitch is controlled by the cam mechanism. The pertinent calculated technical data are given in the figure.

6-5 BONMARTINI TRACK (Refs. 7 and 8)

The Bonmartini track is another experimental concept designed to bridge the gap between conventional tracks and pneumatic-tired wheels. It was developed in Italy by Count Giovanni Bonmartini for use in airplane landing gears and ground vehicles and was later applied to agricultural tractors as well. It is also referred to as a "pneumatic track" because it is inflated much in the same manner as a pneumatic tire.

Before being mounted on the vehicle, the track resembles a large diameter pneumatic tire with a relatively small cross-sectional diameter. This is stretched over concave faced driving and idler wheels to form an endless track configuration. The track is tensioned by various mechanical and hydraulic means to about 1000 to 1800 pounds (for a vehicle of about 2,200 lb gross weight), and the inflation pressure varies between 10 and 28 psi

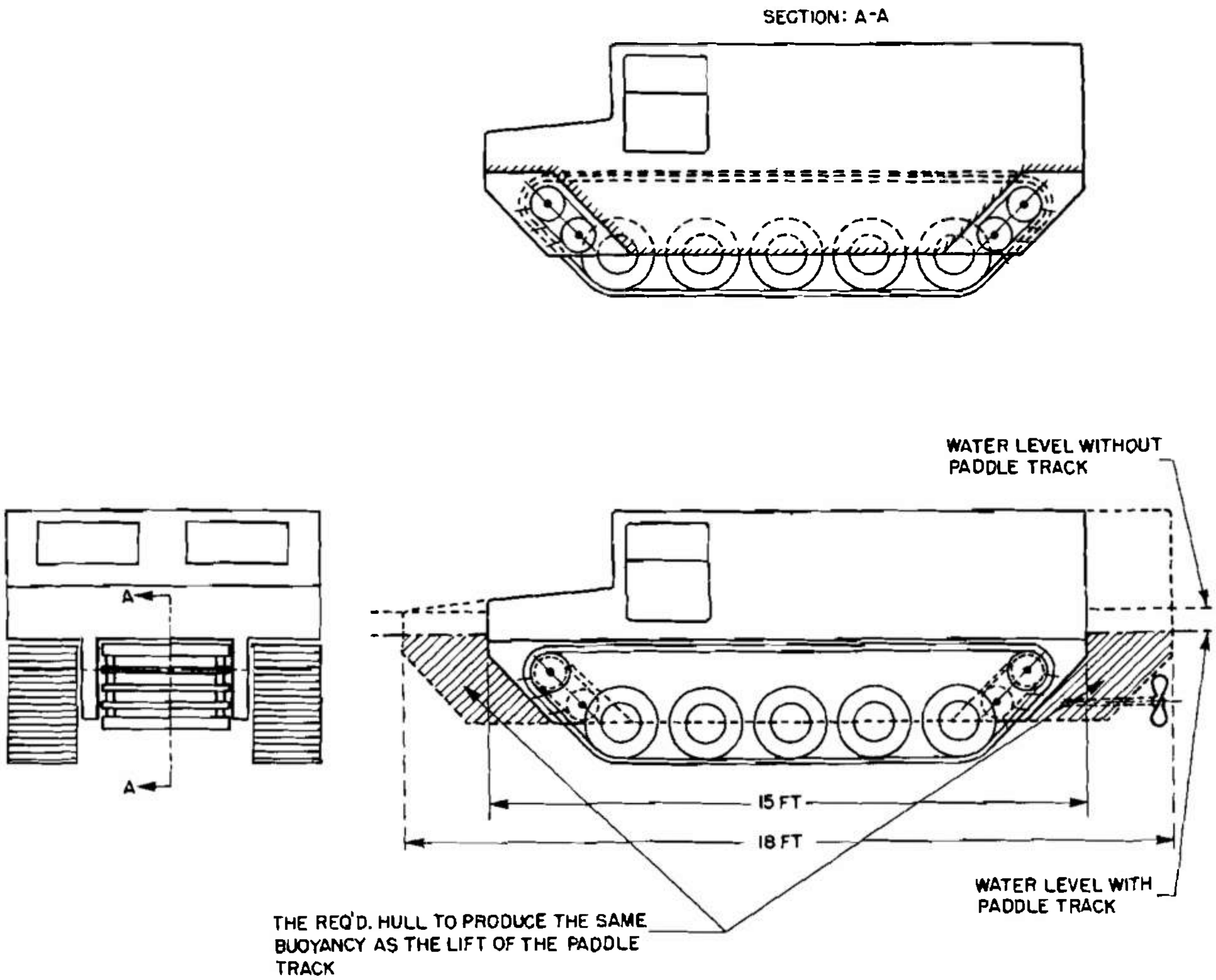

Figure 6-12. Paddle Track Applied to Amphibious Vehicle

depending upon the operational environment. The pneumatic track is molded of rubber and fabric, is of tubeless construction, and is reinforced with steel wires (in the sidewalls of the aviation type track) or a flexible, inextensible belt (in the tractor type track) imbedded longitudinally in the track walls to absorb the track tension and to improve the track's lateral stability.

The aviation-type track (Figure 6-14) is smooth on both surfaces, i.e., on its tread surface and on the surface which contacts the driving and idler wheels. In addition, the surface which contacts the wheels is lubricated.

The tractor-type track is raised cleats or lugs molded onto its tread surface. The inner or wheel-contacting surface is molded with deep, transverse grooves which mesh with sprocket-like lugs across the concave face of the driving wheel (Figure 6-15). This results in a positive track drive. The wheel-contacting surfaces of the tractor-type track are not lubricated.

Intermediate idler wheels, or rollers, are placed between the main driving and idler wheels to improve the load distribution on the ground. These can be seen in Figures 6-16 and 6-17.

Performance tests in which the performance of a lightweight agricultural tractor equipped with Bonmartini pneumatic tracks was compared to that of comparable tractors equipped with four-

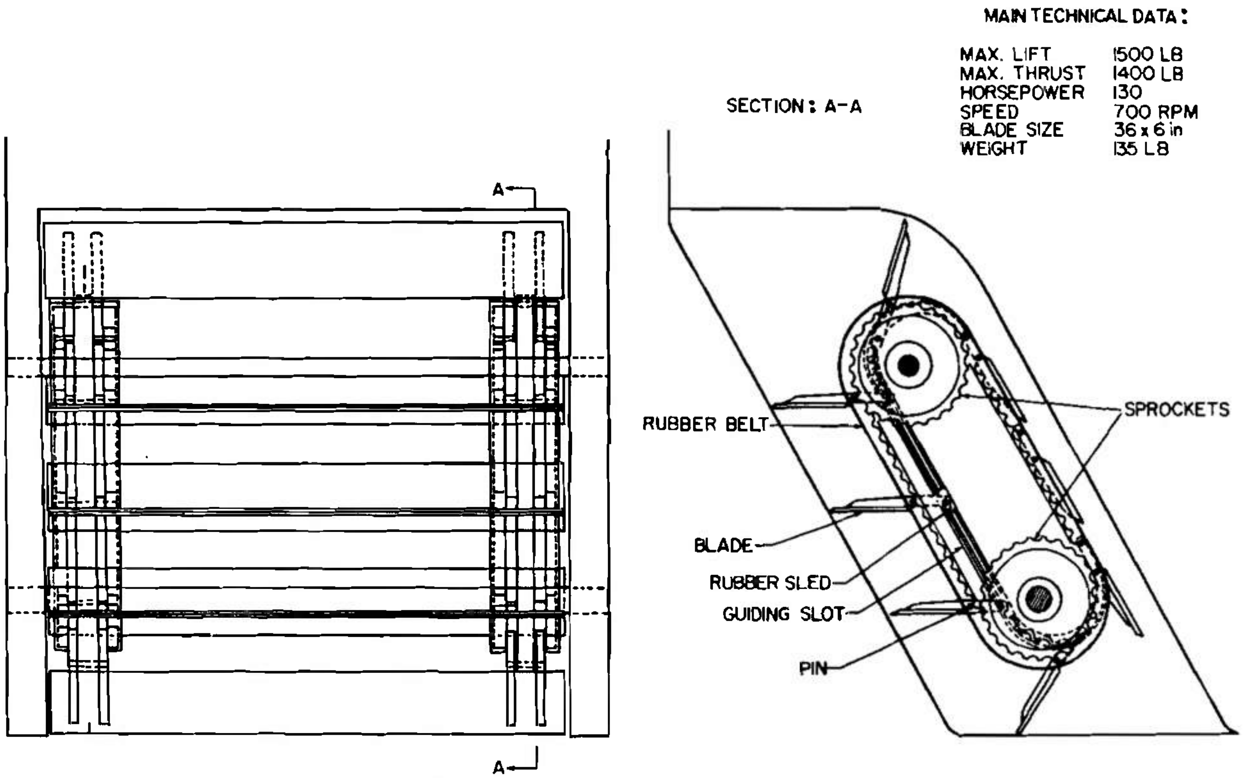

Figure 6-13. Schematic Drawing of Paddle Track

wheel drive and with steel tracks led to the following conclusions (Ref. 8):

(a) The Bonmartini pneumatic track showed greater drawbar pull on worked terrain than either of the other tractors.

(b) The ground pressure under the Bonmartini track was significantly lower than under the wheels of the four-wheeled tractors.

(c) Tractor performance of the Bonmartini-equipped tractor was less affected by soil conditions than were the wheeled tractors.

(d) Vibration levels in the Bonmartini-equipped tractor were considerably lower than those in the same tractor equipped with steel tracks.

SECTION II UNUSUAL WHEEL TYPES

6-6 CONDUAL TIRES

The advantages of a long, narrow ground-contact area to traction and vehicle performance have been pointed out in other chapters. The contact area of wheeled vehicles can be lengthened by increasing the wheel diameter, by decreasing the inflation pressure, or by a combination of both. Obviously, there are limits to these techniques; increasing wheel diameter increases the vehicle height as well as tire and wheel costs, and decreasing the inflation pressure reduces tire life and is limited by the introduction of lateral instability.

The condual tire is an experimental tire concept which permits a large radial deflection, and hence a long ground print, without exceeding acceptable limits of wheel diameter or safety. Where the cross section of a conventional tire is generally round, i.e., approximately as high as it is wide, and is comprised of one toroidal air chamber, the cross section of the condual tire is approximately

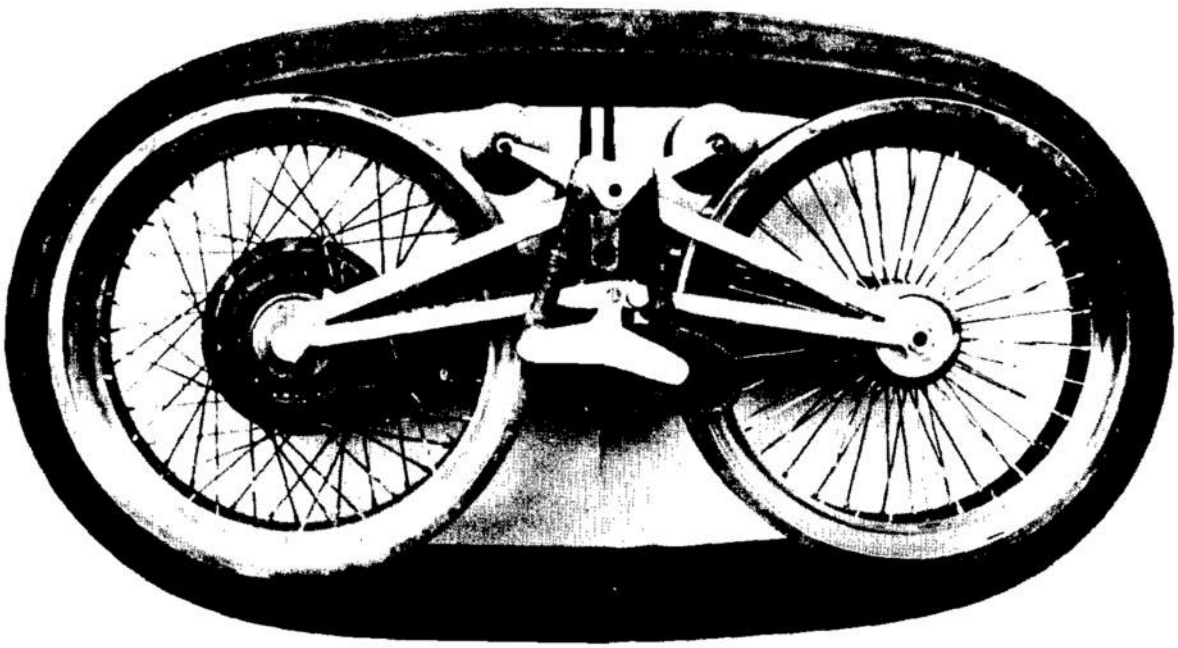

Figure 6-14. Aviation-Type Bonmartini Track

twice as high as it is wide and is comprised of two coplanar, concentric toroidal air chambers. Figure 6-18 shows a comparison between a pair of air chambers. Figure 6-18 shows a comparison between a pair of convenient dual tires (Figure 6-18(A)) and a condual (concentric dual) tire of the same outside diameter (Figure 6-18(B)). If the carcass construction material and inflation pressures are the same for the two types, the condual tire will exhibit a greater deflection per unit tire load (spring rate) than will the dual pair. The situation is analogous to having two compression springs in parallel (the dual pair) vs two springs in series (the condual pair). The greater radial deflection of the condual tire will produce more pull at the same slippage, or less slippage at the same pull.

This concept was evaluated, first analytically and then experimentally, at the Land Locomotion Research Laboratory of the U.S. Army Tank-Automotive Center (Ref. 9). The results are shown graphically in Figure 6-19 along with tire dimensions and loads.

6-7 SPOKELESS, HUBLESS, ELASTIC WHEEL

The spokeless, hubless, elastic wheel concept is shown schematically in Figure 6-20. It represents another attempt to merge the advantages of a wheel with those of a track to develop a long footprint without resorting to the large diameter of a conventional wheel. Basically, this concept consists of elastic rims, or hoops, to replace the regular wheels. The load is applied to the top of the hoops through suitable supports comprised of a number of rollers mounted to the vehicle frame Propulsion is provided by driving one or more rollers of each set. The elasticity of the hoops causes them to assume eliptical shapes under load, resulting in elongated ground-contact areas corresponding to conventional wheels of much larger

Figure 6-15. Detail of Driving Wheel Gripping Tractor-Type Bonmartini Track

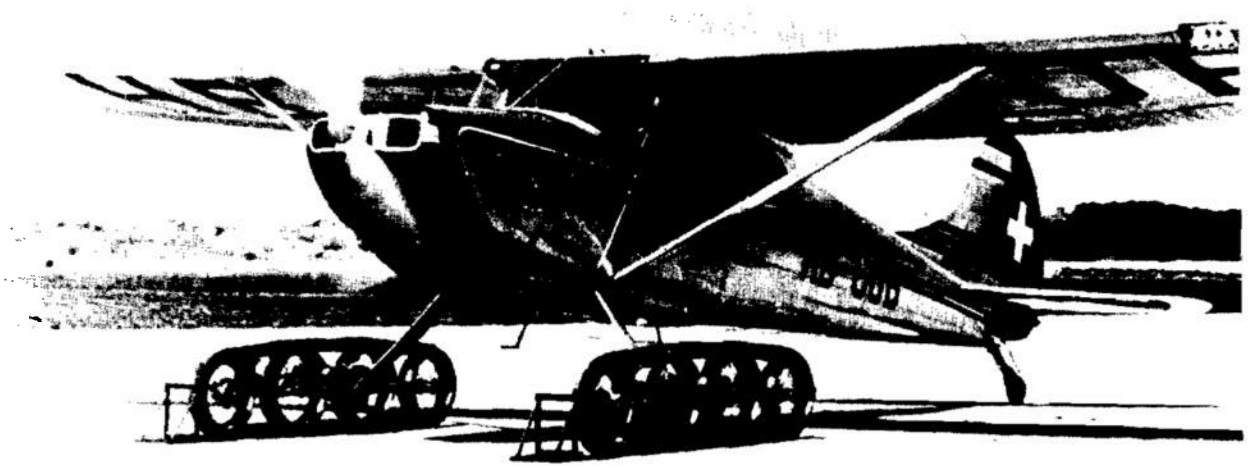

Figure 6-16. Bonmartini Pneumatic Track Applied to Aircraft

diameter (superimposed in Figure 6-20). Cleats can be added to the outer surface of the hoops to improve traction.

The system as here described makes no provision for lateral stability of the hoops, leveling of the vehicle during slope operations or accelerations, or damping of vertical oscillations. These are necessary features which can be readily incorporated into the vehicle design, however, and do not present insurmountable problems.

Where an extremely low ground pressure is required, the spokeless, hubless, elastic wheel concept can be applied in a hooped-track configuration. To do this, only one hoop need be used on each side of the vehicle; and the hoops would be proportioned to suit the vehicle length.

Steering a vehicle of this type is best accomplished by articulated steering, as indicated in the schematic vehicle of Figure 6-20, or by skid steering. The latter would be best suited to a hooped-track vehicle. Ackermann steering would be obviously impractical, although a steering system based upon a controlled warping of the hoop might be made to work satisfactorily if small turning radii are not requirements.

An interesting characteristic of this hoop concept is that the unit pressure exerted on the ground by the hoops does not increase with increased vehicle load. As the load increased, the hoops deflect more, increasing the length of their footprints and maintaining a uniform ground pressure. A similar action takes place when additional tractive effort is required. Here, again, the length of footprint increases, maintaining a uniform ground pressure. This may be a decided advantage in soils exhibiting very low bearing strength.

In summary, it must be stated that this concept is novel and relatively untried. It is doubtful whether it can be applied successfully to large,

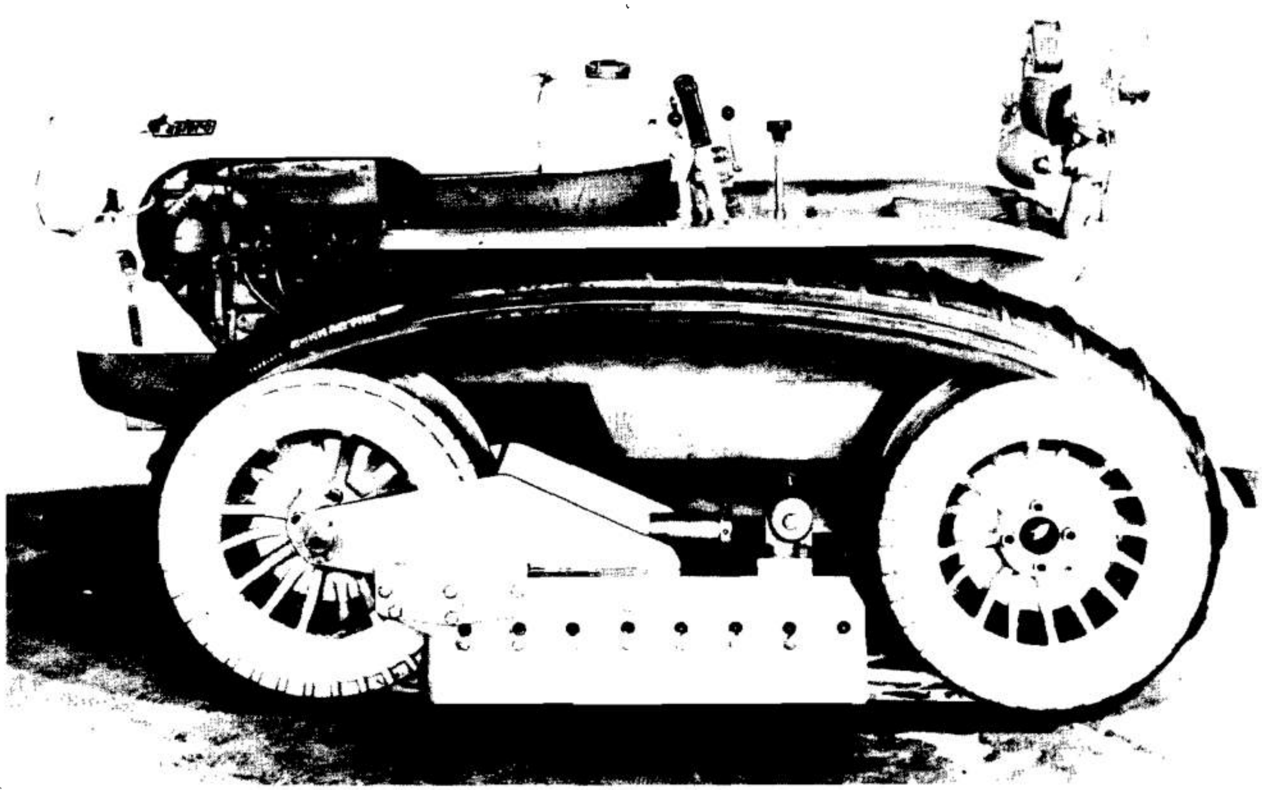

Figure 6-17. Banmartini Pneumatic Track Applied to Agricultural Tractor

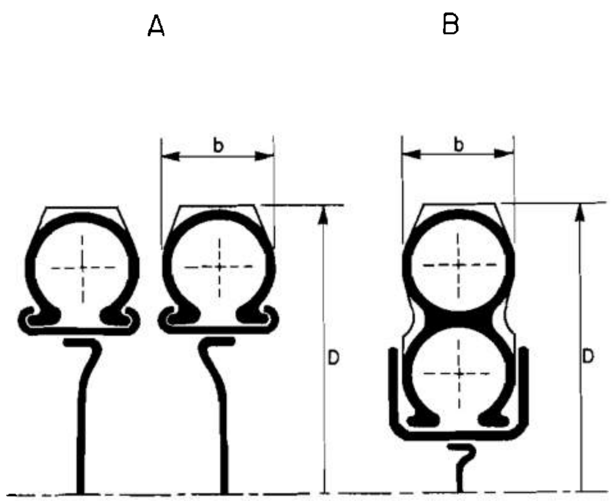

Figure 6-18. Dual and Condual Tire

heavy vehicles, but it may prove useful for small, light-weight vehicles.

6-8 METALASTIC WHEELS

Metalastic wheels is a term applied to a series of novel wheel concepts which utilize an elastic hoop for a rim and support it on a number of flexible spokes which spiral outward from a single central hub. Figure 6-21 shows one wheel of this type. The central disk serves a dual function: it limits the amount of vertical deflection of the rim, and it provides lateral stability. In operation, the metalastic wheel develops the long footprint of a track without a track's friction and weight penalties. It, too, like the spokeless, hubless, elastic wheel discussed in the preceding paragraph, maintains a constant unit ground pressure under varying loads, accelerations and tractive effort. Thus, it

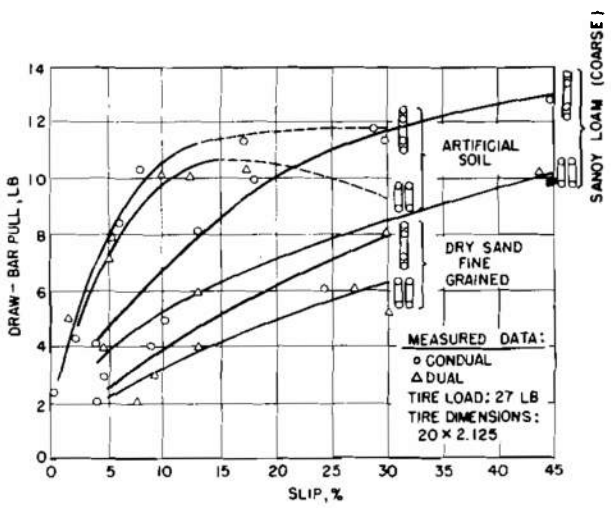

Figure 6-19. Comparative Performance of Dual Versus Condual Tires

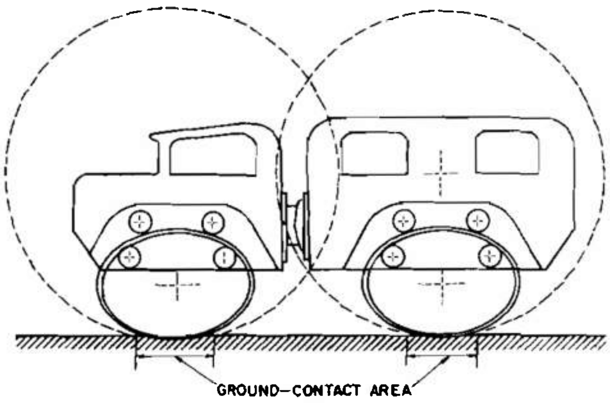

Figure 6-20. Spokeless, Hubless, Elastic Wheel

offers the load-carrying ability and dynamic stability of a wheel several times its diameter.

Wheels of this type were first conceived for locomotion over the lunar surface by unmanned, remotely controlled lunar roving vehicles, but their application to off-road, terrestrial environments is quite apparent. Tests have shown that wheels of this type, in comparison with rigid wheels of equal diameter, provide a 50 percent decrease in rolling resistance, an increase of about 40 percent in drawbar pull, and demonstrated an impressive ability to climb obstacles. With respect to obstacle climbing ability, flexible metal wheels reportedly climbed obstacles three-quarters the size of the wheel's relaxed diameter (Ref. 10). A conventional, rigid wheel can normally climb an obstacle equal to about one-third of its height.

Cleats, or grousers, are usually added to the outer periphery of the flexible rim to increase traction. The spaced-link principle, discussed elsewhere in this handbook, is applied to distinct advantage in spacing the grousers.

Spring rates of elastic wheels have been calculated between 20 and 100 pounds per inch of vertical hub deflection. When this is compared to spring rates between 1,000 and 5,000 pounds per inch for standard rigid wheels, it shows that elastic wheels will give a soft, smooth ride.

SECTION III SKIS, SLEIGHS, AND TOBOGGANS

6-9 GENERAL DISCUSSION

Skis, sleighs, and toboggans are a category that is often overlooked in literature dealing with land transportation despite the fact that many areas exist where snow covers the earth's surface for considerable portions of the year. Furthermore, these types of vehicles find use in providing transportation for men and equipment over extremely wet, swampy terrain where other forms are not efficient or are entirely ineffective. Aside from towed vehicles of the toboggan or bobsled types, self-propelled vehicles utilizing ski or sled principles can be ski-borne air screw-propeller machines, motor sleds, or ski-track machines.

Ski-borne air screw-propeller machines have certain inherent drawbacks which limit their range of application to services where extremely lightweight machines can be used or to areas where snow conditions and levelness of the terrain limit the maximum thrust-to-weight ratio required to approximately 0.2 (Ref. 11). However, Dr. Kamm (Ref. 12) recommends a minimum thrust-to-weight ratio of 0.5 for general all-around service where reasonable slope-climbing ability and reserve power

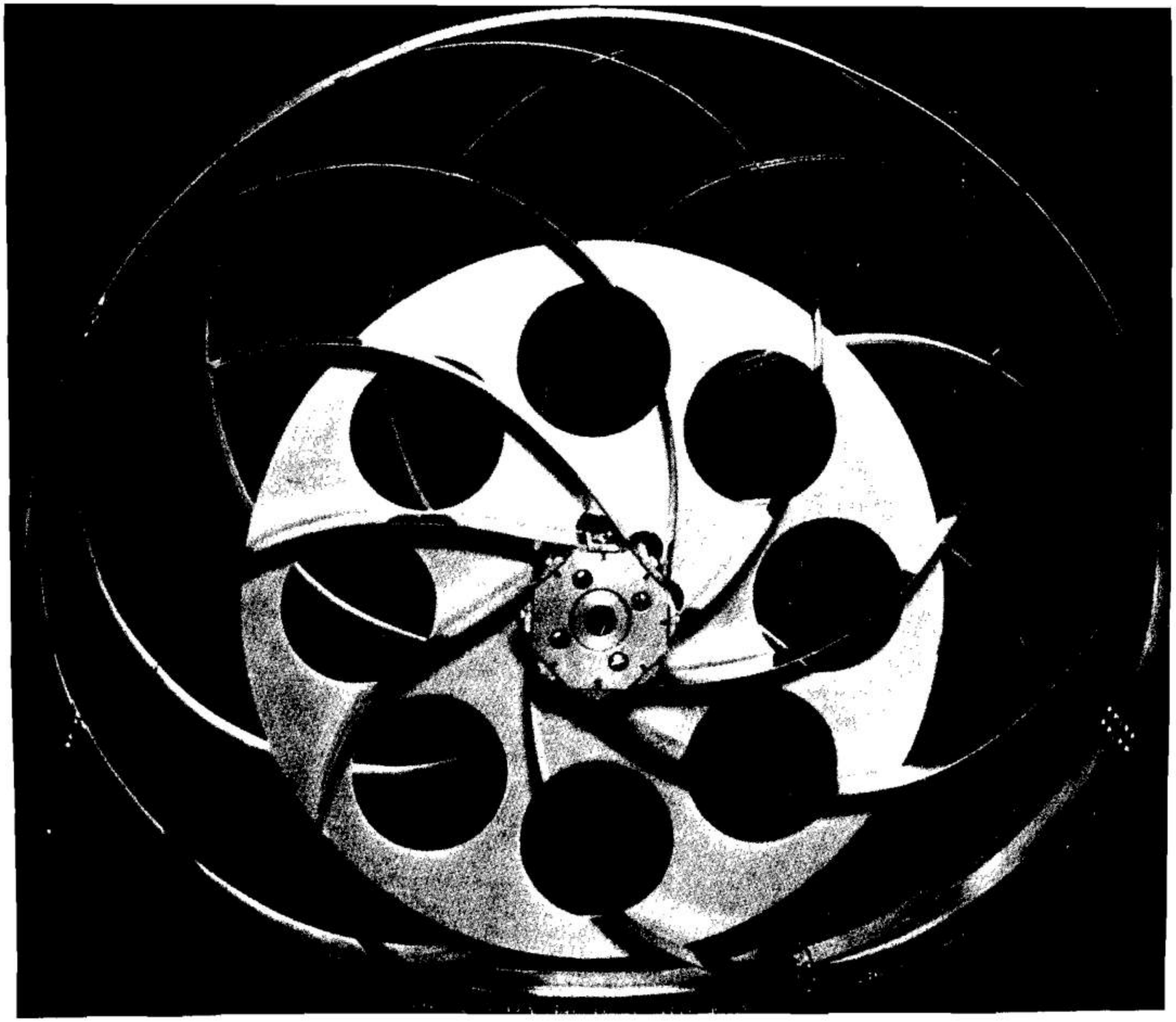

Figure 6-21. Metalastic Wheel

to cope with freeze-down after parking are a requirement, suggesting further that a value of 0.7 would be even more desirable. This would require the installation of approximately 350 horsepower per ton of gross vehicle weight, which is not practical for a military vehicle because of the prohibitive rate of fuel consumption. Under favorable operating conditions, a vehicle with such a high power-to-weight ratio would have an extremely high speed potential and would, therefore, have to be carefully designed as to aerodynamic stability and its suspension characteristics.

Motor sleds are generally lightweight, cargo-carrying bobsled- or toboggan-type vehicles propelled by a small, floating track unit which projects downward through an opening in the rear sliding surface. The track unit is intended for propulsion only, and toward this end, carries about

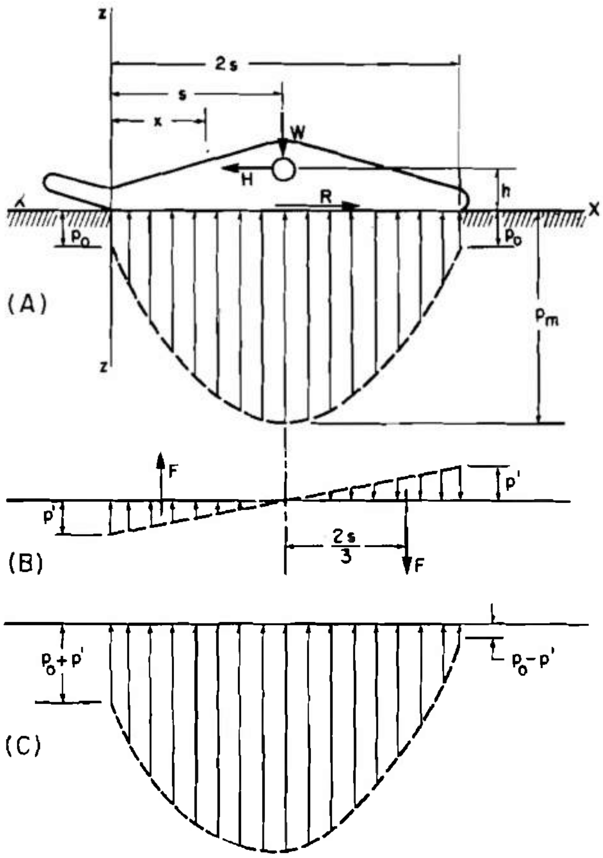

Figure 6-22. Pressure Distribution Under Ski

20 percent of the vehicle gross weight. Since the propulsion track is relatively lightly loaded, its tractive effort is limited. It is important, therefore, that motion resistance be reduced to a minimum by designing the runners for minimum sinkage. The extremely low unit pressures necessary for this in light, powdery snows restricts the expansion of this type of vehicle into a true load-carrying machine suitable for heavy military cargo applications.

The ski-track machines are functional opposites of the motor-sleds. Here the bulk of their weight is carried on the tracks while the skis carry only sufficient weight to provide effective steering. This is usually about 20 to 30 percent of the gross weight on the skis, depending upon the relative moment arms of the skis and tracks. Thus, the ski-track machines are essentially tracked vehicles with ski-steering substituted for skid-steering. This provides more stable directional performance and reduces the digging-in of tracked vehicles in soft snow, which is a characteristic of skid-steering that can cause loss of steering and even complete immobilization of the vehicle. Provisions for skid-steering, in addition to ski-steering, is considered desirable, however, as it assists in close maneuvering and gives better control on hard packed snow and ice.

6-10 PRESSURE DISTRIBUTION AND SKI AXLE LOCATION

The pressure distribution under a ski or a sled runner can be discussed best by referring to Figure 6-22. Illustrated is a ski whose length in contact with the snow is $2s$, and which is connected to the vehicle at one suspension point located above the center of the effective length as shown. The ski cross section is rectangular and of a constant width but the depth of the ski increases toward the midpoint in such a manner as to maintain a modulus of rigidity proportional to the bending moment produced by the snow reactions. Under these conditions, it can be shown (Ref. 13) that the pressure distribution under the ski will be parabolic in nature. During conditions of static loading, the pressure distribution will be approximately as shown in Figure 6-22(A). The pressures at the ski ends p_o and under the midpoint p_m will be

$$p_o = \frac{W}{2sb} - \frac{c'ks^2}{3}\text{, psi} \qquad (6\text{-}8)$$

$$p_m = p_o + \frac{c'ks^2}{2}\text{, psi} \qquad (6\text{-}9)$$

and the pressure at any point, x, along the ski will be

$$p = \frac{W}{2sb} - \frac{c'ks}{3} + \frac{c'k}{2}(2sx - x^2),\ \text{psi} \qquad (6\text{-}10)$$

where W = gross weight on the ski, lb

s = one-half of the effective ski length, in. (see Figure 6-22(A))

b = width of ski, in.

$c' = \dfrac{M}{EI}$, in.$^{-1}$

M = total bending moment at section under consideration, in.-lb

TABLE 6-1
SOME TYPICAL CHARACTERISTICS AND (ESTIMATED) CORRESPONDING MECHANICAL PROPERTIES OF SEVERAL GENERALIZED SURFACE SNOW TYPES

A. DESCRIPTIVE CHARACTERISTICS (Roughly averaged)

Snow Type	Grain Form	Grain Size	*In Situ* Specific Gravity	Free-water Content, %	Degree of Bonding	Normal Maximum Depth, ft
NEW SNOW						
1. Wild	Fluffy, dendritic	Fine	0.03	0	Some crystal interlocking	6
2. Ordinary dry	Dendritic	Fine	0.06	0	Some crystal interlocking	4
3. Wet	Dendritic to amorphous	Medium	0.15	10	Some capillary cohesion	2
SETTLING SNOW						
4. Powder	Compact dendritic	Fine	0.2	0	None	3
5. Bonded	Compact dendritic	Fine	0.2	0	Some ice bridging	3
6. Wet	Dendritic to amorphous	Medium	0.2	10	Capillary cohesion	2
7. DEPTH HOAR	Cup crystals	Medium to large	0.2	0	None	3
8. BARRENS "SAND" SNOW	Saltlike	Fine	0.3	0	None	2
SETTLED SNOW						
9. Powder	Saltlike	Fine to medium	0.4	0	None	2
10. Bonded	Saltlike	Fine to medium	0.4	0	Ice bridging	2
11. Wet	Saltlike to amorphous	Medium to large	0.4	10	Some capillary cohesion	2
OLD SNOW, FIRN, OR NÉVÉ						
12. Disaggregated	Almost spherical	Medium to large	0.5	0	None	2
13. Bonded	Almost spherical	Medium to large	0.5	0	Some ice bridging	2
14. Wet	Spherical	Large	0.5	15	None	2
15. SPRING, CORN, OR SUGAR SNOW	Spherical	Large	0.6	20	None	2
16. ROTTEN SPRING SNOW OR SLUSH	Spherical	Large	0.7	30	None	2
17. ICE	—	—	0.9	0	Full	—
18. FROZEN MUSKEG	—	—	1.1	0	Full	—

E = modulus of elasticity of the ski material, psi

I = the moment of inertia of the ski section, in.^{4}

k = a snow sinkage parameter discussed below, lb/cu in. (for $n = 1$)

n = exponent of deformation (see below), dimensionless

The snow parameter k, appearing in the preceding equation, relates the sinkage z in inches, of an object in snow to the unit load p in psi, imposed upon it, as $p = kz^n$. This is the same relationship discussed in Chapter 4 with respect to sinkage of loaded areas in soils. Despite the fact that snows do not react the same way under bearing loads as do soils, the $p = kz^n$ relationship (popularly referred to as the Bernstein Equation) does appear to be usable in snow and seems to be the best expression of the relationship of sinkage to load available at this time. Thus, $k = \frac{p}{z^n}$, (lb/in.$^{n+2}$, in general) and for most types of snow $n = 1$ (see Table 6-1). Therefore, for reasonably accurate approximations, k can be equated to $\frac{p}{z}$ (lb/cu in.); and the z parameter can be determined from suitable tests. In this respect, it should be recalled from the discussions of paragraphs 4-4.1 and 4-4.2 of Chapter 4 that sinkage is a function of the geometry, size, and properties of the loading area as well as of the strength and compactive

TABLE 6-1 (cont'd)
SOME TYPICAL CHARACTERISTICS AND (ESTIMATED) CORRESPONDING MECHANICAL PROPERTIES OF SEVERAL GENERALIZED SURFACE SNOW TYPES

B. STRUCTURAL OR STATIC MECHANICAL PROPERTIES (Mostly estimated)

Snow Type	Principal Behavior Under Bearing Loads	Temp, °C	Shear Strength, psi	Tensile Strength, psi	Linearized Shear-normal Stress Parameters		"Open Field" Bernstein Parameters		Uniaxial Compression Parameters		Critical Density, g/cc	Cone Index
					c, psi	$\tan\phi$	k, lb/in.$^{n+2}$	n, dimensionless	K, psi	N, dimensionless		
NEW SNOW												
1. Wild	Compacts	−10	0	0	0	2	0.01	1	2	2	0.4	0
2. Ordinary dry	Compacts	−10	0.2	0.3	0.1	2	0.1	1	6	2	0.4	0
3. Wet	Compacts	0	0.3	0.4	0.1	1	0.2	1	7	2	0.5	0
SETTLING SNOW												
4. Powder	Compacts	−10	0	0	0	1	1	1	30	2	0.4	0
5. Bonded	Compacts	−10	0.5	0.7	0.3	1	1	1	35	2	0.4	10
6. Wet	Compacts	0	0.5	0.7	0.4	0.7	2	1.5	40	3	0.5	10
7. DEPTH HOAR	Shear flow	−10	0	0	0.1	0.3	0.2	1	25	3	0.3	0
8. BARRENS "SAND" SNOW	Shear flow	−20	0	0	0	0.4	0.5	1.5	100	3	0.4	0
SETTLED SNOW												
9. Powder	Compacts	−10	0	0	0	0.5	5	1.5	500	3	0.45	10
10. Bonded	Compacts	−10	1.0	1.2	0.4	0.5	10	1.5	600	3	0.45	30
11. Wet	Compacts	0	0.2	0.2	0.1	0.4	5	2	700	4	0.55	30
OLD SNOW, FIRN, OR NÉVÉ												
12. Disaggregated	Shear flow	−10	0	0	0	0.4	1	1	500	3	0.5	10
13. Bonded	Hydro flow	−10	5	7	2	0.4	20	1.5	600	3	0.5	50
14. Wet	Hydro flow	0	0	0	0	0.3	2	1	60	2	0.5	10
15. SPRING, CORN, OR SUGAR SNOW	Hydro flow	0	0	0	0	0.2	0.1	1	100	2	0.6	0
16. ROTTEN SPRING SNOW OR SLUSH	Hydro flow	0	0	0	0	0.1	0.02	1	—	2	0.7	0
17. ICE	Elastic	−10	100	200	100	0		$E = 1.5 \times 10^6$			0.9	—
18. FROZEN MUSKEG	Elastic	−10	—	100	—	—	—	—	-	—	1.1	300

properties of the snow. Furthermore, the range of loads and method of load application should correspond closely to the actual snow-vehicle conditions anticipated.

Nuttall and Finelli (Ref. 11) have compiled a table in which they list 18 generalized types of snows and snow conditions, including ice and frozen muskeg, for which they give various physical characteristics and estimated values of pertinent mechanical properties. The data were collected mostly from technical literature and their validity and accuracy were evaluated only by methods of cross-comparison, interpolation, and intuition. It was intended as a general guide to the magnitude and ranges that can be encountered in different types of snows and "to indicate the type and form of presentation of snow information that might be desired by, and be useful to, vehicle designers." This table is reproduced here as Table 6-1 and can be used in the absence of more accurate data. It should be noted that the values given for mechanical properties apply to static loading conditions. Information on dynamic properties of snow is very scarce.

The Unaxial Compression Parameters K and N, given in Table 6-1, are soil parameters similar to the Bernstein parameters k and n which are obtainable from a relatively simple test. A snow sample is confined in a cylinder and subjected to slowly increasing axial loads. Its change in height Z with respect to its original height L is recorded at various loads. Wall friction is minimized by gently heating the cylinder walls, and errors which might be introduced by this procedure are minimized by using a cylinder of fairly large diameter. Tests have shown that the experimental pressure-sinkage curve is approximately of the form

$$p = K\left(\frac{Z}{L}\right)^N \tag{6-11}$$

The Critical Density given in Table 6-1 is the density of compacted snow beyond which it will not compact further by the normal order of vehicle loadings. Below this density, the snow will absorb compaction stresses by compacting further; above this density, the same snow, when it fails under bearing loads, will fail by flowing more-or-less hydrodynamically, or by the classical general shear failure discussed for soils.

Equations 6-8 to 6-10 apply to a stationary ski. In order for the ski to move forward, a force H must be applied at the hinge to overcome the resistance force R acting on the ski. Since the hinge point is at some height h above the snow-ski interface, an overturning couple is created equal to Rh (Figure 6-22(A)). This results in a gradually increasing ground pressure on the forward half of the ski and a corresponding decrease on the trailing half, the maximum values of p' occur at the ends of the ski as shown in Figure 6-22(B). If the assumption is made that the ski is rigid

$$F = \tfrac{1}{2}\, p' s\, b, \text{ lb}$$

and from the equilibrium of moments

$$\frac{4Fs}{3} = Rh$$

and combining these two equations results in

$$p' = \frac{3Rh}{2s^2 b}, \text{ psi} \tag{6-12}$$

Thus, the pressure beneath the front p_f and rear p_r of the ski in psi will be

$$\left.\begin{aligned} p_f &= p_o + p' = \frac{W}{2sb} - \frac{c'ks^2}{3} + \frac{3Rh}{2s^2b} \\ p_r &= p_o - p' = \frac{W}{2sb} - \frac{c'ks^2}{3} - \frac{3Rh}{2s^2b} \end{aligned}\right\} \tag{6-13}$$

In order to equalize the pressure distribution under a moving ski, the axle, or hinge point, is displaced rearward from the center of the ski to offset the Rh moment. This is a particularly useful arrangement when applied to high speed, propeller driven sleds or ski-mounted aircraft where it creates a lift component similar to that of a hydrofoil or aircraft wing. If the resistance R acting on the ski is known, the optimum location of the ski axle can be calculated. The load distribution of an elastic ski will never be uniform and, particularly when very flexible runners are used, the difference between p_m and p_o will be very large indeed.

6-11 LOAD-BEARING CAPACITIES OF SKIS AND TOBOGGANS

For the purpose of this discussion, the distinction between a ski (or a sled runner) and a toboggan is that the ski is quite narrow with respect to its length and develops a base for its support in snow through sinkage and the resulting snow compaction. The toboggan, although usually also narrower than it is long, has a much higher width-to-length ratio. Its chief application is in handling heavy loads over swampy, plastic ground; and, therefore, strives for maximum flotation through minimum sinkage. In order to understand the principles governing these behaviors, it is necessary to refer to some basic theory.

Bekker (Ref. 14) wrote a general equation for the bearing capacity of an ideal rectangular strip of soil (or snow) of width $2l$ and length s in the form below. This equation is merely a rewritten version of a classic soil mechanics theory propounded by Terzaghi (Ref. 15).

$$W = 2ls\,(cN_c + qN_q + \gamma l N_\gamma) \tag{6-14}$$

where W = maximum bearing load that can be supported by the strip, lb

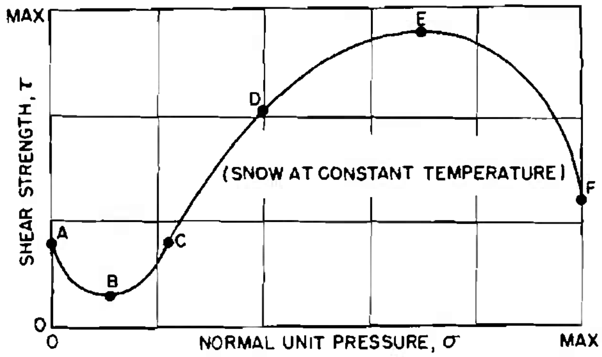

Figure 6-23. Typical Curve Showing Relationship Between Shear Strength and Normal Pressure for Snow

q = surcharge loading, psi
c = soil cohesion factor, psi
γ = soil density, lb/cu in.
l and s are as defined above, in.
N_c, N_q, N_γ = bearing capacity factors which are functions of the internal friction φ of the or snow, dimensionless

Along with equation, it is necessary to be aware of the rather unique variation of the shear strength τ of snow when subjected to a gradually increasing compressive stress σ at constant temperature. This relationship is shown by the "lazy S" shaped curve of Figure 6-23. This behavior of snow is explained by the effects of initial compaction (curve from point A to B), changes in snow structure (points B-C-D), metamorphosis into ice (D to E), subsequent plastic flow (E to F), and finally the complete liquefaction of the snow (at F). The A-B-C portion of the curve is obtained at very low pressures (less than 1 psi) and, therefore, has little useful significance to a vehicle designer. The portion C-D-E represents pressures σ from approximately 1 to 4 psi. Furthermore, experiments have shown that this portion of the curve can be reasonably approximated by the classic soil mechanics equation

$$\tau = c + \sigma \tan \varphi \qquad (6\text{-}15)$$

Thus, if the curve C-D-E can be determined experimentally for a particular snow, the tangent to the curve at any given pressure σ will determine the corresponding approximate value of c and φ. These can then be applied to Equation 6-14 to evaluate the maximum bearing capacity of that particular snow.

Soil and snow characteristics are such that the lower the unit compressive load σ, the larger is the internal friction angle φ, and the smaller is the cohesion c. Thus, at low compressive loads σ, the bearing capacity increases faster with increases in ski width $2l$ than for increases in ski length s because the value of φ, and hence N_γ, is relatively large, making the $\gamma l^2 N_\gamma$ term of Equation 6-14 have a significantly large influence upon the value of W. At large compressive loads, the curve levels off causing φ to become quite small. The cohesion c becomes so large under these conditions as to make N_γ ineffectual and the bearing capacity is essentially proportional to the surface area $2ls$ irrespective of its form. Thus, under heavy loads, a long narrow runner in snow will support the same safe load as a short wide one without causing general or local shear failure if their areas are equal.

Where the sled will operate over soft, plastic soil masses such as slushy, wet, plastic snow resting on a hard base, or supersaturated clay on a hard substrate (where $\varphi = 0$), Equation 6-14 does not apply. Under these conditions, the following modified form of this equation seems more appropriate in estimating the maximum safe load W.

$$W_c = 2lscN_ck_s \qquad (6\text{-}16)$$

where k_s is an arbitrary safety factor. Reasonable correlation has been demonstrated between values of W_c calculated by Equation 6-16 and experimental results in wet, plastic, slushy snow resting on a hard base.

6-12 TOTAL RESISTANCE TO SLIDING

The task of predicting the total resistance of a ski R_s on snow or ice presents complex problems which are not yet thoroughly understood. Bekker (Ref. 13), in summarizing much of the work that has been done in this field, makes use of a composite coefficient of sliding friction μ_s comprised of the sum of the coefficients of snow compaction μ_c capillary drag μ_r, viscous drag μ_v, and dynamic resistance μ_d, such that

$$R_s = W\mu_s \qquad (6\text{-}17)$$

and

$$\mu_s = \mu_c + \mu_\tau + \mu_v + \mu_d \quad (6\text{-}18)$$

where W = gross weight per ski.

The *coefficient of snow compaction* is a function of the sinkage z and the ski length s (the length of ski-snow contact area) in the following relationship

$$\mu_c = \frac{z}{s(n+1)} \quad (6\text{-}19)$$

where n is the Bernstein sinkage exponent discussed in Chapter 4 and in paragraph 6-10 of this chapter. Thus, when

$$n = 1;\ \mu_c = \frac{1}{2}\ \frac{z}{s} \quad (6\text{-}20)$$

$$n = \frac{1}{2}\ ;\ \mu_c = \frac{2}{3}\ \frac{z}{s} \quad (6\text{-}21)$$

$$n = 0;\ \mu_c = \frac{z}{s} \quad (6\text{-}22)$$

Capillary drag, whose coefficient is represented in Equation 6-18 by μ_τ, results from capillary forces occurring in the minute droplets which make up the lubricating film of water between the moving ski and the snow surface. This drag is significant only when the droplets are exceedingly small (less than 0.001 inch in diameter). The maximum capillary drag coefficient is constant at ski speeds above 0.4 mm per second, assuming that all conditions are ideal for the development of an increasing number of droplets of decreasing diameter to support the load. Under these conditions, the maximum coefficient of capillary drag becomes

$$\mu_\tau = 0.096 \quad (6\text{-}23)$$

The size and number of water droplets between the ski and snow depends upon several factors such as load, temperature, and frictional conditions of the ski and snow. As the droplets become large, the capillary forces become negligible. More research is required on this subject to correlate the many factors involved.

While capillary forces of the water droplets might be insignificant, their viscous forces may not be. The *coefficient of viscous drag* μ_v may be assessed by the following equation:

$$\mu_v = \eta\ \frac{v}{4\tau_s} \quad (6\text{-}24)$$

where η = viscosity of the water, poises (g/cm-sec)
v = ski velocity, cm/sec
τ_s = surface tension of the water, g/sec^2

Water at 32° F has a η of 1.83×10^{-2} poises and a τ_s of 70 g/sec^2. When these are substituted into Equation 6-24, the expression becomes

$$\mu_v = 6.5 \times 10^{-5} v \quad (6\text{-}25)$$

The *dynamic resistance* results from the planing aspect, or trim, of the ski as it moves over the snow with its fore-end raised higher than its rear due to either an accidental load distribution or from the deliberate location of the axle toward the rear of the ski. The planing area thus established may produce a significant drag and a certain amount of lift, particularly at speed in excess of 50 miles per hour. The coefficient of dynamic resistance μ_d can be calculated as

$$\mu_d = \tan\alpha \quad (6\text{-}26)$$

where α is the angle which the longitudinal axis of the ski makes with the surface of the snow.

Considerable experimental work is needed in the field of snow-ski relationships. A great deal of information found in the literature is controversial and contradictory. The information given in this section is based upon theory and seems reasonable. It is, therefore, offered as a guide until better information becomes available.

SECTION IV WALKING AND LEAPING SUSPENSION CONCEPTS

6-13 JUSTIFICATION AND BACKGROUND

Some comparisons are made in Chapter 3 of the effectiveness of wheeled and tracked vehicles in off-the-road, cross-country travel and that of animals. It is pointed out that the evolution of animal forms has resulted in configurations that are far better suited to cross-country movement in their respective environments than are those of man-made vehicles; and the principal similarity among these natural cross-country locomotors, it is pointed out, is the total absence of the wheel. Various advantages of walking, running, and leaping concepts are discussed along with a brief narrative of the evolution of wheels in a direction that has made them less suitable for off-road travel.

These advantages of animal mechanics have not gone entirely unnoticed by engineers; in fact, they have been considered very seriously by the formulators of concept vehicles for future travel over the moon's surface (Refs. 16 and 17). Since much of our present knowledge of lunar soil mechanics is based upon imagination and conjecture, these engineers sought to give the lunar vehicle every advantage known; thus indicating their recognition of the shortcomings of wheels. It is ironic that such a fundamental concept for terrestrial locomotion as walking machines was first seriously considered for use on the moon despite its prolific use on earth in nature. Designers quickly realized, however, that a true walking machine is a highly complex mechanism which cannot be developed overnight. Despite the fact that nature has provided many working examples, the development of practical mechanical counterparts has long been neglected. Thus, the visionary designers of lunar vehicles, for the most part, were forced to abandon walking concepts in preference to easier-to-attain wheeled configurations.

Until recently, there have been few attempts at developing walking machines. What machines have been made have been crude devices that didn't really walk, since they lacked many of the characteristics usually associated with true walking; but, since they were neither wheeled nor track-laying devices, they are classified as walking mechanisms for want of a better classification. One example of these was a walking barge. This was a large (28 ft $\times$ 60 ft $\times$ $9\frac{1}{3}$ ft deep) amphibious barge capable of carrying a payload of 35 tons. It had a large center pontoon (16 ft wide) which was moved forward and back through an oval path to motivate the vehicle on land. Steering, on land, was accomplished by lowering steering pads located on both sides at the rear of the vehicle. In this manner, the vehicle was able to "walk" on firm ground with a speed of $\frac{1}{2}$ mile per hour. Over soft to muddy terrain the speed varied downward from $\frac{1}{2}$ mile per hour to one mile per day.

Other examples of the walking concept can be found applied to very large power shovels and drag lines used in open-pit mining operations. These move at extremely slow speeds (on the order of $\frac{1}{8}$ mile per hour), but they are monstrously large machines weighing upward of 1800 tons. In general, they are walking mechanisms based upon mechanical linkages and eccentrics, although hydraulic mechanisms are also employed.

An interesting walking mechanism using hydraulically powered legs was applied to a large coal conveyor by the Weserhuette Otto Wolff GmbH. of Bad Oeynhausen, Germany (Ref. 18). It consists of hydraulically operated support cylinders and hydraulic control cylinders which swing the retracted support of cylinders individually in the direction of travel. The novelty of this system, as it is designed, is that the vehicle is able to move in any direction (forward, backward, sideward, obliquely) as well as to rotate in one place.

A small, lightly armored, one man, amphibious combat vehicle was once proposed to the Ordnance Tank-Automotive Command. One version of this vehicle was a walking concept which made use of eight legs (four on each side of the vehicle). A quarter-size working model was constructed which demonstrated that the principle was workable. The vehicle walked and swam quite well. One feature of this vehicle was that it did not become immobilized when one or more of its legs were damaged, as by a land mine, provided that the undamaged legs were properly located with respect to the vehicle's center of gravity. It was reported that, with selected legs remaining operable, the vehicle could continue to travel on a minimum of three legs.

During World War II, various concepts of jumping vehicles were investigated. Two mock-ups were constructed, one of an eight-wheeled vehicle with a 189-inch wheelbase and a 121-inch width. Its total weight, including two engines and a 3-inch gun, was estimated at under 20,000 pounds. The other mock-up was of a four-wheeled version designed to carry a gun of about 40 mm. It had a wheelbase of 137 inches, a width of 121 inches, and weighed under 10,000 pounds. The jumping mechanism was hydraulically operated and was designed to accelerate the vehicle vertically while the vehicle's forward momentum would carry it over an obstacle. The sequence of events described in the paragraph below constituted the jumping cycle.

While moving forward, the operator would actuate the mechanism to cause the sprung mass to squat on its suspension. At the proper instant, he would release the jumping mechanism to accelerate the sprung mass upward. The unsprung mass would follow the sprung mass upward, due to the transfer of momentum, which would place the entire vehicle in mid-air. Upon returning to the ground, the jumping cylinders would absorb the landing shock and return the sprung mass to its normal ground clearance.

A full scale wheel unit, with drive, suspension, hydraulic jumping cylinder, and adjoining frame members, was constructed and tested. In one jump, the wheel cleared a height of more than 47 inches. For a vehicle with a 137-inch wheelbase traveling at 40 miles per hour, this would be the equivalent of jumping a distance of 47 feet. Thus, the tests showed that the concept was fundamentally sound. No pilot model was ever constructed.

6-14 SOME CURRENT STATE-OF-THE-ART

The requirements of modern warfare do not permit the use of roadbound vehicles, nor do they tolerate the marginal level of performance achieved in difficult terrain by cross-country vehicles whose design is based upon conventional wheeled and tracked concepts. Military tacticians speak of off-road speed requirements that are far in excess of those that seem feasible with conventional methods. This has led to new developments in fixed- and rotary-winged aircraft and in air-delivery systems.

But taking to the air is not a solution to the land transportation problem; it is an avoidance or circumvention of it. This would be an acceptable solution if it met all the tactical requirements but it does not. Since the ability to move on, and in contact with, the land is still a requirement, the flying machines end up by transporting the land machines over the difficult terrain and permitting them to operate on their own where the terrain is favorable. Thus, the aircraft perform a supporting role, and in doing so, greatly increase the logistic requirements of the operation.

It is quite possible that walking machines may be the eventual solution to the attainment of relatively high off-road speeds in difficult terrain. In pursuit of this concept, the Land Locomotion Research Laboratory of the U.S. Army Tank-Automotive Center has supported several formal studies of this problem. Among the first of these was the work of Professor R. K. Bernhard of Rutgers University (Ref. 19) wherein he analyzed the running, leaping, galloping, bouncing, and jumping motions from their theoretical aspects for application to vehicular propulsion. He found that a bouncing motion, similar to that of a grasshopper, could result in relatively high speeds although this type of motion was hampered by unpleasant shock and vibration problems and by problems of control. Similar problems hampered the leaping and galloping concepts. The running concepts were the least attractive because of their low speed potential, but this was offset by greatly reduced control and vibration problems. He concluded, therefore, that running concepts might be the most likely to succeed.

Another noteworthy study in this field was that of Professor Joseph E. Shigley of the University of Michigan (Ref. 20). This work is perhaps the most complete study, to date, of the mechanics of walking vehicles. In it, he established criteria for the evaluation of walking machines and points out the problems that must be solved in developing a practical machine. These are sufficiently important to be enumerated here. An ideal walking mechanism, according to Shigley, is one which meets the following requirements:

(a) The machine must have a uniform velocity while the "feet" are in contact with the ground.

(b) The stride must be long in relation to the physical dimensions of the walking machine to achieve adequate speeds.

(c) The height of the step (return stroke of "foot") should be large in comparison with the dimensions of the walking machine.

(d) The height and length of the stride must be controllable by the operator.
(e) The stride time to return time ratio of the "feet" should be high. This means that the "foot" will be in contact with the terrain during a major portion of the operating cycle.
(f) A mechanism integral with the "legs" must be provided for steering the vehicle.
(g) The vehicle must be capable of moving in either the forward or the reverse directions.
(h) The inertia forces and inertia torques acting on the vehicle must be balanced.
(i) The energy used in lifting the "foot" at the end of a stride should be recovered in lowering it at the beginning of the next stride.
(j) The height of the "body" of the walking machine should be controllable by the opoperator.

In his study, Professor Shigley investigated many types of mechanisms which are discussed in his report. Subsequently, working models of the most promising of these were constructed and operated at the Land Locomotion Research Laboratory of the U.S. Army Tank-Automotive Center. No full sized vehicle was built based on any of these concepts. The ideal walking vehicle concept proposed by Professor Shigley had sixteen legs arranged in groups of four legs with one group at each corner of the vehicle. The problem of controlling the sixteen legs as to length, height, and speed of stride—all in different amounts and at different times as the vehicle moves over rugged terrain and various obstacles—has not yet been solved.

A current program aimed at the development of a walking vehicle with a practical solution to the control problem proposes a vehicle comprised of multiple two-legged units each controlled by a human operator aboard the unit. The legs of each unit would be slaved to the legs of the human operator through a force feed-back servo system similar to those of the already successful remote manipulators designated as cybernetic anthropomorphous machines (acronym, CAM). When the human operator moves his foot, the mechanical foot will do likewise. When the human presses his foot down and back, the machine will follow and proceed forward, while the operator will feel the reflected force of the weight of the machine. Thus, the gait of the machine will be governed by the natural gait of the human operator.

Similarly, the balance of the machine will be sensed by the human operator. As the operator shifts his center of gravity to restore his equilibrium, the machine will follow with an analogous movement. In an extreme case, the operator can take corrective action by moving his foot in the direction of teeter, and the machine will respond by moving its foot comparably.

A machine of this type would, obviously, have no place on a road, nor could it compete with modern road vehicles from speed considerations alone. It has been calculated that the fastest that a human operator can move his legs for extended periods is about 6 miles per hour. If the walking vehicle is assumed to have legs 18 feet long as compared to the man's 3-foot-long legs, the vehicle could move about 35 miles per hour. This is obviously slow for highway travel, but is an enviable speed across difficult terrain.

The ability to maintain balance and an erect posture on two feet is the initial, and most difficult, phase of walking. To reflect briefly, an infant can walk, if he is supported in an upright position, long before he is able to balance himself unaided. Walking, therefore, is relatively simple—standing and balancing, however, are much more difficult.

In an attempt to evaluate the feasibility of the concept of a two-legged walking machine controlled by the natural walking motions and reactions of a human operator, a balance demonstrator was developed for the Land Locomotion Research Laboratory of the U.S. Army Tank-Automotive Center, Warren, Michigan. Since the ability to control balance and postural attitude while standing in place is a significant factor in the functions required for walking, the successful demonstration of this capability in a machine was the objective of the balance demonstrator.

The primary body motion involved in maintaining balance while standing still is forward and backward body sway about the ankle axis. Therefore, the demonstrator was built with 18-foot legs, the feet fixed to the floor, and provisions were made to permit forward and backward swaying motions about an ankle axis. An operator's cab is mounted above the legs. This is provided with a pedal platform upon which the operator stands and a harness frame for his back. The cab can tilt forward and back with respect to the legs. The platform and ground surface (or foot) remain parallel while the tipping angles change. Ankle motions are controlled by forces exerted upon the toe or heel of the pedal platform, while the action of the cab is controlled by the back harness. The operator feels the forces that he generates or encounters as he maintains his balance upon the platform. Proportions of these forces are transmitted by means of hydraulic servomechanisms to the ankle and cab pivots to maintain correct spatial correspondence between operator and machine.

Test results have shown that: (a) unpracticed subjects can easily and effectively balance the demonstrator after a maximum of 5 minutes' indoctrination and practice; (b) the operator adapts quickly to the balancing as a natural task requiring little attention; (c) the operator can effectively perform other tasks while balancing the machine; and (d) the control concept used will allow the operator to effectively perform the series of motion controls required for a complete walking machine.

The results of studies to date indicate that this concept is feasible from a design viewpoint and that no insurmountable human factors problems exist. The creation of a working prototype is pending the availability of funds. It is conceivable that walking machines may some day become a reality and will find use in achieving a high level of off-road performance for unusual, high value payload missions over terrain that is impassable for conventional vehicles.

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CHAPTER 7

LABORATORY AND FIELD TEST TECHNIQUES, EQUIPMENT, AND INSTRUMENTATION*

SECTION I METHODS OF SOIL ANALYSIS

7-1 IDENTIFICATION OF SOIL CONSTITUENTS

7-1.1 MINERAL TYPE AND CONTENT (Ref. 1)

Soil is a heterogeneous accumulation of uncemented or loosely cemented grains enclosing voids of varying sizes. These voids may contain air, water, or both, and/or organic matter in various amounts. Generally, the solids in natural soil formations have resulted from the disintegration of rocks and are, therefore, of mineral composition (principally silicates of aluminum). Coarser soils contain mainly particles of the primary minerals while clay particles are composed of secondary minerals (minerals resulting from chemical alteration of primary minerals) developed during soil formation. Primary minerals have no planes of marked weakness in any one direction. Consequently, particles of the primary minerals are predominantly bulky in shape. Secondary minerals have planes of weakness and are subject to what is known as basal cleavage. As a result, particles of the secondary minerals are of platelike shape. Bulky particles, as a rule, are found in the coarse-grained soils; namely, gravel, sand, and silt, while platelike particles predominate in most clays.

Properties of minerals such as solubility, hardness, and resistance to wear and to crushing are important. Also important is the base exchange capacity of clay minerals. The mineralogical content often determines the shapes of the individual soil particles. Quartz and feldspar are bulky, whereas, mica and alumina are flat in shape. The presence of only a relatively few flat-shaped grains greatly affects the porosity of a soil. Flat grains also have an important effect on the plastic properties of soil.

Organic material in soil is derived from the disintegration of plant or root growth and consists of muck or fibrous material such as peat. Soils are roughly classified as organic or inorganic, depending on whether or not they contain significant amounts of organic material. The presence of organic material is important to engineering application of the soil because even small quantities may significantly influence soil compressibility.

7-1.2 PARTICLE SIZE AND SHAPE

Particle size is expressed in terms of a single particle dimension, customarily considered to be the diameter, even though most particles are not spheres. For larger particles, the dimension is expressed as the size of the smallest square hole through which the particle will pass. For example, a parallelepiped with dimensions of 6 mm × 3 mm × 2 mm would not pass through a 2 mm × 2 mm hole but would pass through a 3 mm × 3 mm hole. Thus this particle would be said to have a particle diameter of 3 mm.

The ordinary mineral particles of which soils are composed range in size from about 3 inches to 0.01 micron which includes the gravel (3 inches to 2 mm), sand (2 mm to 0.074 mm), silt (0.074 mm to 0.002 mm), and clay (less than 0.002 mm) fractions. Particles of a size of 2 microns and smaller are considered to be of colloidal nature. The size fractions cited above will vary slightly with the textural classification used. Also, it should be noted that particle size is not a sufficient criterion for distinguishing between silt and clay. Plasticity is a unique property of clays. Therefore, regardless

* Written by William B. Truesdale and Dr. Delon Hampton of the IIT Research Institute, Chicago, Ill.

of particle size, if a soil possesses plasticity it is a clay.

Soil particles also vary in shape. Two major distinctions as to shape are generally recognized to be important in engineering, although there are many subdivisions. On the one hand are particles whose three dimensions are of the same order of magnitude; these are described as being bulky in shape. Many of these bulky particles are approximately spherical. The opposite extreme of shape is exemplified by particles of platelike shape; many of which have a thickness less than one-hundredth of their diameter. This extreme difference in shape is a reflection of a difference in the crystal structure and characteristics of the mineral of which the particle is composed.

The effect of particle size and shape on the engineering properties can be better understood by considering the magnitude of the surface area to volume ratio as a function of particle size. For example, visualize a cube of any material whose sides are 1 cm long. The volume of such a cube is 1 cc; the total surface is 6 cm^2; and the specific surface is 6 cm^2/cc. Let us suppose that the original cube is subdivided into cubes whose sides are 1/10,000 as originally, i.e., 1μ (micron). These particles will be about the size of fine clay particles, and such a division will produce 10^{12} cubes with a total surface of 60,000 cm^2 Thus, it may be seen that the specific surface increases from 6 to 60,000 cm^2/cc as the size of the cubes decrease from 1 cm (gravel) to 1μ (fine clay). It should be noted that clay particles are plate shape and not cubical, and that the majority of the clay fraction is much smaller than 1μ; therefore, the actual increase in surface area would be much greater than indicated above. A better estimate of the surface area of two of the most common clays, kaolinite and illite, are 40 and 808 m^2/g respectively.

The large surface-area-to-volume ratio of clay particles results in the properties of this type soil being predominantly a function of the strength of the electrical forces on the surface of the clay particles. Conversely, the relatively small surface-area-to-volume ratio for gravel, sand, and silt particles results in a negligible effect of any electrical forces which may be on the surface of these particles and a predominance of gravity forces in determining the properties of these soils.

Particle shape influences to varying degrees the maximum density, the compressibility, the shearing strength, and other engineering properties of soils. In some instances it has greater influence than particle size. Accumulations of platelike particles, regardless of individual particle size, are usually more compressible than materials composed of particles of equal diameter but of bulky or angular shape. It has been possible to demonstrate that the inclusion of only moderate amounts of mica flakes in sand increases the compressibility of the latter to a value approximating that of clay. It may also be shown that coarse-grained soils with angular particles have greater strength and bearing capacity than those whose particles are rounded; while in clays, the inclusion of interlocking, rod shaped particles is a source of added strength.

7-1.3 SPECIFIC GRAVITY (Ref. 1)

Specific gravity is defined as the ratio between the unit weight of a substance and the unit weight of some reference substance, usually water at 4°C. The specific gravity for the majority of soils falls between 2.65 and 2.85. A high organic content will lead to lower values while the presence of some of the less common heavy minerals may lead to higher values. Most soils are composed of numerous minerals each having a different specific gravity. Unless otherwise mentioned, the specific gravity refers to the average value for all grains.

The specific gravity is relatively unimportant as far as its effect on the behavior of soils is concerned. It is, however, necessary for determination of important properties such as void ratio and degree of saturation.

7-2 DESCRIPTION OF COMPOSITE MAKEUP

7-2.1 GENERAL IDENTIFICATION (Ref. 1)

It is important that an engineer be capable of properly identifying soils in the field to aid in estimating the engineering characteristics of the soil types encountered. Even where laboratory tests are needed, identification is necessary to reduce to a minimum the duplication of samples.

There are several simple tests that may be employed.

Sands and gravels can be easily identified by visual inspection. Important physical characteristics such as texture, gradation, and particle shape may also be described qualitatively on the basis of a visual examination.

Two simple tests can be utilized to detect the presence of silt and clay size particles since they cannot be distinguished with the unaided eye. If a handful of dry soil is sifted through the fingers and allowed to fall on the ground, the quality of silt and clay size particles present can be estimated from the amount of dust created. Secondly, with the soil either wet or dry, a small quantity is placed in a test tube or bottle, water is added, and then the bottle is shaken vigorously. Particles finer than about 0.05 mm in size will have settled approximately 3 inches in 30 seconds. Consequently, if after about ½ minute the top 3 inches of water are not substantially clear, the presence of a significant quality of silt and clay size particles is indicated.

Inorganic silts, in the wet state, can be rolled into thin threads; but if the threads are more than a few inches long they will not support their own weight when held by one end. In addition, at a moisture content corresponding to the pastic limit —the moisture content representing the moisture content of the soil at the lower limit of its plastic range, i.e., the moisture content representing the boundary between the semi-solid and plastic states of the soil—these threads will be soft and friable.

When mixed with water to the consistency of a thick paste, the surface of an inorganic silt will appear wet and shiny if a pat is shaken in the palm of the hand. If the pat is squeezed by cupping the hand the surface will dry up, appear dull, and eventually crack. A clay, similarly treated, will not change its appearance in any way.

Lumps of air-dried silt can easily be crushed between the fingers. If the dry powder of the coarser silts is rubbed between the tips of the fingers, a gritty feeling results.

Clays exhibit plastic properties over a moderate to wide range in water content. They can be molded and rolled into thin threads without breaking or crumbling and threads of considerable length will support their own weight when held by one end when their moisture content is in the plastic range.

At moisture contents corresponding to the plastic limit, the more plastic the soil the harder and tougher it will feel. Consequently, the consistency of a soil at a moisture content corresponding to the plastic limit distinguishes between silts and clays, and also affords a simple method of evaluating the engineering characteristics of the various clays by measuring qualitatively their degree of plasticity.

Lumps of air-dried clay are relatively hard and the more plastic the clay, the harder they are. These lumps can often be broken but cannot generally be crushed and powdered by hand.

The color of the soil is helpful in distinguishing between soil strata and in identifying soil constituents but it does not signify that the soil possesses certain specific properties. Color of soil often varies with the degree of moisture present; hence, it necessary to consider the condition of the soil. There is, generally, more contrast in color when the soil is in a moist condition and the colors become less distinct as the moisture content is reduced. In fine-grained soils, certain dark or drab shades of gray or brown—including colors that are almost black—are indicative of organic colloidal matter. In contrast, clean and bright looking colors—including medium and light gray, olive green, brown, red, yellow and white—are generally associated with inorganic soils. Soil color may also indicate the presence of certain chemicals. Red, yellow, and yellowish-brown soil colors may be a result of the presence of iron oxides. White to pinkish colors may indicate presence of considerable silica, calcium carbonate, or aluminum compounds. Grayish-blue, and gray and yellow mottled colors frequently indicate poor drainage.

Organic soils can be identified by their gray to black color, by the presence of undecomposed vegetable matter or shells, and by odor—in many instances they contain gas which is released when the soil is excavated. The odor generally reduces with exposure to air but becomes pronounced if a wet sample is heated. If a sample is rolled into a thin ball and exposed to the atmosphere for a few days, it will usually shown annular rings of discoloration when broken open. The discoloration is the result of the oxidation of thin layers of organic material. Organic soils are highly compressible and are undesirable construction materials.

The general identification of a soil should include prominent characteristics of the undisturbed materials. Sand and gravel are described as being loose, medium, or dense while clays are described as hard, stiff, medium, or soft. The proper designation is arrived at on the basis of simple tests and observations, previously described, such as the difficulty or ease of pushing a probe into the soil, the ease or difficulty of excavating the soil and/or consistency tests.

7-2.2 MOISTURE CONTENT (Ref. 1)

7-2.2.1 General Discussion

An understanding of the way in which soils are affected by water is probably more important than any other single factor. The moisture contained in a soil is important to considerations of such soil phenomena as shear resistance, frost action*, volume change with a change in moisture content, and also for an understanding of the properties of compacted soil. Consider the two following phenomena as examples of this effect. (a) It is not uncommon for clays to undergo very large volume changes with variations in moisture content—evidences of this are the shrinkage cracks developed in a lake bed as it dries. (b) Unpaved clay roads, hard when sun-baked, lose stability and turn into impassible mud in a rainstorm. Either of these results are of great importance to an engineer and both are functions of the change in moisture content of the soil.

Coarse-grained soils are much less affected by moisture than are fine-grained soils. Coarser soils have larger void openings and drain more rapidly. Capillarity is no problem in gravels and sand having only very small amounts of fines mixed with them. These soils will not usually retain large amounts of water if they are above the ground water table and are generally immune from the detrimental effects of frost action.

Moisture content (MC) is the term used to define the amount of water present in a given soil. It is expressed as a percentage and is equal to the ratio of the weight of water to that of soil solids.

7-2.2.2 Nuclear Moisture Meter (Refs. 2, 3)

In recent years, nuclear testing apparatus has been developed to measure directly the moisture present in the soil. The apparatus is advantageous for field use because it enables a rapid, on-the-spot, nondestructive measurement of moisture to be made. Reference 2 and 3 describe the operation of the meter.

The main disadvantage of nuclear moisture meters is the effect of soil surface conditions on the number of detections. Consistent and reliable readings may be obtained only if the soil surface under the instrument is very smooth and flat.

7-2.3 GRAIN-SIZE DISTRIBUTION

7-2.3.1 General Discussion

Few natural soils consist of grains all the same size. The property of a soil which describes the distribution of size groups represented is called gradation. Knowledge of the grain-size distribution (gradation) is required in order to classify a soil.

A well-graded soil is composed of particles that vary in amount according to size in such a manner that the void spaces formed by the larger particles can be filled by smaller and smaller particles so that a minimum volume of voids is achieved. A soil composed of particles ranging in size between wide limits, but lacking the proper proportions of the respective sizes to quality as well graded, is said to be poorly graded; if one or more size ranges are completely absent, the soil is said to be gap-graded. A uniformly graded soil is one composed essentially of particles of equal size. These four categories of gradation, i.e., well-graded, poorly graded, gap-graded and uniformly graded, include all possible gradations of soil.

7-2.3.2 Sieve Analysis (Ref. 1)

The process by which the grain size distribution of the coarse fraction of soils (gravel and sand) is determined is the sieve analysis procedure. A sample, air-dried and with the aggregates thorough-

* Frost action is a general term used in reference to freezing and thawing of moisture in materials and the consequent effects on these materials and the structure of which they are a part, or with which they are in contact.

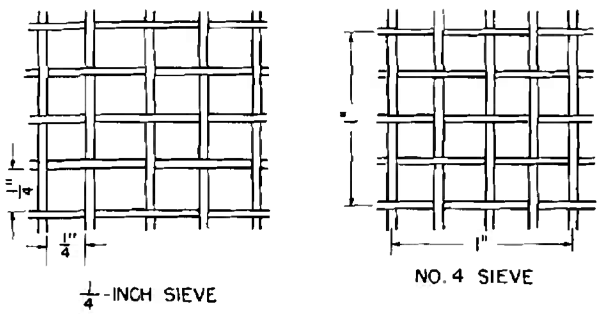


Figure 7-1. Sieve Nomenclature

ly broken up, is passed through a set of sieves having openings ranging from 3 inches to 0.074 mm, and the weight of material retained on each sieve is determined. The number and sizes of sieves used depend on the range of particle sizes in the soil and on the precision desired in the determination of the gradation curve.

Sieve sizes are denoted by two nomenclatures. One refers to the sieve screen size in inches, the other by number. The inches designation refers to the actual linear dimensions of the openings between the wires of a sieve screen. The number designation of a sieve indicates the number of openings existing in the screen per linear inch. Figure 7-1 shows the difference more clearly.

The difference in the size of the openings of these two sieves is a result of the thickness of the wires being included along with the four openings per linear inch of the No. 4 sieve. By measurement, the opening in the No. 4 sieve is 4.75 mm while that of the ¼-inch sieve is 6.35 mm. The sieves with openings larger than ¼-inch are referenced according to the size of the opening. Sieves ¼-inch or smaller are identified by the number of openings per linear inch.

The percentage of material by weight retained on the various sieves is computed as follows:

$$\text{Percent retained} = \frac{\text{weight in grams retained on a sieve}}{\text{total weight in grams of oven-dry sample}} \times 100$$

The results of the sieve analysis are presented in the form of a grain-size distribution curve on a semi-logarithmic chart. The grain-size distribution curve is obtained by plotting particle diameters (sieve openings) on the abscissa (logarithmic scale) and the percent finer by weight on the ordinate (arithmetic scale).

7-2.3.3 Hydrometer Analysis (Ref. 5)

A hydrometer analysis is required to determine the grain size distribution of particle sizes smaller than 0.074 mm. The use of an immersion hydrometer to measure the specific gravity of a liquid is well known. This principle is utilized to measure the varying specific gravity of a soil suspension as the grains settle. This information is used to determine the grain-size distribution curve.

The first step in the presentation of this method is to obtain an expression for the unit weight γ of a suspension at any time and at any depth. A suspension of volume V containing a total weight of a suspended soil W_s is initially mixed thoroughly. At the instant sedimentation starts, every element of volume contains the same concentration of suspended soil; for example, a 1-liter suspension taining 50 grams of soil will have 50 milligrams of soil in every cubic centimeter. The initial unit weight γ_i of the suspension is easily expressed. In each unit of volume of the suspension, the weight of solids is $\frac{W_s}{V}$, and the corresponding volume of solids is $\frac{W_s}{G\gamma_w V}$, where G is the specific gravity of the soil particles and γ_w is the unit weight of water. The volume of water in the unit volume of suspension is then $1 - \frac{W_s}{G\gamma_w V}$. The initial unit weight can then be computed as

$$\gamma_i = \gamma_w + \frac{G-1}{G} \cdot \frac{W_s}{V} \qquad (7\text{-}1)$$

At any depth z below the surface of the suspension, the diameter D of a particle which would fall from the surface to the depth z in time t is computed from Stokes' Law.

$$v = \frac{z}{t} = \left(\frac{\gamma_s - \gamma_w}{18\mu}\right) D^2 \qquad (7\text{-}2)$$

where

v = terminal velocity of the particle, cm/sec

γ_s = unit weight of the solid particle, g/cu cm

γ_w = unit weight of the fluid (normally water), g/cu cm

μ = viscosity of the fluid, g-sec/sq cm

Rearrangement gives the particle diameter in cm

$$D = \sqrt{\left(\frac{18\mu}{\gamma_s - \gamma_w}\right)\left(\frac{z}{t}\right)} \qquad (7\text{-}3)$$

The diameter D, as expressed by Equation 7-3, is called a limiting diameter because no particles of a diameter greater than D will be present at the depth z, i.e., all larger diameter particles will have fallen a greater distance.

At the depth z the weight of particles of smaller diameter than D will be unchanged because as many particles will have settled to that depth as have settled from it. If N is the ratio of the weight of particles smaller than D to the weight of all particles in the original sample, the weight of solids per unit volume at depth z at time t is $\frac{NW_s}{V}$. The unit weight of the soil suspension (g/cm^3) at depth z is then

$$\gamma = \gamma_w + \frac{G-1}{G}\left(\frac{NW_s}{V}\right)$$

from which

$$N = \frac{G}{G-1}\left(\frac{V}{W_s}\right)(\gamma - \gamma_w). \qquad (7\text{-}4)$$

The ratio N represents the portion of soil finer than the diameter D of Equation 7-3. Hence, a determination of unit weight at any known depth z for any elapsed time t furnishes a point on the grain-size distribution curve.

The unit weight at depth z is determined by means of a hydrometer. The hydrometer is calibrated to read unity in pure water at the temperature of calibration. In other fluids

$$r = \frac{\gamma}{\gamma_c} \qquad (7\text{-}5)$$

where

r = specific gravity of fluid (at the temperature of calibration), dimensionless

γ = unit weight of fluid (at the time of the hydrometer reading), g/cm^3

γ_c = unit weight of water (at the temperature of calibration), g/cm^3

Equation 7-4 may then be written as

$$N = \frac{G}{G-1}\left(\frac{V}{W_s}\right)\gamma_c(r - r_w), \qquad (7\text{-}6)$$

where

r_w = hydrometer reading in water at the test temperature

The hydrometer analysis is satisfactory when the grain-size curve is desired only for sizes greater than 1 or 2 microns. In fine clays, a large percentage by weight may be finer than these sizes. These particles are colloidal and tend to remain in suspension indefinitely.

An apparatus has been developed for testing fine clays in which the hydrometer jar is suspended at the end of rotating arm. The centrifugal force caused by rotation introduces acceleration many times greater than gravity causing much faster settlement than occurs under gravity alone. Grain-size distribution curves which extend to 0.01 micron may be obtained in a few hours with this apparatus.

The hydrometer analysis is subject to several limitations. Stokes' Law defines the phenomenon of a single spherical particle (of constant specific gravity) falling through a fluid which is indefinite in extent. None of these conditions are satisfied. However, the only limitation which cannot be overcome is the requirement of a spherical particle —clay particles are generally plate shaped. Consequently, in connection with the hydrometer analysis, it is necessary to understand the concept of an equivalent diameter. Any particle which has the same velocity of fall as a sphere of the same unit weight and of diameter D will be said to have an equivalent diameter D_1. Therefore, the diameter scale for any mechanical analysis plot based on Stokes' Law shows equivalent diameters rather than any actual dimensions of particles.

7-2.4 DENSITY

Density is defined in physics as the mass per unit of volume and usually relates to a single homogeneous, or isotropic substance. In soils engineering, however, density refers to weight per unit volume and is generally expressed as either wet density or dry density. Wet density is computed as

$$\gamma_m = \frac{W_s + W_w}{V_t} \qquad (7\text{-}7)$$

where

γ_m = wet density, g/cm^3
W_s = weight of soil solids, g
W_w = weight of water, g
V_t = total volume occupied by the soil particles including voids within and between particles, cm^3

Dry density is computed as

$$\gamma_d = \frac{W_s}{V_t} = \frac{\gamma_m}{1 + MC} \qquad (7\text{-}8)$$

where MC is the moisture content of the soil, expressed as a decimal.

Several techniques are used to measure soil density. The most common are outlined in the paragraphs which follow.

7-2.4.1 Direct Measurement

The total weight of a soil sample W is first determined and then its volume V_t is ascertained through immersion in mercury. The sample is then placed in an oven to dry until a constant weight W_s is achieved. If W, W_s, and V_t are known the wet and dry densities can be computed by Equations 7-7 and 7-8.

7-2.4.2 Sand Cone Method (Ref. 4)

The sand cone method is intended for determining the in-place density of either fine- or coarse-grained materials. This test is so named because sand is used, following prescribed procedures, to determine the volume of the hole from which a sample has been taken. The test consists of digging out a sample of the material, determining the volume of the hole from which the sample was removed, and determining the wet and dry weight of the sample. The procedure described is restricted to soils containing particles not larger than 2 inches in diameter. The standard apparatus is shown in Figure 7-2.

Any clean, dry, free-flowing sand can be used, but a uniformly graded sand consisting of rounded particles gives the best results. The density of the sand is obtained by determining the weight of sand required to fill a container (approximately 5 inches in diameter and 5 inches deep) of known volume. The container is filled by allowing the sand to

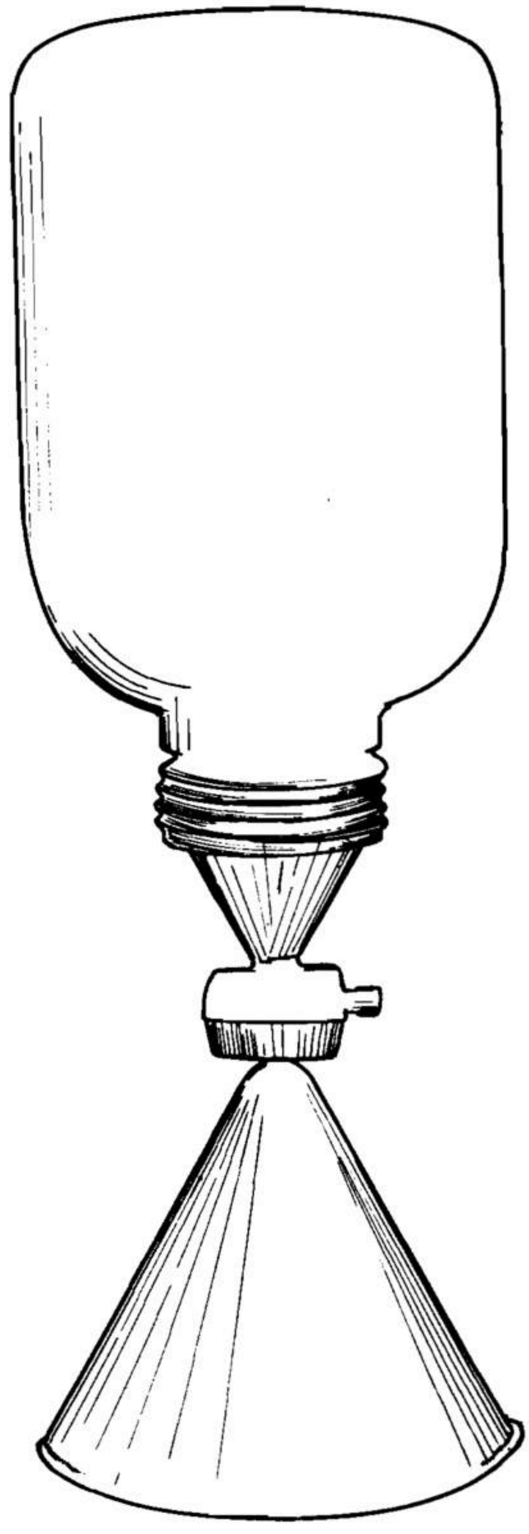


Figure 7-2. Sand Cone Apparatus

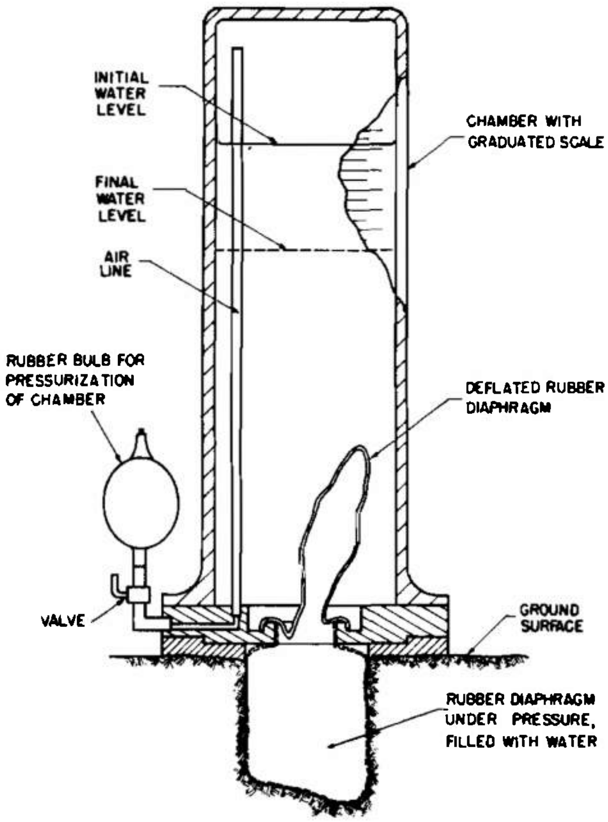


Figure 7-3. Rubber Balloon Apparatus

flow from the bottle, through the funnel, and into the container.

If the density of the sand and the weight of sand required to fill a hole excavated in soil whose density is to be determined is known, the volume of the hole can be computed. By dividing the volume of the hole into the weight of the soil excavated, the wet density of the soil can be computed. The exact procedure for calculating the density of a soil by the sand cone method is as follows:

(a) Seat the apparatus on a level surface and mark the outline of the funnel.
(b) Dig a test hole inside the funnel mark of a size approximately equal to the funnel. Weigh the material excavated and determine its moisture content.
(c) Seat the apparatus in the previously marked position, open the valve, and, after the sand has stopped flowing, close the valve.
(d) Weigh the apparatus with the remaining sand and determine the weight of sand used in the test. Compute the volume of sand used and subtract the volume of the funnel to determine the volume of the soil cavity.
(e) Compute wet density from Equation 7-7 and dry density from Equation 7-8.

7-2.4.3 Rubber Balloon Method (Ref. 4)

This test is restricted to test hole volumes between 0.025 and 0.050 cubic feet, and to bonded soil masses in which the maximum particle size is one-half inch. The volume of a test excavation is measured by determining the volume of liquid, contained in a thin flexible membrane, required to fill the cavity. The strength of the membrane is required to be such that not more than 5 psi is necessary to cause it to take the shape of the test hole. The standard apparatus is shown in Figure 7-3.

The test is performed using the following procedures:

(a) Seat the apparatus on a level surface and apply 5 psi to the surface of the liquid, causing the balloon to conform to the ground surface. Record the volume of liquid remaining in the apparatus.
(b) Dig a test hole approximately 3 inches in diameter, weigh the material excavated, and determine its moisture content.
(c) Seat the apparatus over the hole and apply 5 psi to the liquid surface. Record the volume of liquid remaining in the apparatus. The difference in the reading obtained here and from step (a) is the volume of soil excavated.
(d) Compute wet and dry densities from Equations 7-7 and 7-8.

7-2.4.4 Drive Cylinder Method (Ref. 4)

The drive cylinder method of determining in-place density can be used satisfactorily in moist, cohesive, fine-grained materials and in sands which exhibit tendencies toward cohesiveness. An undisturbed sample is obtained by driving a small, thin-walled cylinder into the material with a special driving head. The cylinder is driven in until its top is approximately one-half inch below the original surface. The driving head is then removed and the cylinder is dug from the ground. If the tare weight of the cylinder is known, the wet weight of the sample, with any excess soil trimmed off, can be obtained immediately. The sample is then removed from the cylinder and the moisture content determined. The volume of the sample is equal to the volume of the cylinder. Wet and dry densities may then be computed by Equations 7-7 and 7-8.

7-2.4.5 Nuclear Density Probe (Refs, 3, 4)

As with the nuclear moisture meter, the nuclear density probe is also of rather recent development. The primary components of the density probe are a gamma ray source, a gamma ray detector, and a scaler to count the number of detections.

Two types of density probes are available. The most common is the backscatter type in which both the source and detector are mounted in a device which rests on the ground surface. Both the source and the detector are shielded from each other in the instrument. The second type is the direct transmission density probe in which either the source or the detector, but not both, is inserted into the soil to various depths—up to 12 inches.

The nuclear method of determining density is based on the absorption of gamma rays. A radioactive source is used which emits gamma rays into the soil (or into some other material, the density of which is to be measured). As the gamma rays pass through the soil, they collide with electrons, lose part of their energy, and continue their travel along deflected paths. Through a series of collisions the gamma-rays are scattered in all directions and their energy may eventually be reduced to a level where the gamma rays are no longer detectable.

The absorptive capacity of the soil depends on electron density (number of electrons per unit volume), which is a function of the density of the soil. Obviously, the greater the absorptive capacity of the soil, the smaller the number of gamma rays reaching the detector. In other words, an inverse relationship exists between soil density and the gamma ray intensity indicated by the detector. Because of the complexity of the scattering phenomenon, the functional relationship between absorptive capacity and soil density must be determined experimentally (calibration).

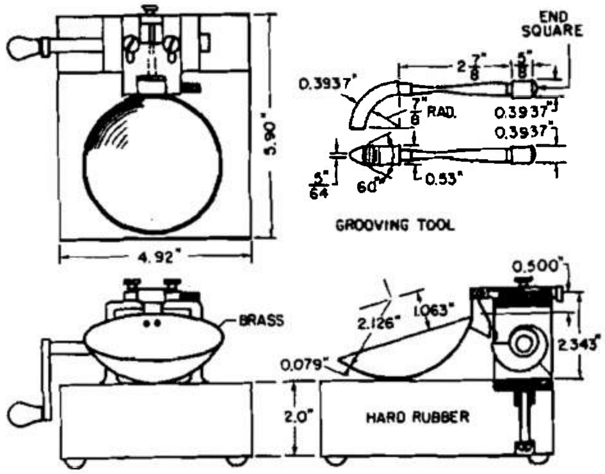


Figure 7-4. Mechanical Liquid Limit Device

As with the nuclear moisture meter, the measurements made with density probes are very sensitive to surface conditions under the instrument. Air gaps under the instrument of one eighth inch can cause errors of 10 percent or more.

7-2.5 ATTERBERG LIMITS (Refs. 3, 4)

7-2.5.1 Liquid Limit

The liquid limit LL is the water content corresponding to the arbitrarily determined boundary between the liquid and plastic states of consistency of a soil. It is considered to be the moisture content at which two halves of a soil pat separated by a groove of standard dimensions will close at the bottom of the groove for a distance of one-half inch under the impact of 25 blows in a standard liquid limit apparatus (Figure 7-4). Trials are performed with samples at several different moisture contents and a plot is made of moisture content (arithmetic scale) versus the number of blows (log scale) required to close the groove over the prescribed length. The liquid limit can then be read from the linear plot obtained and a trial at exactly 25 blows is not required.

7-2.5.2 Plastic Limit

The plastic limit PL is the water content corresponding to an arbitrarily defined limit between the plastic and the semisolid states of consistency of soil. It is obtained by rolling soil samples at slowly decreasing water content until a water content is reached at which a soil thread one-eighth inch in diameter just begins to crumble. The soil threads are rolled on a glass plate with the hand; consistent results can be obtained without special apparatus.

7-2.5.3 Plasticity Index

The plasticity index PI is the range of water content between the liquid and plastic limits, i.e., the range in moisture content over which the soil will behave as a plastic. This index, coupled with the liquid limit, is an excellent indicator of the plastic behavior of soils. Since plasticity is a unique property of clays, the plasticity index is an indicator of the relative clay content of similar soil types. Furthermore, soils with higher plasticity indexes will exhibit greater volume charges with varying moisture contents and will suffer greater loss of stability when wet than the more "lean," low plasticity, soils.

7-2.5.4 Shrinkage Limit

The shrinkage limit w_s is the water content of a clay soil at which the soil ceases to change in volume upon further loss of water due to drying. Two methods are used for determining the shrinkage limit. When the specific gravity is not known, the shrinkage limit can be calculated from the following equation:

$$w_s = \frac{(W_1 - W_s) - \gamma_w(V_1 - V_2)}{W_s} \quad (7\text{-}9)$$

where

V_1 = initial volume of soil specimen, in.^{3}

W_1 = initial weight of the soil specimen, lb

V_2 = volume of the oven dried specimen, cu in.

W_s = weight of the oven dried specimen, lb

γ_w = unit weight of water, lb/in.^{3}

If the specific gravity of the soil is known, the shrinkage limit may be determined from the values of W_s and V_2 as

$$w_s = \frac{\gamma_w V_2}{W_s} - \frac{1}{G} \quad (7\text{-}10)$$

where G = the specific gravity of the soil solids, dimensionless.

7-3 DETERMINATION OF PHYSICAL CHARACTERISTICS

7-3.1 SOIL STRESS-STRAIN AND STRENGTH TESTS

7-3.1.1 Unconfined Compression (Ref. 4)

This test is intended to determine the unconfined compressive strength of cohesive soils. A cylindrical soil specimen of a length at least twice its diameter is loaded axially to failure or to 20 percent strain depending on which occurs first. Two methods of load application may be used: controlled stress or controlled strain. In the first method, load is applied in increments at equal intervals of time and the axial shortening of the specimen is recorded. In the controlled strain method, the specimen is deformed at a constant rate and the load is recorded at intervals as required to obtain the desired degree of precision in the load-deformation curve. In saturated cohesive soils the shear strength is assumed to be half the unconfined compressive strength and is independent of confining pressure.

7-3.1.2 Triaxial Compression (Refs. 8, 9)

The triaxial test is presently the most widely used laboratory method for the measurement of soil strength and stress-strain relationships. Its principal advantages are that it permits precise control of confining pressure, drainage, and rate of loading, and provides for the measurement of pore pressures and volume changes. Its versatility in these respects has caused it to be used extensively not only for investigating strength and stress-strain characteristics of soil under conditions representative of those existing in natural soils and embankments, but also as a means of exploring the basic parameters affecting the shearing resistance of soil.

The standard triaxial test specimen is a soil cylinder, normally of a length at least twice its diameter. It is placed between two plates—either pervious or impervious depending upon test conditions—sealed in a watertight membrane and enclosed in a pressure cell. The specimen is subjected to an all-around confining pressure and loaded axially by means of a piston acting on an end plate.

The most common triaxial tests are the following:

(a) *Drained Tests.* Drainage is permitted throughout the test with full pore pressure dissipation, permitting direct measurement of effective stresses.

(b) *Consolidated-Undrained Tests.* Drainage is permitted until the sample is fully consolidated under the confining pressure. No drainage is permitted during the application of the axial load. Pore water pressures may or may not be measured during the test.

(c) *Undrained Tests.* No drainage, and hence, no dissipation of pore pressure is permitted under either the confining pressure or the axial load.

The methods of loading normally used are:

(a) *Controlled Stress.* Load is applied in increments.

(b) *Controlled Strain.* Load is measured at intervals of deformation as the specimen is being deformed axially at a constant rate.

The test is performed with a number of different lateral pressures. Test data are analyzed graphically by use of Mohr circles to determine the cohesion and angle of internal friction of the soil. Also, consideration is given to the shape of the stress strain curve for each test.

For a very good discussion of triaxial testing and its significance, see Ref. 5.

7-3.1.3 Direct Shear (Ref. 4)

A soil specimen is placed in a split mold and shearing forces are applied to cause one portion of the specimen to slide in relation to the other portion. The test is conducted on several specimens each with a different load acting normal to the shearing force. The unit normal forces applied and the shear stresses at failure are plotted to determine the angle of internal friction and cohesion of the soil. The direct shear test is used for both cohesive and cohesionless soils.

7-3.1.4 Torsional Shear (Ref. 4)

In the torsional shear test a specimen of solid circular or annular cross section, usually confined between rings, is subjected to an axial load and to shear in torsion. The annular cross section is the more difficult with respect to test preparation but is the more preferred because the shearing stresses are more uniform on the cross section and, hence, may be computed more readily.

In-place torsion shear tests are performed by pressing a dentated solid circular or annular plate against the soil and measuring the resistance to its rotation under a given axial load.

7-3.1.5 Vane Shear (Ref. 6)

The device used for the determination of vane shear consists of a four-bladed vane fastened to the bottom of a vertical rod. The vane is pushed into soil and the torque required to rotate the vane is measured. The soil fails on an assumed cylindrical surface passing along the outer edges of the vane. Average shearing resistance can be computed from the dimensions of the vane and the measured torque.

The vane may then be rotated rapidly through several revolutions to remold the soil. The shearing strength can again be measured to determine the sensitivity of the soil to remolding.

This device is best suited for use in cohesive soils and is not recommended for use in sandy soils, gravel, or soil deposits containing many large pebbles.

7-3.1.6 Soil Penetrometer (Refs. 4, 6)

A penetrometer is a probe which is forced into the ground to obtain a measure of soil strength. If the penetrometer is pushed into the ground slowly, the test is classified as being a static penetration test. If the probe is driven into the ground, the test is classified as a dynamic penetration test. Static tests are generally performed in soft, cohesive soils and dynamic tests in cohesionless or hard soils.

The most common test performed is the standard penetration test. This is made by driving a split-barrel sampler (Figure 7-5) into the soil by dropping a 140 lb hammer onto it from a height of 30 inches. The penetration resistance is the number of blows required per 12 inches of penetration. Penetration resistance can be related to soil density in accordance with Table 7-1. These

TABLE 7-1
RELATIONSHIPS OF PENETRATION
RESISTANCE AND RELATIVE DENSITY

Sand, No. of Blows	Relative Density	Clay, No. of Blows	Consistency
0–4	Very Loose	1–2	Very Soft
4–10	Loose	2–4	Soft
10–30	Medium	4–8	Medium
30–50	Dense	5–15	Stiff
>50	Very Dense	15–30	Very Stiff
		>30	Hard

data may be considered fairly reliable with sands but only an approximation for clays.

In a sandy soil located below the water table, the blow count may indicate a relative density considerably greater than the actual relative density of the deposit. Under these conditions, for blow counts greater than 15, the values should be corrected to 15 + ½ (blow count − 15) before Table 7-1 is used. This is necessary because the material tends to increase in volume upon shearing, producing an increase in effective stress which results in an apparent increase in penetration resistance. In deposits containing boulders, the results may be unreliable because of the small size of the sampling spoon compared to that of the boulders.

The standard penetration test cannot be regarded as a highly refined and completely reliable method of investigation; however, the blow count gives a useful preliminary indication of the consistency or relative density of most soil deposits.

7-3.1.7 California Bearing Ratio (Ref. 8)

A piston of 3 in.^{2} cross sectional area is forced into a compacted sample of 6 inches diameter in a mold. Rate of piston movement is controlled, and pressure readings are taken for various penetration depths. The standard of comparison for computing a material's bearing value is based upon the penetration and load, or pressure, on a standard, well-graded crushed stone (1000 psi to produce 0.1 inch penetration). The bearing value of a sample is determined for a specific penetration by dividing the load for the penetration by the standard load for the same penetration. For example, if

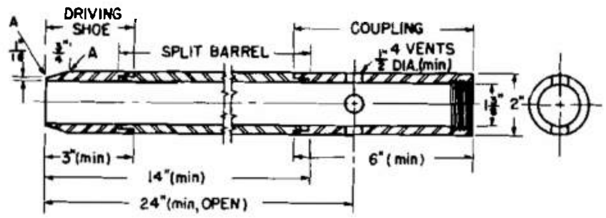


Figure 7-5. Standard Split-Barrel Sample Assembly (from ASTM Standards)

a specimen requires a load (or pressure) of 450 psi to obtain 0.1-inch penetration, its bearing value will be (450/1,000) × 100 = 45 percent. This bearing value has become known as the California Bearing Ratio, generally abbreviated to CBR, with the "percent" omitted. The Corps of Engineers and some highway departments use the CBR principle in conducting tests to evaluate the bearing value of materials.

7-3.1.8 Cone Penetrometer (Ref. 7)

The cone penetrometer was developed to obtain an empirical field measure of soil bearing-traction capacity. A right circular 30-degree cone is pushed slowly into the soil and the force required is recorded at desired increments. This device and procedures for its use were developed by the Corps of Engineers to measure the trafficability of soils.

The measure of a soil's trafficability is referred to as a *rating cone index*. It is obtained as follows: the cone index of an "undisturbed" soil sample, confined in a small cylinder, is measured before and after tamping with 100 blows of a 2½-pound tamper falling 12 inches. The ratio of the index before and after tamping is called the *remolding index*. The rating cone index is then obtained as the product of a soil's *in situ* cone index and its remolding index.

The cone penetrometer's value lies in the fact that it is simple, and that a vast number of tests have been performed with it and their results correlated with other soil tests and other soil classification systems. It is one of the most widely used field tests for determining an index of the soil strength.

7-3.1.9 Bevameter (Refs. 8, 9)

The Bevameter was developed by the Land Locomotion Research Laboratory of the U. S. Army Tank-Automotive Center for predicting the performance of wheels on soil. It is a device for determining the load-versus-sinkage curves for two different diameter circular plates when pushed into the soil, and also for determining the torsional resistance of a cleated circular plate under a controlled normal load. The data obtained from these tests are used to determine the soil values k_c, k_ϕ, and n (paragraph 4-4.2) required in formulas derived for computing traction, rolling resistance, and sinkage of wheels.

7-3.2 CONSOLIDATION TESTS (Ref. 14)

The consolidation test is a one-dimensional compression test which involves, simultaneously, the slow escape of water and a gradual compression of the soil structure. The apparatus used for the study is called a consolidometer. The soil sample is placed in a standard ring with porous disks at the top and/or bottom to provide drainage. The specimen is loaded in steps, each load doubling the previous value of load on the specimen. New loads are not added until compression of the specimen under the previous loading has practically ceased. This may take only a few minutes with sands, but in clays, compression continues for a long period of time and one-day intervals between loadings are quite common. A curve of compression versus time is plotted for each increment of load. The final data consist of time-compression curves for each load increment and a load-compression curve for the test as a whole. The load-compression data, or final stress-strain relationships, are presented in the form of a curve of log of pressure versus void ratio for the final condition of each load increment.

7-3.3 PERMEABILITY TESTS (Refs. 4, 6, 10)

A material is said to be permeable if it contains continuous voids. Every soil and every rock satisfy this condition. There are, however, large differences in the degree of permeability of the various earth materials. The quantity of flow through a dense rock may be so small as to pass unnoticed because evaporation prevents the accumulation of water on the exposed face; yet the flow of water through such a material may produce seepage pressures between the mineral grains larger than those exerted in more permeable materials under the same conditions of hydraulic head.

The flow of water through permeable substances is governed by an empirical relationship first stated by Darcy

$$v = k\,i \tag{7-11}$$

where

v = discharge velocity (the quantity of water that percolates through a unit area perpendicular to the direction of flow, in unit time), cm/sec

i = hydraulic gradient in the direction of flow cm of head per cm of flow distance), dimensionless

k = coefficient of permeability, cm/sec

In general, k is a function of the permeable substance and the unit weight and viscosity of the fluid. Because fluid viscosity changes with temperature, the coefficient of permeability, in the general case, also varies with temperature. However, in soil mechanics the only fluid of interest is water and its changes in viscosity over the range of temperature of groundwater are negligible. Hence, the coefficient of permeability is considered a property of the soil or rock.

The coefficient of permeability k generally increases with increasing size of voids which in turn increases with increasing grain size. However, the shapes of the void spaces also have a marked influence on the permeability. As a consequence, no simple relationships have been found between permeability and grain size except for fairly coarse soils with rounded grains. For example, Allen Hazen (Ref. 5) found that, for filter sands, in the cgs system of units

$$k = 100\ D^2_{10}\ \text{cm/sec} \qquad (7\text{-}12)$$

where D_{10} is the effective diameter (particle diameter corresponding to 10 percent fines on the grain size curve of the soil considered), cm. The conversion factor 100 changes the units to cm/sec.

Similarly, it has been found that the coefficient of permeability of coarse-grained soils varies approximately as the square of the void ratio. No such simple relation has been established for soils containing flake-shaped particles.

Permeability tests are performed with both constant-head and falling-head permeameters. In either case, an average coefficient of permeability is determined by measuring the quantity of water which flows through a test specimen in a given time period. The constant head permeability test is recommended for highly permeable materials such as sand and gravel. The value of the coefficient of permeability is in ft/sec, computed from experimental results as

$$k = \frac{QL}{hAt} \qquad (7\text{-}13)$$

where

Q = quantity of flow in a given time, ft^3
L = sample length, ft
h = head loss, ft
A = sample cross sectional area, ft^2
t = time, sec

The falling-head permeability test is recommended for materials of low permeability. The value of k (units as in Eq. 7-13) is computed from experimental results as

$$k = 2.3 \left(\frac{aL}{At}\right) \log \left(\frac{h_o}{h_1}\right) \qquad (7\text{-}14)$$

where

a = cross sectional area of the stand pipe which provides the hydraulic head, ft^2
h_o = original hydraulic head, ft
h_1 = final hydraulic head, ft

The other symbols are defined as in Equation 7-13.

7-3.4 COMPACTION AND MOISTURE-DENSITY RELATIONSHIPS

7-3.4.1 General Discussion

The properties which a soil displays may vary widely depending upon the degree of moisture present in the soil and upon the density of the soil. The density to which a soil may be compacted varies to a great extent with the amount of moisture present and the compactive effort applied, particularly in cohesive soils. For a given controlled method of compaction, the moisture content which permits the greatest density to be obtained is referred to as the optimum moisture content. The optimum moisture content for a given soil is not a constant value but will vary for different methods of compaction and for different compactive efforts. Because of these effects, it is necessary that standard tests be used to define the degree of compaction of a soil. The most common of these tests are outlined in the paragraphs which follow.

7-3.4.2 Proctor Compaction (Ref. 4)

Compaction is a process of bringing soil to a dense state by mechanical means. The Proctor Compaction Test was developed as a standard test to define degree of compaction. The apparatus consists of a metal cylinder, 4 inches inside diameter by 4.6 inches high, and a metal tamper, weighing 5.5 lb, with a 2-inch diameter face. The soil is placed in the cylinder in three approximately equal layers, and each successive layer is given 25 compaction blows of the tamper falling freely through a distance of 12 inches. The weight, moisture content, and dry density of the compacted specimen are determined. The test is repeated a number of times with the soil at various moisture contents, and the results are plotted to obtain a moisture-dry density curve. The peak of this curve indicates the maximum dry density, and the moisture content at which this density is obtained is defined as the *optimum moisture content* for that particular soil.

The details of the Proctor Test were selected to yield maximum dry densities comparable to the highest densities obtainable with field compaction equipment. Since the test was devised in 1934, more efficient compaction equipment has become available and the Proctor Test has been generally replaced by the Modified AASHO* Compaction Test. This test is identical to the Proctor Test except that the sample is compacted in five layers (25 blows per layer) with a 10 lb hammer falling 18 inches.

7-3.4.3 Harvard Miniature Compaction Test (Refs. 4, 10)

This test is, as is the Proctor Test, intended to study moisture-density relationships of soil. The apparatus used consists of a cylinder, 1-5/16 inches inside diameter by 2.8 inches high, and a tamper, 0.5 inch in diameter, reacting against a spring of desired stiffness. The tamping device applies a kneading action to the soil in compacting it. The selection of the number of layers in which the soil should be placed, the number of tamps per layer, and the tamping force depend upon the soil type used and the use intended for the compacted material.

7-3.4.4 Compaction of Cohesionless Soils (Ref. 6)

In the laboratory, the greatest densities are obtained with dry sand when it is emplaced using free falling techniques. Since these techniques are impractical in the field, vibrations are the most effective means for field compaction of large deposits of loose, cohesionless soil. By the same token, vibrations constitute one of the most serious causes of settlement of structures erected on these materials.

One of the best known field methods of compaction is the vibroflotation process. According to this method, a heavy, steel capsule containing an internal vibrator is lowered into the soil deposit. At the same time, powerful jets of water are forced into the soil beneath the capsule. Under the combined action of jetting and vibration, the device sinks rapidly and creates a crater at the ground surface. As the crater develops, it is filled with loose soil. Compaction to the depth of penetration for a diameter of 6 to 8 ft is generally quite satisfactory. This procedure is most effective in clean, medium to coarse sands but is not effective in very silty sands or silts.

SECTION II SOIL CLASSIFICATION

7-4 INTRODUCTION

In order to accomplish his goals, the engineer must learn to identify the soil and, where possible, to infer design data from the identification procedure. As an aid to accomplishing this goal, soil classification systems have been developed to identify soils with similar properties and to group, or classify, soils that will perform in a similar manner when their densities, moisture contents, environmental conditions, etc., are similar.

There is no universal classification system. Due to the great magnitude of the possible uses of soil and its many properties which would need consideration, a universal classification system is impossible. Consequently, there are a multitude of classification systems available—each designed with a specific use in mind.

In the paragraphs which follow, the most important classification systems—from the standpoint of the engineer—will be considered. Attention should be called to the fact that engineering soil classification is based on grain-size distribution and Atterberg limits.

7-5 U. S. DEPARTMENT OF AGRICULTURE SYSTEM (USDA SYSTEM)

This system was developed by agriculturists, soil scientists, and pedologists; however, it is used by others, e.g., engineers and geographers. It groups soils into units or classes significant for agricultural purposes, but certain soil units of the pedological system exhibit similar engineering behavior patterns.

The USDA classification system deals primarily with that portion of the soil profile which supports

* American Association of State Highway Officials.

TABLE 7-2
HIGHWAY RESEARCH BOARD CLASSIFICATIONS*
(Classification of Highway Subgrade Materials)

General Classification	Granular materials (35% or less passing No. 200)			Silt-clay materials (More than 35% passing No. 200)			
Group classification	A-1	A-2	A-3	A-4	A-5	A-6	A-7
Sieve analysis, percent passing							
No. 10							
No. 40	50 max	51 min					
No. 200	26 max	10 max	35 max	36 min	36 min	36 min	36 min
Characteristics of fraction passing No. 40:							
Liquid limit				40 max	41 min	40 max	41 min
Plasticity index	6 max	NP		10 max	10 max	11 min	11 min
Group index			4 max	8 max	12 max	16 max	20 max
General rating as subgrade	Excellent to good			Fair to poor			

(Subgroups)

General classification	Granular materials (35% or less passing No. 200)							Silt-clay materials (more than 35% passing No. 200)			
Group classification	A-1		A-3	A-2				A-4	A-5	A-6	A-7
	A-1-a	A-1-b		A-2-4	A-2-5	A-2-6	A-2-7				A-7-5 A-7-6
Sieve analysis, percent passing											
No. 10	50 max										
No. 40	30 max	50 max	51 min								
No. 200	15 max	25 max	10 max	35 max	35 max	35 max	35 max	36 min	36 min	36 min	36 min
Characteristics of fraction passing No. 40:											
Liquid limit				40 max	41 min	40 max	41 min	40 max	41 min	40 max	41 min
Plasticity index	6 max		NP	10 max	10 max	11 min	11 min	10 max	10 max	11 min	11 min
Group index	0		0	0		4 max		8 max	12 max	16 max	20 max
Usual types of significant constituent materials	Stone fragments, gravel, and sand		Fine sand	Silty or clayey gravel and sand				Silty soils		Clayey soils	
General rating as subgrade	Excellent to good							Fair to poor			

*From Highway Research Board

plant life. It considers the effects of five basic factors in classifying soil profiles. These are: (a) age, (b) parent material, (c) climate, (d) slope or topographic position, and (e) vegetative cover. The engineer would not select all of these factors as primary ones in classifying soils.

The various divisions used in this classification scheme are based on similarities of the characteristics or properties of the soil horizons, which are the horizontal layers which make up the soil horizons. Well-developed soil profiles contain three or four major horizons. Soils from a given parent material, having similar horizons except for the surface horizon, are grouped together under a *series* name. The series name plus the textural designation of the surface horizon defines the soil *type*. Pedologists generally map the soils of an area on the basis of soil type. Differences in adjacent soil series may be ascribed to the simple influence of slope or topographic position.

An understanding of the USDA classification system is advantageous to the engineer from the standpoint of utilizing pedologic maps and other soils information. One of the principal limitations to the use of pedologic data has been the quantity of detail which must be sifted through to obtain the desired information. Fortunately, many states have begun supplementing pedologic information with engineering data as part of a program supported by USDA. This will make the information much more accessible to the engineer.

It should be recognized that the pedologic information only extends to the depth of influence of soil forming processes. This depth is relatively shallow and may be exceeded by critical circles in slope analysis, by significantly stressed foundation soils under major embankments, and by foundations for bridge structures. However, it is probably quite adequate for loads imposed on pavements.

Classifying soils according to the USDA system is accomplished by visual observations, and field and laboratory tests conducted as outlined in Ref. 11.

7-6 HRB OR AASHO SOIL CLASSIFICATION SYSTEM

The Public Roads Administration Soil Classification System was revised by a Highway Research Board (HRB) Committee in 1945; it has become known variously as the Highway Research Board, American Association of State Highway Officials, or Modified Bureau of Public Roads System. This system is the most widely known and used in highway practice. It is an engineering properties classification based on field performance of highways.

The soil classifications are divided into two major groups: the granular materials containing 35 percent or less of material passing the 200-mesh sieve, and clay and silt-clay materials containing more than 35 percent passing the 200-mesh sieve. Five soil fractions are recognized: boulders, gravel, coarse sand, fine said, and combined silt and clay.

Whether a soil is "silty" or "clayey" depends on its plasticity index (PI). "Silty" is applied to fine material having a PI of 10 or less, and "clayey" is applied to fine material having a PI of more than 10.

Soils of about the same general load-carrying capacity and service are grouped together in seven basic groups designated A-1 through A-7. The suitability of soils for use as road subgrades decreases with an increase in numerical designation, except that A-3 is more suitable than A-2. Table 7-2 is a tabular form of the classification under consideration.

These seven basic soil groups are further divided into subgroups specified by a group index GI and the soil's Atterberg limits to permit within-group evaluations. Group indexes range from 0 for the best subgrades to 20 for the poorest. Increasing values of the index within each basic soil group reflect (a) the reduction of the load-carrying capacity of subgrades, and (b) the combined effect of increasing liquid limits LL and plasticity indexes PI and decreasing percentages of coarse material.

The group index rating is obtained by a formula based on the gradation LL and PI of the soil:

$$GI = 0.2a + 0.005ac + 0.01bd$$

where

a = that portion of the percentage passing No. 200 sieve greater than 35 percent and not exceeding 75 percent, expressed as a positive whole number (0 to 40)

b = that portion of the percentage passing No. 200 sieve greater than 15 percent and not exceeding 55 percent, expressed as a positive whole number (0 to 40)

c = that portion of the numerical liquid limit greater than 40 and not exceeding 60, expressed as a positive whole number (0 to 20)

d = that portion of the numerical plasticity index greater than 10 and not exceeding 30, expressed as a positive whole number (0 to 20)

7-7 UNIFIED SOIL CLASSIFICATION SYSTEM (Ref. 13)

The Unified Soil Classification System is based on the system developed by Dr. Arthur Casagrande for the Corps of Engineers during World War II. The original classification has been expanded and revised so that it now applies to embankments and foundations as well as to roads and airfields. It

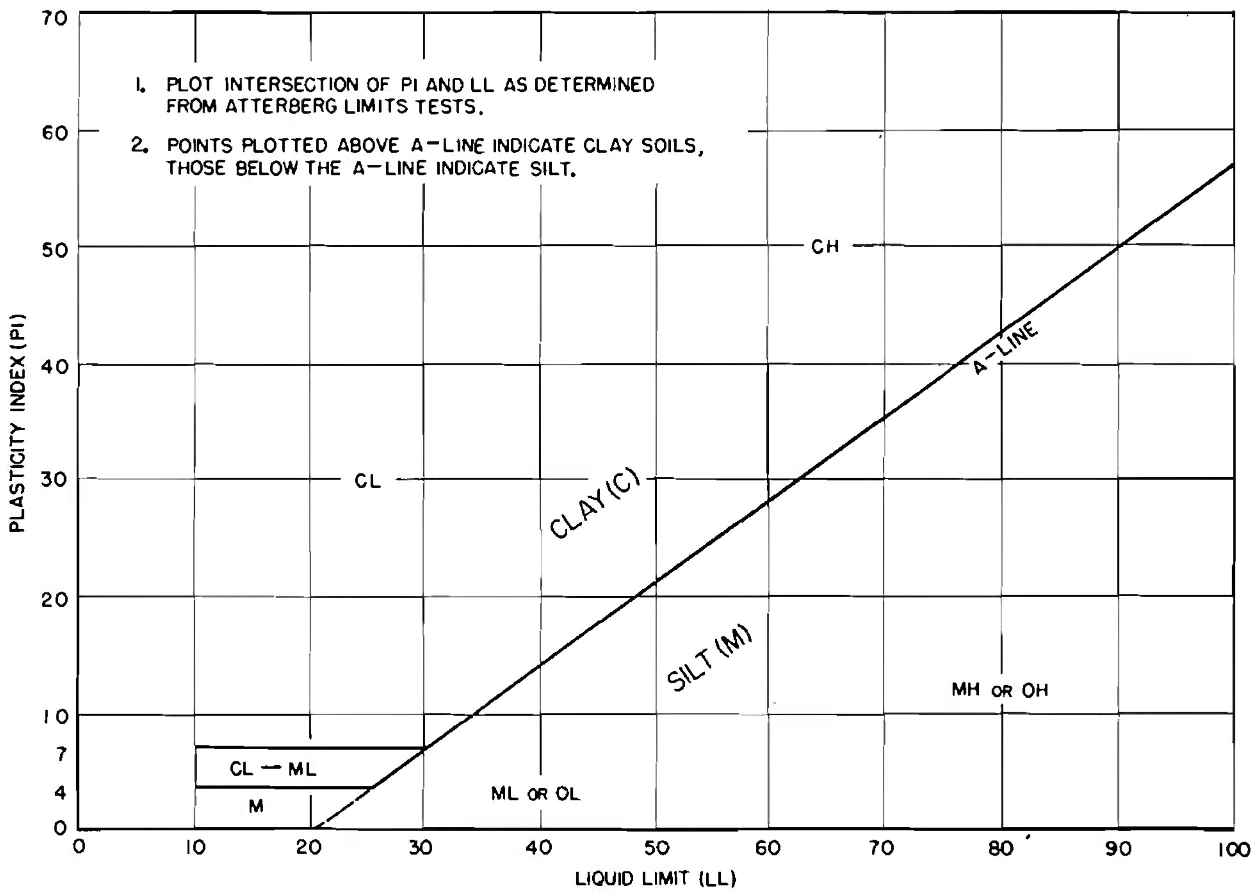


Figure 7-6. Plasticity Chart (from Corps of Engineers)

is used by both the Corps of Engineers and the U. S. Bureau of Public Roads.

The Unified Soil Classification System identifies soils according to their textural and plasticity qualities and groups soils with respect to their performances as engineering construction materials. The following properties form the basis of soil identification:

(a) Percentages of gravel, sand, and fines (fraction passing the No. 200 sieve).
(b) Shape of the grain-size distribution curve.
(c) Plasticity and compressibility characteristics.

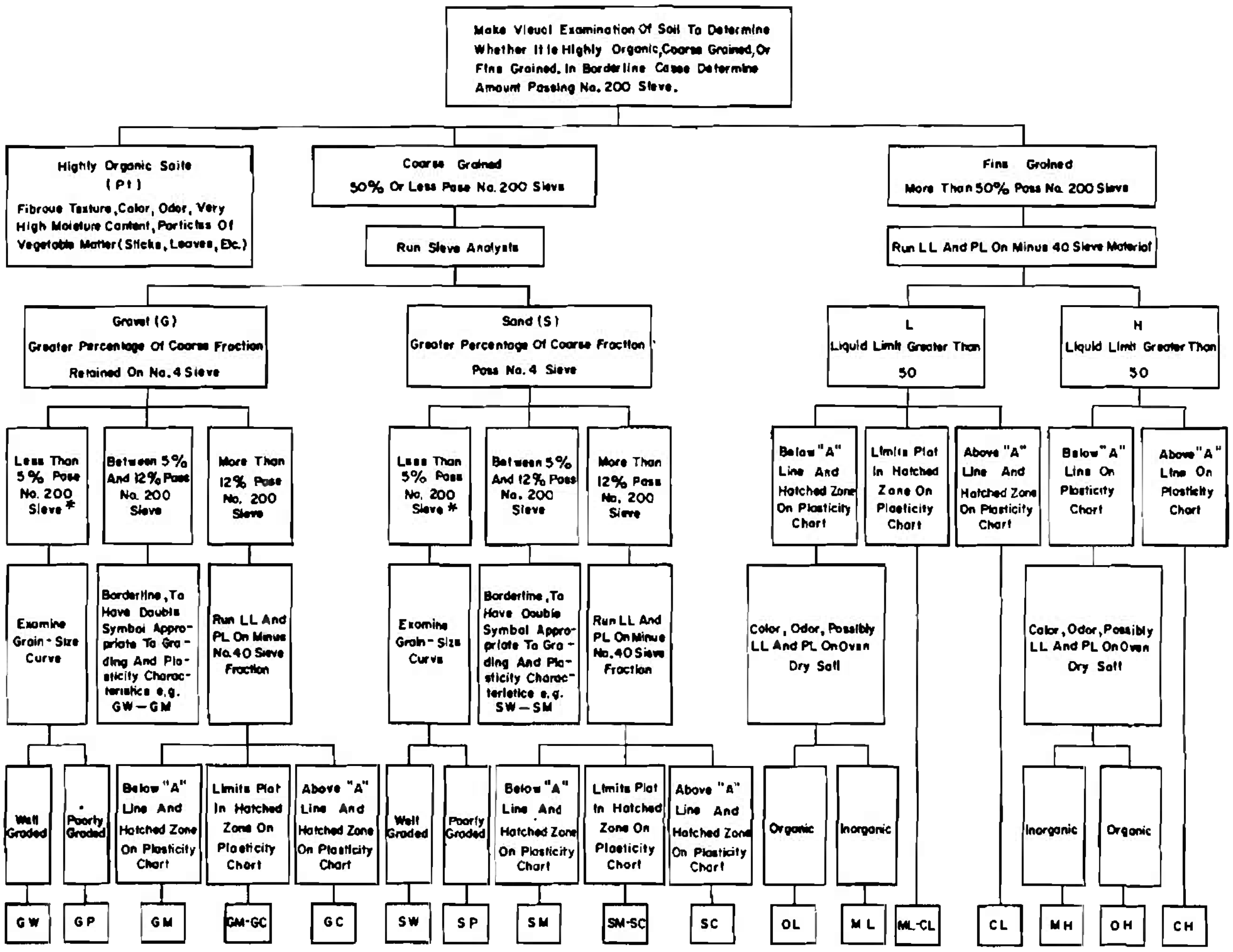
The soil is given a descriptive name and a double letter symbol indicating its principal characteristics. The symbols are as follows:

G—gravel
S—sand
M—silt
C—clay
W—well-graded
P—poorly graded
U—uniformly graded
L—low liquid limit
H—high liquid limit

Combinations of the letters designated above are used to identify soils. For example, GP indicates a poorly graded gravel.

Soil components used in the Unified Soil Classification System are:

Cobbles—above 3 inches
Gravel—3 inches to a No. 4 sieve
Coarse sand—No. 4 sieve to No. 10 sieve



NOTE: SIEVE SIZES ARE U.S. STANDARD
*IF FINES INTERFERE WITH FREE-DRAINING PROPERTIES USE DOUBLE SYMBOL SUCH AS G.W.-GM, etc.

Figure 7-7. Chart for Auxiliary Laboratory Identification Procedure

Medium sand—No. 10 sieve to No. 40 sieve
Fine sand—No. 40 sieve to No. 200 sieve
Fines—passing No. 200 sieve

The soils are divided as (a) coarse-grained soils, (b) fine-grained soils, and (c) highly organic soils. The coarse-grained soils (sands and gravels) contain 50 percent or less of material smaller than the No. 200 sieve and fine-grained soils contain more than 50 percent of materials smaller than the No. 200 sieve. Highly organic soils are fine-grained soils which contain a high percentage of organic matter.

The fine-grained soils are subdivided into silts M and clays C depending on their liquid limit and plasticity index. Silts are those fine-grained soils with a liquid limit and plasticity index that plot (Figure 7-6) below the *A* line; clays are those that plot above the *A* line with the exception of organic clays, for which the liquid limit and

TABLE 7-3
CLASSIFICATION OF SOILS FOR AIRPORT CONSTRUCTION*

	Mechanical Analysis					
		Material Finer Than No. 10 Sieve				
Soil Group	Retained on No. 10 Sieve, %	Coarse Sand, Pass No. 10, Ret. No. 60, %	Fine Sand, Pass No. 60, Ret. No. 270, %	Combined Silt and Clay, Pass No. 270, No. 270, %	Liquid Limit (LL)	Plasticity Index (PI)
E-1	0–45	40+	60–	15–	25–	6–
E-2	0–45	15+	85–	25–	25–	6–
E-3	0–45	–	–	25–	25–	6–
E-4	0–45	–	–	35–	35–	10–
E-5	0–45	–	–	45–	40–	15–
E-6	0–55	–	–	45+	40–	10–
E-7	0–55	–	–	45+	50–	10–30
E-8	0–55	–	–	45+	60–	15–40
E-9	0–55	–	–	45+	40+	30–
E-10	0–55	–	–	45+	70–	20–50
E-11	0–55	–	–	45+	80–	30+
E-12	0–55	–	–	45+	80+	–
E-13		Muck and peat-field examination				

*Federal Aviation Agency.

plasticity index fall below the *A* line. The silt and clay groups have secondary divisions based on whether the soils have a relatively low L or high H liquid limit. A liquid limit of 50 percent distinguishes between high and low compressible soils.

Figure 7-7 is a chart for classifying soil according to the Unified Soil Classification System.

7-8 FEDERAL AVIATION AGENCY CLASSIFICATION SYSTEM (Ref. 14)

The FAA has prepared a soil classification system based on the gradation analysis and the plasticity characteristics of soils. The textural classification is based on a grain size determination of the minus No. 10 material. In addition, the No. 270 mesh sieve is used to distinguish fine fractions of the soil from the coarse fraction. Soils are classified according to groups ranging from E-1 to E-13, depending on mechanical analysis, liquid limit,

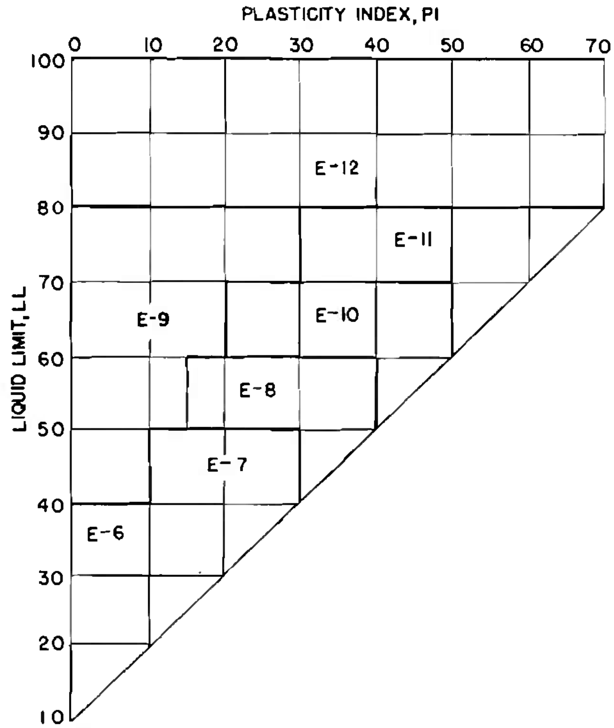


Figure 7-8. FAA Classification Chart for Fine-Grained Soils

and plasticity index data—see Table 7-3 and Figure 7-8. The soil components are designated as follows:

Coarse sand—pass a No. 10 sieve, retained on a No. 60 sieve.

Fine sand—pass a No. 60 sieve, retained on a No. 270 mesh sieve.

Silt and clay—pass a No. 270 mesh sieve.

SECTION III DETERMINATION OF GEOMETRIC TERRAIN VALUES

7-9 DISCUSSION OF THEORY (Ref. 15)

The characteristics of the terrain can be divided into two major areas: the physical properties of the earth as a material and the geometrical properties of the ground as a two-dimensional surface. The physical properties of the earth are those associated with its elasticity, plasticity, and trafficability. The treatment of the ground as a geometrical surface requires methods of analytically characterizing the ground surface in a realistic fashion. Such methods are found within the scope of the theory of probability and random functions. Recently, statistical analyses of road surfaces have been applied in determining the spectral properties of airport runways (Ref. 16).

The power spectrum of the surface appears to be the most readily available way of characterizing the surface statistically. A mathematical model that has been used to represent random surface conditions is the two dimensional analog of the Rice random functions.

$$z(x, y = \sum_{k=-n}^{n} c_k e^{i(u_k x + v_k y + \phi_k)} \tag{7-15}$$

where c_k, u_k, v_k are real numbers satisfying

$$c_k = c_{-k},\ c_o = 0$$
$$u_{-k} = -u_k$$
$$v_{-k} = -v_k$$

u_k, v_k have the dimension of length $^{-1}$

The φ_k's are independent random phases, uniformly distributed on the interval $(0, 2\pi)$ satisfying $\varphi_{-k} = -\varphi_k$. Hence, $z(x, y)$ is then a linear combination of cosine terms with random phases; conceptually this represents a ground surface full of miniature mounds and valleys.

Investigation has been undertaken to determine the nature of real ground spectra (Ref. 17) and make decisions regarding the characterizations of the surface. A device to measure ground profile based on a slope-integration survey method has been developed. A limited series of terrain profiles have been measured and the power density functions developed. The results indicate that terrain profile can, indeed, be described by a probabilistic techniques and that, although a terrain may vary in roughness, the general composition of the roughness is relatively uniform.

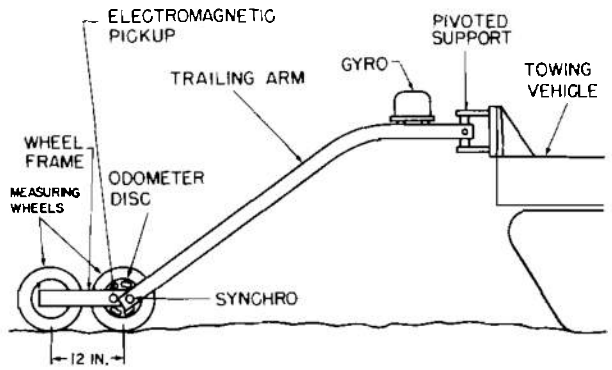


Figure 7-9. Terrain Geometry Measuring Apparatus (Ref. 18)

7-10 DESCRIPTION OF TERRAIN GEOMETRY MEASURING APPARATUS (Refs. 17, 18)

The apparatus for measuring terrain geometry is a towed assembly which utilizes the slope-integration method to continuously measure the slope of the ground over which it travels. The assembly is illustrated in Figure 7-9. It consists of two wheels mounted in tandem in a frame, an angle reference device, and an odometric device.

The pitch angle of the wheel frame is assumed to be identical to the slope of the ground over which the wheels are traveling. Ground slope is determined by the angle reference which consists of a synchro system which senses and records the pitch angle of the wheel frame. A synchro is attached to the lead wheel to indicate the angle between the wheel frame and the trailing arm. A second synchro is attached to a vertical gyroscope on the trailing arm to measure the angle between the trailing arm and the vertical reference. The two synchros are so connected electrically that their output voltage is proportional to the angle between the wheel frame and the vertical reference.

The odometer system provides measurement of distance traveled. A metal disk containing a number of evenly spaced slots is mounted on the front wheel axle. As the wheel rotates, these slots pass an electromagnetic pickup, the reluctance of which varies with its proximity to metal. The voltage output of the pickup closes a relay contact switch each time a slot passes the pickup. Each closure of the switch indicates an increment of travel of the wheel.

The signals from both the angular reference and odometric devices are recorded on magnetic tape. The general purpose data processing facility at the U.S. Army Tank-Automotive Center has developed the capability to convert this raw data into a form such that necessary computations can be carried out on a digital computer. The results are presented in the form of a plot of the x and y coordinates of the ground profile traversed by the apparatus.

SECTION IV SCALE-MODEL TESTING AND DIMENSIONAL ANALYSIS

7-11 GENERAL DISCUSSION (Ref. 12)

Scale model testing has proven invaluable in aerodynamic and hydrodynamic studies for many years. It provides a means for formulating and checking analytical work, and provides semi-empirical or fully empirical solutions to problems which resist straight-forward analytical treatment.

To a major extent, vehicle studies to date have been concerned with the testing of full-size prototypes. This has been due to the relatively small vehicle size and the relative ease of testing as compared to the testing of ships and aircraft. However, important drawbacks exist to full-size prototype testing such as the complexity of the full-size vehicle, the difficulty in exercising control over test conditions, the time necessary to complete a test vehicle, and the high costs that are involved. These can obscure the specific behavior patterns under study and preclude the testing of simplified concepts essential to a sound analysis.

The alternative to full-size testing is some form of analog testing. The most obvious analogs are scale models, although models of electrical analogy or high-speed computer simulations can be helpful in understanding some problems.

In general, all analog testing has certain advantages provided the development of basic apparatus, techniques, and interpretive understanding does not have to be done at the expense of a single job. The time and cost of obtaining a given amount of data will be less with the analog than with the prototype. These savings will accumulate rapidly in a long or extensive program involving many changes of the vehicle and the test conditions. The corollary to this is that more and better data can often be obtained from the analog. The cost of a model of electrical analogy is relatively little, consisting perhaps of only a circuit diagram which may be readily executed in existing apparatus. The cost of what is more usually considered a model, either to scale or distorted, is often quite high and in certain instances may be prohibitive. This is obviously a function of complexity, and all but the simplest objects of interest in motor-vehicle mechanics are almost uniformly mechanically complex. Performance determined by the configuration and running-gear details of the vehicle (as distinct from wear, ruggedness, engine cooling, etc.) can be conducted with scale-model tests and may show great savings in both time and money.

7-12 PRINCIPLES OF DIMENSIONAL ANALYSIS (Refs. 19, 20, 21)

The dimensional analysis approach attempts to arrive at a solution to a problem by obtaining functional relationships between the pertinent parameters or groups of parameters. The method's principal premise is that a phenomenon can be defined by a dimensionally correct equation relating the variables. Of itself, dimensional analysis can provide only qualitative results; however, when combined with experimental studies, prediction equations and quantitative results may be obtained.

The theorems on which dimensional analysis is based are:

(a) An equation which continues to remain true when the size of the fundamental units change can be expressed in a form such that the variables are arranged in dimensionless combinations.

(b) The possible number of dimensionless combinations which can be obtained is equal to the difference between the number of variables and the number of fundamental units (The Π Theorem). In exceptional cases the possible number of combinations may exceed this difference.

(c) Any dimensionless combination can be expressed as a product of the powers of the variables.

An application of these three theorems results in the general theorem that any general connection between the variables can be expressed in the form of an arbitrary function of all of the independent dimensionless products of the variable set equal to a constant.

In performing a dimensional analysis, the first step is to list the variables involved in the problem being studied. The dimensions of each variable are then written out in terms of the fundamental dimensional units. In the mechanical system these are mass M, length L, and time T; and sometimes temperature t. In some instances, force F ($F = MLT^{-2}$) is used rather than mass. The number of independent dimensionless products which can be formed is then, by theorem (b) equal to the number of variables less the number of fundamental dimensionless units (M, L, T, etc.) listed. The dimensionless products may be formed by selecting a number of dimensionally independent variables equal to the number of fundamental units and combining these in dimensionless form with each of the remaining variables. The selection of the dimensionally independent variables should be from those considered most important to the problem but not those desired to be studied, and all of the fundamental units should be included in the variables selected. The procedure is best illustrated by an example.

Consider the problem of steady flow of a liquid through a round pipe under a constant head of pressure. The variables involved are: pipe diameter D (dimension L), pipe length l (dimension L), pressure head h (dimension $ML^{-1}T^{-2}$), fluid viscosity ν (dimension $ML^{-1}T^{-1}$), and quantity of flow per unit time Q (dimension L^3T^{-1}). We, therefore, have five variables (D, l, h, ν, and Q) expressed in three fundamental units (M, L, and T; hence, according to the theorem given under (b), there are two independent dimensionless products. The dimensionally independent variables selected are: pipe diameter D, pressure head h and fluid viscosity ν. Selection of both pipe diameter and pipe length as dimensionally independent variables would not be permissible because both are of dimension L. The two dimensionless products will then be of the form $[lD^{\alpha_1} h^{\alpha_2} \nu^{\alpha_3}]$ and $[QD^{\beta_1} h^{\beta_2} \nu^{\beta_3}]$; or, expressed in dimensional form, the two products are, respectively, $[L(L)^{\alpha_1} (ML^{-1}T^{-2})^{\alpha_2} (ML^{-1}T^{-1})^{\alpha_3}]$ and $[L^3T^{-1}) (L)^{\beta_1} (ML^{-1}T^{-2})^{\beta_2} (ML^{-1}T^{-1})^{\beta_3}]$. There must be dimensionless in M, L, and T. It is convenient, in general, to group the expressions in tabular form as follows:

Table 1

	D^{α_1}	h^{α_2}	ν^{α_3}	l
M	0	$+\alpha_2$	$+\alpha_3$	0
L	$+\alpha_1$	$-\alpha_2$	$-\alpha_3$	1
T	0	$-2\alpha_2$	$-\alpha_3$	0

Table 2

	D^{β_1}	h^{β_2}	ν^{β_3}	Q
M	0	$+\beta_2$	$+\beta_3$	0
L	$+\beta_1$	$-\beta_2$	$-\beta_3$	3
T	0	$-2\beta_2$	$-\beta_3$	-1

Table 1 gives the equations (Theorem c):

$0 + \alpha_2 + \alpha_3 + 0 = 0$, the condition on the exponent of M,

$\alpha_1 + \alpha_2 - \alpha_3 + 1 = 0$, the condition on the exponent of L, and

$0 - 2\alpha_2 - \alpha_3 + 0 = 0$, the condition on the exponent of T, from which $\alpha_2 = \alpha_3 = 0$, and

$\alpha_1 = -1$

The first dimensionless product is then

$[lD^{-1}h^0\nu^0] = \dfrac{l}{D}$, which, of course, could have been seen by inspection.

Table 2 gives the equations:

$$0 + \beta_2 + \beta_3 + 0 = 0,$$
$$\beta_1 - \beta_2 - \beta_3 + 3 = 0, \text{ and}$$
$$0 - 2\beta_2 - \beta_3 - 1 = 0,$$

from which $\beta_1 = -3$,

$$\beta_2 = -1, \text{ and}$$
$$\beta_3 = 1$$

The second dimensionless product then is

$$\left[QD^{-3}h^{-1}\nu \right] = \frac{Q\nu}{D^3h}$$

The general solution, therefore, is a functional relationship of the two dimensionless products

$$\psi_1 \left[\frac{l}{D}, \frac{Q\nu}{D^3h} \right] = \text{Constant}$$

or

$$\frac{Q\nu}{D^3h} = \psi_2 \left(\frac{l}{D} \right)$$

or

$$Q = \frac{D^3h}{\nu} \psi_3 \left(\frac{l}{D} \right)$$

where ψ_1, ψ_2, and ψ_3 are arbitrary functions. Hence, the delivery Q varies directly as the pressure head and inversely as the viscosity.

7-13 THEORY OF MODELS (Refs, 12, 19)

The use of models is particularly well adapted to the design of large, expensive, or complicated units but its satisfactory use requires a clear understanding of the principles involved in the relationship between model and prototype. Unless properly interpreted, the information obtained from models may be quite misleading.

The principles which underlie the proper design and construction, operation, and interpretation of test results from models comprise the theory of similitude. The theory of similitude includes consideration of the conditions under which the behavior of two separate systems will be similar and the techniques of accurately predicting results on one from observations on the other. The theory is developed by dimensional analysis.

There are four types of models: true models, adequate models, distorted models, and dissimilar models.

(a) *A true model* is a scaled reproduction of prototype. Models of this type are used to predict performance characteristics of the prototype.

(b) An *adequate model* is used to simulate particular characteristics of the prototype and permits study only of those characteristics.

(c) *Distorted models* are imperfect simulations of the prototype. Model performance cannot be used directly to predict prototype performance; corrections of some form must first be applied to account for the distortion.

(d) *Dissimilar models* bear no apparent resemblance to the prototype but give accurate indications of prototype behavior. For example, the characteristics of a vibrating mechanical system may be simulated electronically on a computer.

The general theory of models is developed from Buckingham's Pi Theorem. The most general form of equation describing a phenomenon may be written as:

$$\Psi[\pi_1, \pi_2, \ldots . \pi_{(n-k)}] = 0 \qquad (7\text{-}16)$$

where

π = dimensionless product
n = number of variables
k = number of fundamental units

Since this equation is completely general, it applies to all systems which are functions of the same variable; two such systems are the prototype and model. To study any one dimensionless product in relation to the others, Equation 7-16 may be written

$$\pi_1 = f[\pi_2, \pi_3 \ldots . \pi_{(n-k)}] \qquad (7\text{-}17)$$

For the particular systems, prototype and model,

$$\pi_{p1} = f[\pi_{p2}, \pi_{p3}, \ldots . \pi_{p(n-k)}] \text{ and} \qquad (7\text{-}18)$$

$$\pi_{m1} = f[\pi_{m2}, \pi_{m3} \ldots . \pi_{m(n-k)}] \qquad (7\text{-}19)$$

The equation for predicting π_{p1} from π_{m1} is then

$$\frac{\pi_{p1}}{\pi_{m1}} = \frac{f[\pi_{p2}, \pi_{p3} \ldots . \pi_{p(n-k)}]}{f[\pi_{m1}, \pi_{m2} \ldots . \pi_{m(n-k)}]} \qquad (7\text{-}20)$$

7-14 CONDITIONS FOR MODEL TESTING (Ref. 19)

In order for one system to be a true model of another, certain conditions of similarity must exist. Specifically, in the general Equation 7-16

$$\psi[\pi_1, \pi_2, \pi_3 \ldots . \pi_{(n-k)}] = 0 \qquad (7\text{-}21)$$

$$\pi_{p1} = \pi_{m1}$$
$$\pi_{p2} = \pi_{m2}$$
$$\pi_{p3} = \pi_{m3}$$
$$\vdots \qquad \vdots$$
$$\pi_{p(n-k)} = \pi_{m(n-k)}$$

Hence, similarity exists only if each of the dimensionless products has the same numerical value for both the model and the prototype.

Each dimensionless product π_i is formed as a product of the powers of certain variables. If

$$\pi_{pi} = Q_1^{a_1}\ Q_2^{a_2}\ Q_3^{a_3}\ Q_4 \text{ and}$$
$$\pi_{mi} = Q_4^{a_1}\ Q_2^{a_2}\ Q_3^{a_3}\ Q_4$$

then $\pi_{pi} = \pi_{mi}$ only if the scaling relationships

$$\bar{Q}_1 = S_1Q_1$$
$$\bar{Q}_2 = S_2Q_2$$
$$\bar{Q}_3 = S_3Q_3$$
$$\bar{Q}_4 = S_4Q_4$$

exist such that

$$S_1^{a_1}\ S_2^{a_2}\ S_3^{a_3}\ S_4 = 1 \qquad (7\text{-}22)$$

Since there are $(n-k)$ independent dimensionless products, there will be $(n-k)$ scale factor relationships of the type in Eq. 7-22; hence, the number of fundamental scale factors is k, the same as the number of fundamental dimensions.

To illustrate, consider the use of a model for the example problem of paragraph 7-12 when the fluid of interest is water. For the model, it is desired to use mercury flowing through a pipe of 1/10th the diameter of the prototype and to restrict the quantity of flow to 1/1000th of the flow of the prototype. The fundamental scale factors are then

$$S_D = 0.1$$
$$S_\nu \approx 1.5$$
$$S_Q = 0.001$$

The dimensionless products are $\left[\frac{D}{l}\right]$ and $\left[\frac{Q\nu}{D^3h}\right]$

The scale factors must satisfy

$$\frac{S_D}{S_l} = 1$$

$$\frac{S_QS_\nu}{S_DS_h} = 1$$

From which $S_l = 0.1$ and $S_h = 0.015$. This means that the model must be 1/10th the length of the prototype and the mercury must be under a pressure 0.015 times that of the water in the prototype.

If the model is a distorted model rather than a true model, the conditions for test become more complex. For a distorted model,

$$\pi_{p1} = \delta\pi_{m1}$$

To evaluate δ, the ratio of the functions,

$$\pi_{p1} = f[\pi_{p2}, \pi_{p3}, \ldots \pi_{p(n-k)}]$$
$$\pi_{m1} = f[\pi_{p2}, \pi_{p3}, \ldots \pi_{m(n-k)}]$$

must be evaluated. This requires either experimental evidence or knowledge of how the π_i influence the function.

7-15 REDUCTION TO PRACTICE

The problem of testing model vehicles in soil is complex because, not only are vehicles complex mechanisms, but, theoretically, it is necessary to scale the soil properties—an impossible problem. The objective of the dimensional analysis-model approach is to obtain the functional relationships between parameter groups. The success of the method relies to a great extent on the ability to select all of the important parameters and to omit the unnecessary or redundant ones. In this regard, soil parameters have presented the greatest problem.

In 1951, Nuttall (Ref. 22) began investigating wheel performance using scale models in the laboratory. Reasonable correlation was reported over a limited range of test conditions but there was some question as to the suitability of the soil parameters used. A recent paper by Nuttall and McGowan (Ref. 23) summarizes the current thinking which has evolved from the earlier work. The analysis pertains to the entire vehicle performance, not just that of the wheel. The following list of factors was chosen for inclusion in the scaling relationships.

Factors relating to the vehicle

(a) Size, denoted by a characteristic dimension such as wheel diameter
(b) Geometry, denoted by ratio of important dimensions such as wheelbase and length, to the characteristic dimension
(c) Gross vehicle weight

Factors relating to the operation of the vehicle

(a) Speed
(b) Slip

Factors relating to the soil

(a) Depth of layer
(b) Before-collapse structural cohesion
(c) After-collapse dynamic shearing resistance (cohesion and friction)
(d) Initial density
(e) Plastic kinematic viscosity
(f) Slope of surface of material
(g) Coefficient of friction between soil and vehicle
(h) Shear stress-strain parameter

Factors relating to the system as a whole

(a) Acceleration of gravity

Dependent variables

(a) Vehicle sinkage
(b) Trim of vehicle
(c) Drawbar pull

With so many parameters, especially for the soil, some simplification was necessary to provide useful equations. The following factors were among those eliminated:

(a) Speed, because there was no evidence that this factor, *per se*, influences performance. (This assumption may be wrong.)
(b) Slope of surface, assuming level terrain in general.
(c) Coefficient of friction, assuming that it is constant.
(d) Soil density, on the basis that its effect is negligible.
(e) Cohesion portion of dynamic shearing resistance, assuming application to sand and snow only.

With this type of simplification the general equation for the problem was reduced to the form

$$\psi\left[\frac{z}{d}, \frac{W}{c_s d^2}, \varphi, S, \frac{h}{d}\right] = 0 \qquad (7\text{-}23)$$

z = vehicle sinkage
d = wheel diameter
W = gross vehicle weight
c_s = soil cohesion
φ = dynamic internal friction
S = slip ratio
h = effective depth of material

If the soil is the same for the model, and if φ may be considered constant with depth; then the problem of scaling becomes primarily one of keeping the factor $W/c_s d^2$ constant. This, in turn, reqnires a suitable measure of c_s. To circumvent the problem of trying to assign an absolute measure of c_s to a given soil which will account for variations with depth, a method of obtaining relative values was devised using plate bearing tests. Basically this is done by obtaining load-penetration curves for two circular plates whose diameter ratios are in proportion to the geometric scaling ratio between model and prototype. The relative value of c_s are chosen so that the data from the two plates are coincident when plotted in the form q/c_s vs z/D, where D is the plate diameter.

The basic approach appears to be quite reasonable. The analysis begins with a long list of soil parameters but, in the final equations, these reduce essentially to a single parameter determined from vertical load-penetration tests. The system for obtaining relative values of c_s recognizes that, even in the same soil area, nonuniformity of soil with depth affects the model and prototype differently. Some such method as this will probably be required for valid scaling. However, it still may be necessary to use a more extensive system of soil parameters and to incorporate rate effects.

There has been some objection to the use of angle of internal friction and structural cohesion terms to represent the soil conditions on the basis that these parameters do not account for the scale effects between model and prototype, and that they are not suitable for unconsolidated surface soils. It has been recommended that the sinkage parameters—k_c, k , and n—be used instead (see Chapter 4) and dimensional analysis equations be derived on this basis (Ref. 24). In addition to the three sinkage parameters the other parameters used are:

(a) Wheel-to-soil coefficient of friction μ
(b) Wheel diameter d
(c) Wheel aspect ratio α
(d) Soil depth D
(e) Wheel load W

The selected dependent variables are wheel rolling resistance R and wheel sinkage z. The resulting equations are of the form

$$\frac{R}{W} = \left[\frac{d}{D}, \frac{W}{d^{n+2}k_\phi}, \frac{k_c}{dk_\phi}, n, \alpha, \mu\right] \qquad (7\text{-}24)$$

These equations require that the soil properties be changed to accommodate a change in size of the model. This requirement places a great limitation on the use of scale models to predict prototype performance because (1) it may not be possible to provide the necessary soil properties and (2) even if it were, there is the strong possibility that other soil properties, not directly accounted for by the sinkage parameters but influencing wheel behavior, would not be properly scaled. In fact, to date, this approach has only been used in dry sand where $k_c = 0$, hence, making it possible to satisfy the scaling relationships with the same material.

The arguments made for selecting the sinkage parameters for representing the soil conditions rather than using the approach by Nuttall are subject to question. The consequences of the former are highly undesirable (which does not make them either right or wrong). A careful examination of Nuttall's method reveals that structural cohesion denotes, with a single parameter, the effective difference in soil properties between two plates whose sizes scale as the model and prototype, and that this parameter is equivalent to the combined effects of k_c, k_ϕ , and n. Hence, it appears that c_s not only does account for scaling but also eliminates two additional soil parameters. Furthermore, it does not appear that the use of c_s restricts the test to any particular soil type. It can be argued that c_s is not adequate to represent the soil conditions but it does not appear that this parameter is any less comprehensive than the three sinkage parameters. The implications of such a consequence are significant and should be examined further.

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PART THREE
SUSPENSION SYSTEMS, COMPONENTS, AND DESIGN DATA

CHAPTER 8

FUNDAMENTAL SUSPENSION SYSTEM CONCEPTS

SECTION I SUSPENSION SYSTEM CLASSIFICATIONS AND CHARACTERISTICS* (Ref. 1)

8-1 PASSIVE SUSPENSION SYSTEMS

8-1.1 GENERAL DESCRIPTION

A *passive* suspension system is a system combining fixed springing, either linear or nonlinear, with conventional shock absorber damping characteristics, either velocity or displacement sensitive.

8-1.2 CHARACTERISTICS

Although significant improvements have been made in military vehicle suspension design, from the rigid suspension systems of the first tanks through successor vehicles having vertical or horizontal volute spring bogie-type suspension units to the independently sprung torsion bar suspension units which are currently used on tanks, all of the present standard military vehicles have passive systems. These suspension systems are limited with respect to cross-country mobility, since spring and damping rate requirements vary with respect to changes of the terrain conditions. Computer studies have shown that a passive suspension system significantly limits the maximum speed of a vehicle over rough, changing terrain. These studies have demonstrated that a controlled variable suspension system would permit higher maximum speeds over rough terrain. This has led to analytical and experimental research programs on nonpassive suspension systems.

*Written by Nicholas R. Rome of the IIT Research Institute, Chicago, Ill.

8-2 SEMI-ACTIVE SUSPENSION SYSTEMS

8-2.1 GENERAL DESCRIPTION

A *semi-active* suspension system is a system having modulated spring and/or damping characteristics during dynamic interaction between the ground-contacting elements, e.g., road wheels, and the terrain. Dynamic actions such as accelerations of the sprung mass are sensed to control the spring and/or damping characteristics to optimize performance.

8-2.2 CHARACTERISTICS

A semi-active suspension system requires an acceleration sensing system and an elastic support system that can be continuously modulated. The same general types of components as are used in passive suspension systems comprise semi-active systems. The chief difference between the two is that the elastic characteristics of the latter are varied to bring them into closer agreement with the terrain requirements. The result is a more uniform ground loading, a more comfortable ride, and increased cross-country speed. The degree of improvement is dependent upon the response characteristics of the sensors and the modulated elements.

8-3 ACTIVE SUSPENSION SYSTEMS

8-3.1 GENERAL DESCRIPTION

An *active* suspension system is a system that is power-actuated to approach constant wheel loading through modulation of wheel displacement and springing and/or damping characteristics. The desired characteristics are selected simultaneously with or prior to the dynamic interaction of the ground contacting elements and the terrain. Terrain or inertial sensors are used to actuate the system.

8-3.2 CHARACTERISTICS

Research programs are being conducted on various methods of sensing the terrain conditions. Types of sensors under investigation are (a) mechanical, (b) electrical, (c) electromagnetic, (d) sonic, and (e) inertial. Limited laboratory investigations have covered the following specific methods of terrain sensing: (a) sonic reflectance, (b) microwave reflection and interference, (c) optical radar, (d) infrared, (e) ultra-violet, and (f) capacitance.

Various elastic supports, damping devices, and linkage arrangements for active suspension systems are under consideration. Included are electro-hydraulic, electropneumatic, electromechanical, and combinations of these systems.

SECTION II MAJOR INFLUENCES ON MOBILITY AND RIDE CHARACTERISTICS (Ref. 2)

The primary functional objective of a military vehicle suspension system is to improve mobility by permitting an increase in the speed at which the vehicle may traverse uneven terrain without detrimental or unacceptable impacts and vibrations to the vehicle and to the personnel, and without intolerable variations of the traction with the terrain. The degree of attainment of the functional objective for a particular vehicle may be restricted by physical features which are essential to the utility of that vehicle. However, for a given vehicle, the achievement of the best available mobility and ride characteristics results from an engeneering approach in which due consideration is given to the significance of the relevant factors.

8-4 TERRAIN CONTOUR

One of the first factors to be considered is the fact that, for any vehicle, the portion of the suspension system that follows the irregular contour of the terrain surface is subjected to relatively large and erratic vertical accelerations. These accelerations cause a variation in the terrain loading. The magnitude of the load variations is a function of the acceleration and the mass of the suspension elements forced to follow the terrain contour. Hence, to minimize the variations of terrain loading and the resulting variation in the tractive effort, the unsprung/sprung mass ratio for a given vehicle should be as small as practicable. A more extensive discussion of terrain and its influence upon the vehicle is given in paragraph 2-10.

8-5 UNSPRUNG MASS

The unsprung mass is another major factor that influences vehicle mobility and ride characteristics. It is defined and its significance is discussed in paragraph 1-19.1 of Chapter 1. In general, it is comprised of those components of a vehicle that are not supported by the primary elastic elements of the vehicle. Hence, in a wheeled vehicle, its principal components are the wheels, axle assemblies, and proportionate parts of the springs, shock absorbers, and suspension linkages; and, in a tracked vehicle, its principal components are the road wheels, road wheel spindles, and proportionate parts of the road wheel arms, shock absorbers, and track. The magnitude of the unsprung mass affects wheel dance which, in turn, has a direct influence on such things as terrain loading, traction, steering control, and secondary vibrations transmitted to the body of the vehicle. It is generally agreed that a smaller (lighter) unsprung mass is more advantageous than a heavier one. For further discussion of this subject see paragraph 1-19.1.

8-6 SPRUNG MASS

The main portion of the vehicle, which includes all of the mass other than the previously described unsprung masses, is called the sprung mass. This principal mass system is suspended and carried on the running gear by means of an elastic support system.

The significant properties of the sprung mass system are the magnitude of the mass, the location of the center of gravity, the respective moment of inertia about each of two normal axes through the roll and pitch centers (these axes are the longitudinal and the lateral axes), and the dimensional constant which expresses the relationship between the moment of inertia about the lateral axis and the mass. See paragraph 1-19.1 for a further discussion of sprung mass.

8-7 ELASTIC SUPPORT SYSTEM

The elastic support system is positioned between the unsprung elements which tend to follow the terrain contour and the principal or sprung mass. The elastic system should permit relative motion between the sprung and unsprung masses such that the restraint is independent of the actual velocities or accelerations of those motions. Furthermore, the elastic support system should return the sprung mass to a normal static position after the disturbing forces are removed.

The term suspension system usually includes the entire assembly of linkages and components between the vehicle frame, body or hull, and the ground as discussed in Chapter 1, Sect. III.

The term elastic support system is used to describe the elements of the suspension system that, collectively, elastically support and/or elastically restrain the sprung mass system relative to the road wheels.

The significant characteristics of the elastic system are the vertical rate, the effective center of the elastic system, the angular rate about a lateral axis through the effective center of the elastic system, the angular rate about a longitudinal axis passing through the roll center, and the dimensional constant which expresses the relationship between the lateral axis angular rate and the vertical rate.

Many of the components of a vehicle other than the elastic suspension components, including the hull or frame, will experience appreciable elastic deformation under operational dynamic loading. The elastic deformation of major components, such as hulls and tracks of track-laying vehicles, can be of such magnitude as to significantly affect the performance of a vehicle. Experience has shown that under severe cross-country service elastic deformation of tank hulls has been sufficient to cause structural failure of the hull at the gun traveling locks with the gun tube in traveling position.

Although the elastic behavior of all vehicle components must be considered in the design analyses, this handbook will be limited to discussions of the elastic support system *per se* and to the track and drive sprocket system.

8-8 PRIMARY VIBRATIONS

A general discussion of primary vibrations is given in paragraph 1-19.6 of Chapter 1. Basically, they are the oscillations experienced by the sprung mass of a vehicle that originate in the interaction of the vehicle's ground-contacting elements (wheels or tracks) with terrain irregularities and are transmitted to the sprung mass by the vehicle's elastic support system. The effect of the primary vibrations upon the vehicle, its cargo, and its crew depends upon the frequency and amplitude. Higher frequencies and greater amplitudes increase the acceleration forces to which vehicle components, cargo, and crew are subjected. Since the crew is generally the least tolerant to this environment, maximum frequency and amplitude are usually limited by human factors considerations. A discussion of these is given in paragraph 2-7 of Chapter 2.

Within the human tolerance range, the primary vibration frequencies appropriate to particular vehicles depend upon the intended purpose of the vehicle. A relatively stiff suspension system is generally more conducive to stability during rapid maneuvers and in side-slope operations. Such a suspension system, however, has a higher natural frequency. Hence, where superiority of performance and maneuverability are vital to the mission, as is the case with fighting vehicles, it is reasonable to design for a higher primary vibration frequency than would be considered desirable for pleasure vehicles.

Furthermore, it is desirable to have similar primary vibration frequencies for the pitch and vertical (bounce) oscillations. One reason is to avoid rapid fluctuations in the phase relationships of the two vibrations, inasmuch as this condition results in spasmodic and erratic motions and accelerations. Another reason is that the lower frequency (if the two differ) is the limiting factor on the speed at which the vehicle can safely traverse rough terrain, that limit being the speed at which the terrain undulations occur in resonance with the lower frequency vibrations.

If the total sprung mass is considered as suspended at either end by separate elastic systems (coupling effects neglected for simplicity), the question arises as to the relationship between the natural frequencies of the front and rear suspensions. Three relationships are possible; namely:

(a) Natural frequency at front higher than at rear.
(b) Natural frequencies at front and rear equal.
(c) Natural frequency at front lower than at rear.

When either end of a vehicle is disturbed vertically, as when encountering a bump or pothole, it will vibrate at the natural frequency of the suspension at that end of the vehicle. Since damping is always present (inherent plus deliberately added), the vibration will have a decreasing amplitude—assuming that it receives no further disturbance. The decrease is approximately exponential. Plots can be made of the exponentially diminishing sine curve displacements of each end of the vehicle. If the wheelbase and speed of the vehicle are known, the time interval between front and rear wheels encountering the disturbance can be calculated and the phase relationship between the two curves can be shown. The difference in amplitude of the two curves at any instant represents the pitch of the vehicle. This difference, when plotted against a time scale, will show graphically the pitching of the vehicle caused by the vertical disturbance.

When this procedure is applied to the aforementioned three conditions, the following pitch-response characteristics are demonstrated.

(a) The first condition results in a tendency for the pitch amplitude to build up to a maximum occurring about one or two cycles after the start of the disturbance.
(b) The second condition results in some tendency to pitch but at a constantly diminishing amplitude.
(c) The third condition results in practically no tendency to pitch after the rear wheels passed the obstacle.

Experience has confirmed these general relationships between front and rear natural frequencies. The ideal quantitative relationship depends upon the vehicle speed range, length of wheelbase, and the range of suspension frequencies that are practical and appropriate to the specific vehicle.

8-9 SECONDARY VIBRATIONS

The secondary vibrations of the suspended portion of the vehicle are those induced by the oscillation of vehicular components such as road wheels and drive sprockets. The secondary vibrations are normally of higher frequency than the primary vibrations. Road wheel vibration is discussed in paragraph 1-19.5. Secondary vibrations may be induced by sprocket tooth impacts caused by pitch differences in the sprocket and the track, or by the inertial opposition of the track to linear pulsations forced by the alternating change of the effective sprocket radius. Additional discussion of this subject is given in paragraph 1-19.6 of Chapter 1.

8-10 WHEEL DANCE

As previously stated, wheel dance is one of the principal sources of secondary disturbances and vigrations which are transmitted to the sprung mass. The phenomenon of wheel dance or wheel hop occurs when an independently sprung road wheels oscillates at the natural frequency of a vibratory system. Usually, each road wheel has an elastic tire to cushion the impacts between the wheel and the track shoes and, although each wheel is suspended between its tire and a vehicle spring, the spring rate of the tire is the dominant elastic factor associated with wheel dance. Further discussion of wheel dance is given in paragraph 1-19.5.

8-11 DAMPING

(Also see Chapter 1, paragraph 1-19.4 and Chapter 10, Section VIII.)

8-11.1 PURPOSE

Fluctuations of the elastic support system, as a vehicle traverses uneven terrain, induce primary vibrations of the sprung masses. These vibrations must be damped to prevent progressive growth, especially if the inducing impulses are in resonance with one of the primary vibrations. In other words, the total damping system must be capable of dissipating the vibratory energy at a rate equal to the energy input rate of the inducing impulses; otherwise, the amplitude of the oscillations will increase until the vehicle is unmanageable. Thus, the speed of a given vehicle can be limited to that for which the vibratory energy input does not exceed the total available rate of energy dissipation by the damping system.

Usually, the degree of damping provided for the primary vibrations of a vehicle is a compromise between sufficient damping and undesirable harshness, or the degree of damping may be restricted by the capacity of the shock absorbers. Harshness is produced by overdamping.

The principal function of the damping system is the control or modification of the vertical separation of the sprung and unsprung masses so that primary vibrations are repressed, thereby permitting greater vehicle speeds and mobility. These benefits are achieved by virtue of reduced vertical and angular oscillations of the body or hull and reduced variation of the terrain loading. The damping system also reduces undesirable wheel dance.

The desirable type of damping for large amplitude vibrations is that for which the rate of energy dissipation is related to the existing vibratory energy, both being proportional to the second power of the amplitude of oscillation.

8-11.2 FRICTION DAMPING

Friction damping of a vibratory system is characterized by a damping force that is independent of the velocity and acceleration of the oscillating mass. For a reference vibratory system, the rate of amplitude decay is constant; i.e., the curve of amplitude decay, which defines the amplitude limits of the successive oscillations, is an inclined straight line. Hence, the amplitude decrement of the successive oscillations is a constant.

The energy dissipated during each oscillation is directly proportional to the amplitude, whereas the existent vibratory energy is proportional to the second power of the amplitude; hence, frictional damping tends to be unsuitable for the principal damping requirements of a vehicle. In general, friction damping forces which effect suitable energy dissipation for medium amplitude vibrations tend to be inadequate for large amplitude vibration and are excessive for small amplitude vebration. However, the presence of light friction damping is desirable to damp wheel dance in order to minimize the inducement of secondary vibrations in the sprung portion of the vehicle. The friction producing system tends to induee secondary vibrations in the suspended portion of the vehicle, however, these disturbances are normally insignificant compared to the dynamic disturbances of undamped wheel dance. The presence of light friction damping fosters general steadiness of the sprung mass of the vehicle, especially with regard to the suppression of rythmic float and restless buoyancy of the suspended portion over relatively smooth terrain.

8-11.3 VELOCITY DAMPING

Velocity damping is characterized by damping forces which vary with velocity. Velocity damping is most commonly produced by means of hydraulic shock absorbers. Three basic types of velocity damping can be described. These are classified according to the velocity-force relationship as (a) viscous damping, (b) degenerate viscous damping, and (c) hydraulic damping.

8-11.3.1 Viscous Damping

Viscous damping is characterized by a resisting force which is linearly proportional to the velocity of the oscillating mass. In practice it is found where there is relative motion between two well-lubricated surfaces and where a viscous fluid is forced through a relatively long passage of small cross sectional area.

8-11.3.2 Degenerate Viscous Damping

Degenerate viscous damping is characterized by a resisting force which is proportional to a power of the velocity less than unity.

8-11.3.3 Hydraulic Damping

Hydraulic damping is characterized by a resisting force which is proportional to the square of the velocity (Ref. 16). Hydraulic damping occurs, in practice, when a fluid of relatively low viscosity is forced through a sharp-edged orifice.

Figure 10-33 shows the velocity-force relationship of the preceding three damping types.

8-11.4 INERTIA DAMPING

Inertia damping is characterized by a resisting force which is linearly proportional to the rate of the acceleration of the oscillating mass. For inertia damping, the curve of amplitude decay is a logarithmic curve similar to that of velocity damping. The rate of energy dissipation is directly related to the existent vibratory energy, both being proportional to the amplitude of the oscillation. These similarities to velocity damping indicate that inertia damping is also favorable for damping large amplitude vibrations. For vehicular use, inertia damping systems need the addition of light friction damping to subdue the small amplitude oscillations.

The damping force of an inertial damping system is not only proportional to the accelerating force but directly opposes that force; whereas, the damping force of velocity damping lags behind the accelerating forces. Inertia damping also tends to reduce the frequency of the vibration for a given system.

SECTION III SIGNIFICANT PROPERTIES OF THE MASS SYSTEM (Ref. 3)

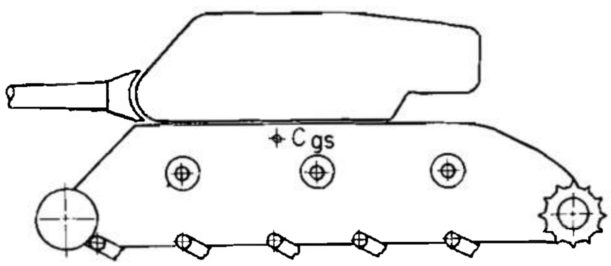


Figure 8-1. Sprung Mass of Track-laying Vehicle Without the Track

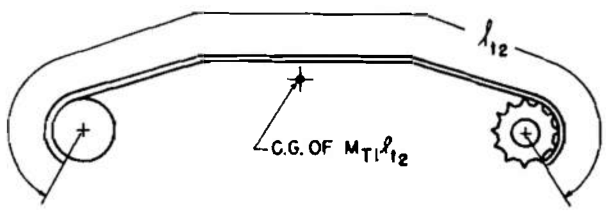


Figure 8-2. Suspended Portion of Track

8-12 MAGNITUDE

The significant properties of the basic mass system (or sprung mass) can be defined and described by considering a typical track-laying military vehicle. As stated in the previous section, a portion of the suspension system of a vehicle may be a part of the sprung mass. In the case of the tracked vehicle, the basic mass system includes the suspended portion of the track. However, for the determination of the sprung mass, the center of gravity of this mass, and the moment of inertia about the center of gravity it is expedient to first determine the properties of the sprung mass without the track, and then to modify those properties by adding the effects of the suspended portions of the track.

Figure 8-1 shows the sprung mass of the vehicle without the track; the mass is represented by M_S. Figure 8-2 shows the portion of the track which is considered suspended. As shown, the suspended length of the track is l_{t2}. If the mass per unit length is expressed as M_{T1}, the total sprung mass M_B of a track-laying vehicle can be expressed as

$$M_B = M_S + 2M_{T1}\, l_{t2} \qquad (8\text{-}1)$$

Figure 8-3 illustrates the transfer of the center of gravity effected by the inclusion of the sus-

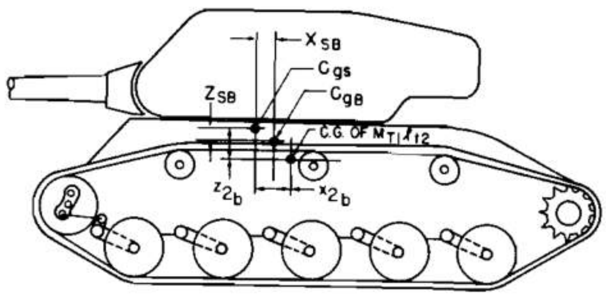


Figure 8-3. Center of Gravity Positions

pended portions of the tracks. The longitudinal and vertical distances of the transfer of centers of gravity, C_{gs} to C_{gB}, are represented by X_{SB} and Z_{SB}, respectively. The values of X_{SB} and Z_{SB} are obtained by taking moments of the track mass $M_{T1}l_{t2}$ on each side of the vehicle about C_{gs}, the C.G. of M_S, and dividing by the combined mass M_B as follows,

$$X_{SB} = \frac{2M_{T1}\,l_{t2}\,x_{2b}}{M_B} \tag{8-2}$$

$$Z_{SB} = \frac{2M_{T1}\,l_{t2}\,z_{2b}}{M_B} \tag{8-3}$$

where x_{2b} and z_{2b} are, respectively, the longitudinal and vertical distance from the center of gravity of the sprung mass M_S to the center of gravity of the suspended portions of the track.

The mass moments of inertia about lateral axes through their respective centers of gravity (slug-in.[2]) are represented by the following terms:

I_T = moment of inertia of the suspended portion of the track on each side of the vehicle about the respective center of gravity of the suspended portion

I_S = moment of inertia of the sprung mass excluding the track about the center of gravity of the sprung mass

I_{Bg} = moment of inertia of the sprung mass including the suspended portions of the track about C_{gB}

The moment of inertia I_{Bg} about the center of gravity C_{gB} of M_B is the summation of I_S, $2I_T$, and the respective products of the masses and the second power of the center of gravity transfer distances. This can be expressed as

$$I_{Bg} = I_S + M_S\left[(X_{SB})^2 + (Z_{SB})^2\right] + 2I_T + 2M_{T1}\,l_{t2}\left[(x_{2b} - X_{SB})^2 + (z_{2b} - Z_{SB})^2\right] \tag{8-4}$$

Equation 8-4 expresses the properties of the sprung mass relative to the center of gravity, mass, and moment of inertia.

8-13 EFFECTIVE MOMENT OF INERTIA OF THE SPRUNG MASS

Angular accelerations of the sprung mass cause considerable longitudinal acceleration of the road wheels and represent a significant inertial factor which is superimposed on the inertial behavior of the basic mass system. An analytical approach to the problem of determining the location of the pitching axis of a sprung vehicle having a trailing link suspension system is presented in Ref. 2. This analysis considers the inertial influence of the road wheels as well as that of the basic mass system. The inertial influences of the road wheels cause the pitching axis to move downward from the center of gravity location as shown by the distance Z_{gBM} in Figure 8-4. The point C_{BM} on the oscillation plane, directly below C_{gB}, is the equivalent center of inertia and, hence, the center of pitching of the total sprung mass. The determination of the magnitude of the distance Z_{gBM} is based on equilibrium of longitudinal inertia reactions.

By reference to Figure 8-4, the influence of the road wheels can be derived. Considering a clockwise angular displacement $d\Theta$ about C_{BM}, the longitudinal displacement (in inches) of the center of gravity of the total sprung mass M_B is $Z_{gBM}d\Theta$, and the longitudinal acceleration (in ft/sec[2]) is

$$\frac{Z_{gBM}}{12}\;\frac{d^2\Theta}{dt^2}$$

The longitudinal inertia reaction force (lb) of the sprung mass is

$$R_1 = M_B\;\frac{Z_{gBM}}{12}\;\frac{d^2\Theta}{dt^2} \tag{8-5}$$

where M_B is in slugs (lb-sec^2/ft)

At the trailing link pivot, the longitudinal move-

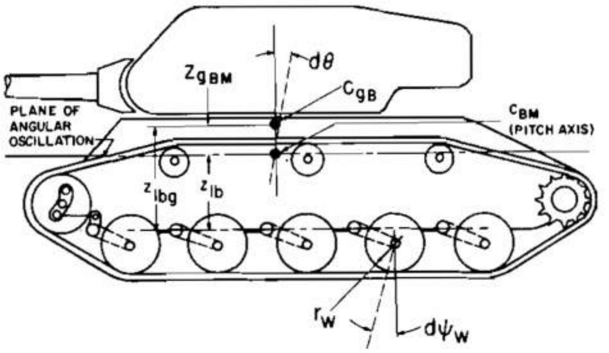


Figure 8-4. Road Wheel Inertial Effects

ment (in inches) is $z_{1b} d\Theta$ or $(z_{1bg} - Z_{gBM})\ d\Theta$. The longitudinal acceleration (ft/sec²) is

$$\left(\frac{z_{1bg} - Z_{gBM}}{12}\right) \frac{d^2\Theta}{dt^2}.$$

The inertia reaction R_i (lb) of the road wheels resisting that longitudinal acceleration can be expressed as

$$R_i = n_w M_w \left(\frac{z_{1bg} - Z_{gBM}}{12}\right) \frac{d^2\Theta}{dt^2} \qquad (8\text{-}6)$$

where

M_w = mass of road wheel including the spindle and associated mounting parts, slugs

n_w = number of road wheels on each side of the vehicle

The longitudinal movement and acceleration at the road wheel spindles are assumed equal to those at the trailing link pivot.

The relationship between the angular displacement of the road wheels and the longitudinal displacement of the spindles is

$$r_w d\Psi_w = (z_{1bg} - Z_{gBM})\ d\Theta \qquad (8\text{-}7)$$

where

r_w = rolling radius of the road wheels, in.

$d\Psi_w$ = angular displacement of a road wheel for a particular longitudinal displacement of a wheel spindle, rad

From which the angular acceleration of the road wheels in rad/sec² is

$$\frac{d^2\Psi w}{d^2 t} = \left(\frac{z_{1bg} - Z_{gBM}}{r_w}\right)\frac{d^2\Theta}{dt^2} \qquad (8\text{-}8)$$

The reaction moment, ft-lb, of each road wheel resisting this angular acceleration is

$$\frac{I_w}{(12)^2}\frac{d^2\Psi w}{dt^2} = \frac{I_w}{(12)^2}\left(\frac{z_{1bg} - Z_{gBM}}{V_w}\right)\frac{d^2\Theta}{dt^2} \qquad (8\text{-}9)$$

where

I_w = moment of inertia of each road wheel about the lateral z axis through its c.g., slug-in.²

The reaction force L_i on one side of the vehicle (due to the angular motions of the wheels) acting at the wheel spindles, and transmitted to the trailing link pivot on one side of the vehicle can be derived by dividing Equation 8-9 by $r_w/12$ and multiplying by the number of road wheels, n_w, per side. Therefore, Equation 8-9 becomes

$$L_i = \frac{N_w I_w}{12}\left[\frac{Z_{1bg} - Z_{gBM}}{(r_w)^2}\right]\frac{d^2\Theta}{dt^2},\ lb \qquad (8\text{-}10)$$

The total longitudinal inertia reactions exerted at the arm hinge axes by the road wheels on each side of the vehicle can then be expressed as

$$R_i + L_i = -\frac{n_w}{12}\left[M_w + \frac{I_w}{(r_w)^2}\right](z_{1bg} - Z_{gBM})\frac{d^2\Theta}{dt^2},\ lb \qquad (8\text{-}11)$$

The negative sign has been added to indicate that the reaction forces are toward the front of the vehicle when the front of the vehicle moves downward in pitch.

The algebraic summation of the longitudinal inertia reactions must be equal to zero. This is accomplished by summing Equations 8-5 and 8-11 (after multiplying the latter by 2) and setting them equal to zero. The resulting expression can be solved for Z_{gBM} in inches, thus

$$Z_{gBM} = \frac{2n_w\left[M_w + \frac{I_w}{(r_w)^2}\right]}{M_B + 2n_w\left[M_w + \frac{I_w}{(r_w)^2}\right]} z_{1bg} \qquad (8\text{-}12)$$

As shown in Figure 8-4, Z_{gBM} is the distance to the reference point C_{BM} of the basic mass system below the center of gravity C_{gB} of the total sprung mass M_B. The position of Z_{gBM} is of interest since it is one method of locating the horizontal plane containing the axes of two primary vibrations of the vehicle. (Another, more commonly used, analysis to determine the axes of oscillation of a sprung mass is presented in Section VI.)

The effective moment of inertia about the reference point C_{BM} of the sprung mass system is the summation, about the lateral axis passing through point C_{BM}, of the moment of inertia of the sprung mass and the equivalent moment of inertia of the longitudinal inertia reactions exerted at the trailing link pivots by the road wheels. This effective moment of inertia, I_B (slug-in.^{2}) is expressed as

$$I_B = I_{Bg} + M_B (Z_{gBM})^2 + 2n_w \left[M_w + \frac{I_w}{(r_w)^2} \right] (z_{1bg} - Z_{gBM})^2 \tag{8-13}$$

Equation 8-13 does not take into consideration the influences of vertical inertia reactions of the road wheels during vertical accelerations of the sprung mass. A comprehensive treatment is given in Ref. 2.

SECTION IV INERTIAL PROPERTIES OF THE BASIC MASS SYSTEM (Refs. 4, 5)

8-14 INTRODUCTORY DISCUSSION

The basic mass system, in the present context, is the total sprung mass of a vehicle. As stated in the previous section, a portion of the suspension system of a vehicle may be a part of the sprung mass. For example, in the case of a tracked vehicle, the basic mass system includes the suspended portion of the track.

The inertial properties of the basic mass system that are significant in dynamic and vibration analysis are the mass, the position of the center of mass (center of gravity), the moment of inertia, the products of inertia, and the location of the principal inertial axes.

8-14 MASS

8-14.1 COMPUTATION OF MASS

The mass m of a body can be computed by integrating the product of mass density ρ and elemental volume dV over the body. This relationship can be expressed as

$$m = \int_v \rho \, dV \tag{8-14}$$

In a body composed of a number of elements, each having an average density, the mass m can be expressed as

$$m = \sum_{i=1}^{n} \rho_i V_i \tag{8-15}$$

8-14.2 EXPERIMENTAL DETERMINATION OF MASS

The mass of a body may be determined directly by applying Newton's second law of motion. A more convenient method consists of weighing the body and calculating the mass using the basic relationship

$$m = \frac{g}{W} \tag{8-16}$$

where

m = mass of the body, lb-sec^2/ft (slug)
W = weight of the body, lb
g = acceleration of gravity at the particular location of the body, ft/sec^2

8-15 CENTER OF MASS

8-15.1 COMPUTATION OF CENTER OF MASS

The center of mass (center of gravity) in a cartesian coordinate system X, Y, Z is located at the point X_c, Y_c, Z_c with

$$X_c = \frac{1}{m} \int_v X_{(v)} \, \rho \, dV \tag{8-17}$$

$$Y_c = \frac{1}{m} \int_v Y_{(v)} \, \rho \, dV \tag{8-18}$$

$$Z_c = \frac{1}{m} \int_v Z_{(v)} \, \rho \, dV \tag{8-19}$$

where

$X_{(v)}$, $Y_{(v)}$, $Z_{(v)}$ = coordinates of the element of volume dV

m = mass of the body, lb-sec²/ft

If the total mass can be divided into elements whose centers of mass are known, the center of mass of the entire body having a mass m can be located by equations as follows.

$$X_c = \frac{1}{m}\sum_{i=1}^{n} X_{ci}\, m_i \tag{8-20}$$

$$Y_c = \frac{1}{m}\sum_{i=1}^{n} Y_{ci}\, m_i \tag{8-21}$$

$$Z_c = \frac{1}{m}\sum_{i=1}^{n} Z_{ci}\, m_i \tag{8-22}$$

where

$X_{ci}, Y_{ci}\, Z_{ci} = X$, Y, or Z coordinates of the center of mass of element m_i.

8-15.2 EXPERIMENTAL DETERMINATION OF CENTER OF MASS

Usually, the center of mass is measured indirectly by locating the center of gravity of the body. The center of gravity of a body may be determined experimentally by various methods.

One method, which is suitable for relatively large and heavy bodies such as vehicles, utilizes a "teeter-table" device. The teeter-table device is, basically, a large rigid platform that is supported transversely on knife-edge bearings. The transverse bearing axis is located below the platform center of gravity. Under no-load conditions, the platform is held in a horizontal position by a suitable elastic member. The system is then calibrated by applying moments about the teeter axis and measuring corresponding angular displacements. The result is a load versus angular deflection curve similar to the spring rate curve of a linear spring. The center of gravity of a vehicle or component is determined experimentally by the procedure which follows.

First, the vehicle or component is weighed. Then it is placed on the platform of the teeter-table and is adjusted longitudinally so that the platform is statically level. The center of gravity of the vehicle or component is located in the vertical plane passing through the knife edges.

The next step is the determination of the load versus angular deflection rate of the combined platform and vehicle system about the knife-edge axis. The angular rate of the platform and mass is determined by again measuring the angular deflections for given loadings. This angular rate will be less than the calibrated angular rate of the apparatus, owing to the moment produced by displacing the center of gravity from the plane of the knife edge. The location of the center of gravity above the knife edge axis can be determined by considering the moments acting about that axis during an angular displacement.

The location of the center of gravity along the transverse axis can be determined by rotating the vehicle 90 deg on the platform and repeating the first step.

8-16 COMPUTATION OF MOMENTS OF INERTIA

Analyses of the motions of bodies frequently consider the body as a system of connected particles and mathematical expressions frequently arise in the analyses that involve the masses of the individual particles and the squares of their distances from a line or plane of reference. The product of these two factors is called the *second moment of mass* of the particle, the *mass moment of inertia* of the particle, or just simply the *moment of inertia* of the particle with respect to the line or plane. The moment of inertia of the system of particles (mass system or body) with respect to a line or plane is the sum of the moments of inertia of the particles with respect to the given line or plane. Thus, if the particle masses of a system are designated as m_1, m_2, m_3. . . , and their distances from the reference axis as r_1, r_2, r_3. . . , the moment of inertia of the system is expressed as

$$I = m_1r_1^2 + m_2r_2^2 + m_3r_3^2 \ldots m_nr_n^2$$

$$I = \sum_{i=1}^{n} m_ir_i^2 \tag{8-23}$$

If the mass system constitutes a continuous body, the preceding summation can be written as

$$I = \int r^2dm \tag{8-24}$$

where dm is an element of the mass of the body and r is the distance of the element from the reference line or plane. The limits of the integral must be so chosen that every element of mass of the entire body is included in the integration.

It is often convenient to express the moment of inertia in terms of the total mass m of the body and the square of a distance k, known as the radius of gyration or

$$I = mk^2 \tag{8-25}$$

Thus, the radius of gyration k of a body is the distance from a reference axis at which all of the mass may be considered to be concentrated and exhibit the same moment of inertia with respect to that axis as does the actual, distributed mass.

Moments of inertia may be referenced to any axis that is advantageous to the analysis. If the moment of inertia of a body with respect to an axis passing through its mass-center (centroidal axis) is known, the moment of inertia with respect to any parallel axis may be found, without recourse to integration, by applying the *parallel axis theorem*. This states that the moment of inertia of a body with respect to any axis is equal to the moment of inertia of the body with respect to a parallel axis through its mass-center plus the product of the mass of the body and the square of the distance between the two axes, or stated mathematically

$$I = \overline{I} + md^2 \tag{8-26}$$

where $\overline{I}$ is the moment of inertia with respect to an axis through the mass center, and I is the moment of inertia about a parallel axis that is at a distance d from the mass center.

The moment of inertia of a body with respect to the line of intersection of two perpendicular planes is the sum of the moments of inertia with respect to the two planes. Or, if I_{xy} and I_{xz} denote the moments of inertia of a body with respect to the xy and xz planes, respectively, the moment of inertia I_x about the x-axis (the intersection of the xy and xz planes) will be

$$I_x = I_{xy} + I_{xz} = \int (y^2 + z^2)\ dm \tag{8-27}$$

where y and z are the distances along the respective axes to the element of mass dm.

Although Equation 8-24 expresses the fundamental definition of the moment of inertia, it can only be applied to continuous geometric bodies whose mass distribution with respect to the reference axis can be described by mathematical equations (definite integrals). Unfortunately, military vehicles and their major components—such as turret assemblies, power plants, transmissions, etc.—are not such bodies. Instead, they are composites of highly complex shapes, totally incompatible to the application of Equation 8-24 or the right hand version of 8-27. Bodies of such complex nature are considered as made up of many finite parts, the moments of inertia of which are known or readily obtainable. The procedure, then, is to translate the moments of inertia of all of the parts to a common reference axis, through the application of Equations 8-26 or 8-27 (whichever is applicable), and then total the moments of inertia of the several parts. The common reference axes usually chosen are those passing through the mass center (center of gravity) of the vehicle.

8-17 EXPERIMENTAL DETERMINATION OF MOMENTS OF INERTIA

Moments of inertia for bodies that have highly irregular shapes and a nonuniform distribution of mass, such as automotive vehicles and their major components, cannot be found by integration; and the method given in paragraph 8-16 of determining and summing-up the moments of inertia of the constituent parts of the composite is extremely laborious and time consuming. The moments of inertia of such complex bodies, however, can be determined experimentally by methods which make use of the laws of motion of a pendulum. Perhaps the simplest, though not the most accurate, method is to suspend the vehicle by means of a suitable cable or chain and, allowing it to oscillate as a compound pendulum, note its frequency. The moment of inertia (ft-lb-min^2) about the center of oscillation can then be calculated from the equation

$$I_o = \frac{W\overline{r}}{4\pi^2 f^2} \tag{8-28}$$

where

W = weight of the vehicle or component, lb

$\bar{r}$ = distance from the center of gravity of the vehicle to the axis of rotation, ft

f = natural frequency of the compound pendulum, cpm

The center of gravity can be located by balancing the vehicle on knife edges as described in paragraph 8-15.2.

The pendulum method just described is relatively simple but is not as accurate as other methods when applied to vehicles. Most vehicles have some form of suspension system which results in both sprung and unsprung masses. When resting on the ground in a normal operating posture, the sprung mass compresses the elastic members and reduces the distance between the two masses. When suspended, however, the unsprung mass stretches the elastic members, thus increasing the distance between the two masses. These differences in weight distribution and geometry have an appreciable affect upon both the location of the center of gravity and upon the magnitude of the moment of inertia.

A more accurate method of determining the mass moment of inertia makes use of the teeter-table device described in paragraph 8-15.2. The vehicle is placed upon the platform of the teeter-table with its center of gravity in the vertical plane of the knife-edge bearing. When the equilibrium of this system about the teeter axis is disturbed, the platform (with vehicle) will oscillate about the teeter axis at the natural frequency f of the system. This natural frequency is measured and recorded for use in the subsequent moment of inertia computations. In addition, the angular spring rate K of the system is determined by plotting applied moments against the angular deflections they produce. The mass moment of inertia of the system (ft-lb-min^2) about the oscillation axis can then be calculated from the following equation

$$I_o = \frac{K}{4\pi^2 f^2} \tag{8-29}$$

where

K = angular (torsional) spring rate of the system, ft-lb rad

f = natural frequency of the system, cpm

It should be noted that I_o in Equation 8-29 is the mass moment of inertia of the *vehicle and platform assembly* about the oscillation axis. To determine the moment of inertia of the vehicle only, it is necessary to determine the moment of inertia of the platform (by repeating the procedure using the platform only) and subtracting this value from that calculated for the vehicle and platform together.

Equations 8-28 and 8-29 give the mass moments of inertia about the center of oscillation. The moment of inertia about the centroidal axis can be found by applying Equation 8-26.

SECTION V CHARACTERISTICS OF THE ELASTIC SUPPORT SYSTEM (Ref. 6)

8-18 VERTICAL SPRING RATE OF ELASTIC SUPPORT SYSTEM

The vertical rate of the basic elastic system of a vehicle is the summation of the vertical rates of the individual suspension units. Figure 8-5 shows a military vehicle and a diagrammatical representation of the suspension system.

The spring rate of the elastic support system is the summation of the individual vertical rates of the suspension units; the equation is

$$\Lambda_B = \sum_{i=1}^{n} \lambda_i \tag{8-30}$$

where

Λ_B = vertical spring rate for the entire vehicle

λ_i = vertical spring rates of the individual suspension units on both sides of the vehicle

n = total of suspension units

The elastic rate of the individual suspension unit is significantly influenced by the tire rate in addition to the principal elastic element (spring). Figure 8-6 shows a schematic representation of the compound spring system of a suspension unit. The principal elastic member, e.g., torsion bar, is in

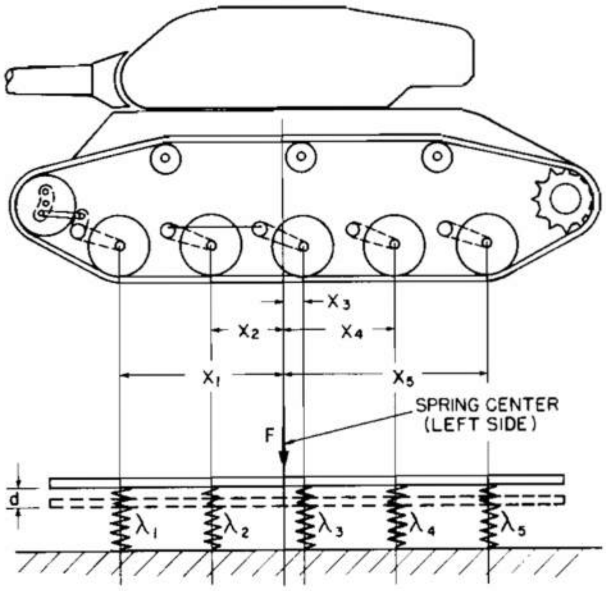


Figure 8-5. Diagrammatical Representation of Suspension System

parallel with the secondary elastic element, e.g., road wheel tire.

The mass of the wheel also influences the total elastic rate of the suspension unit. As shown in Ref. 6, the effective vertical rate of the suspended mass relative to the terrain (considering tire rate and wheel mass as well as the principal spring rate) can be expressed as

$$\lambda_f = \frac{\lambda_s [\lambda_T - M_w (\omega_1)^2]}{\lambda_s + [\lambda_T - M_w (\omega_1)^2]} \tag{8-31}$$

where

λ_f = effective vertical spring rate of the combined system, lb/ft

λ_s = vertical spring rate of the principal spring, lb/ft

λ_T = vertical spring rate of the road wheel tire (may include elasticity of the track), lb/ft

M_W = mass of each road wheel, slug

ω_1 = frequency of the sprung mass, rad/sec

Reference 6 also demonstrates that, for current suspension designs, the dynamic rate influence of the wheel mass is relatively insignificant. Therefore, a simplified expression for the effective spring rate (lb/ft) of the suspension unit can be written as λ_e, where

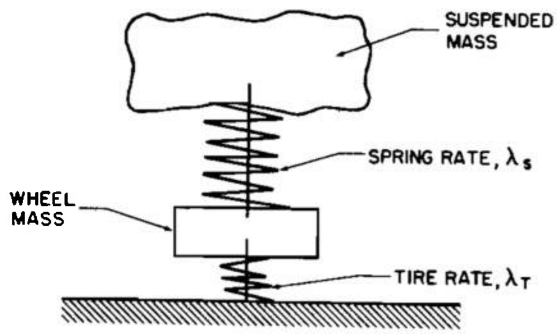


Figure 8-6. Compound Spring System

$$\lambda_e = \frac{\lambda_s \lambda_T}{\lambda_s + \lambda_T} \tag{8-32}$$

All further discussion of the basic elastic system will utilize this simplified form.

8-19 SPRING CENTER

The spring center of the basic elastic system is the longitudinal (fore and aft) location on the vehicle where the application of an additional force will cause only vertical deflection of the vehicle, that is, the additional force will cause equal vertical deflection at all of the suspension units.

The location of the spring center can be de-

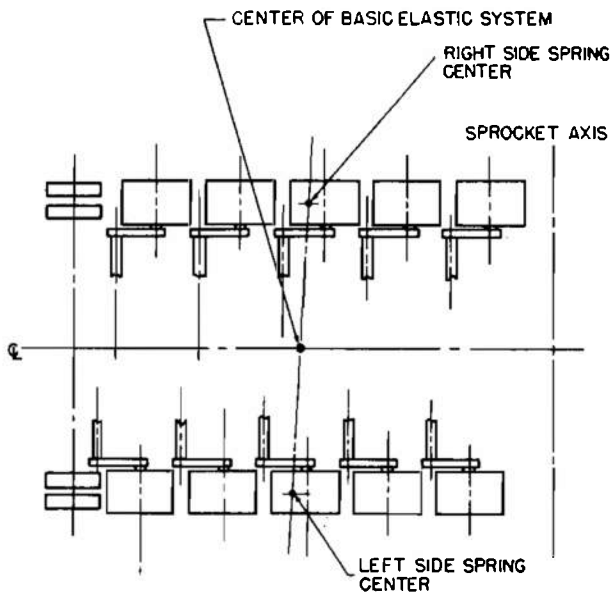


Figure 8-7. Location of Center of Basic Elastic System

termined mathematically by using the principle that the algebraic summation of the rate moments of the individual suspension units about the spring center is equal to zero, each rate moment being the product of the particular vertical spring rate λ and its respective longitudinal distance from the spring center x.

For tracked vehicles having transverse torsion bar springs, it is customary to longitudinally displace the suspension unit on one side of the vehicle relative to those on the other side.

Figure 8-7 shows the offset of typical suspension units for a track-laying vehicle. The spring center on one side of the vehicle is longitudinally offset from that on the opposite side. The longitudinal location of the spring center of the vehicle can be found by passing an axis through the individual centers; the point of intersection of this axis with the longitudinal centerline of the vehicle is the spring center of the basic elastic system.

SECTION VI PRIMARY VIBRATIONS IN THE LONGITUDINAL VERTICAL PLANE (Ref. 7)

The location of the lateral axis of oscillation for a body on resilient supports, e.g., a sprung vehicle, is determined by the combined elastic and inertial behavior of the particular system.

The general procedure for determining the location of the lateral axis of oscillation can be explicated by considering the elastic and the inertial influences separately, and then the combined effect. Figure 8-5 is a diagrammatic representation of a typical vehicular suspension system. The analysis will be simplified if the multiple spring system is replaced by an elastically equivalent two spring system. The two spring system is shown in the upper part of Figure 8-8 and is generated by considering the spring rate moments about the spring center C of the lower system.

8-20 ELASTIC SYSTEM

Points such as A and B are called *elastically conjugate* points. These points are related such that if a vertical force is applied at point A, the body (represented by horizontal axis Y-Y) will rotate about the point B and vice versa. Any pair of points that possess this reciprocal relationship satisfy the equation $pq = ab$ where a, b, p, and q are distances shown in Figure 8-8. Since the dimension q (or p) can be chosen arbitrarily, it follows that there is an infinite number of such pairs of points.

The geometrical construction shown in Figure 8-8 enables the pairs of points to be determined graphically. The hyperbola $pq = ab = c^2$ is drawn relative to the axes XCY. The point B_1 corresponding to B on the hyperbola is projected horizontally to meet the line CZ at A_1. The projection of A_1 on the Y axis gives the point A. It is obvious that as point A approaches point C, point B will approach infinity. Similarly, as point B approaches point C, point A approaches infinity.

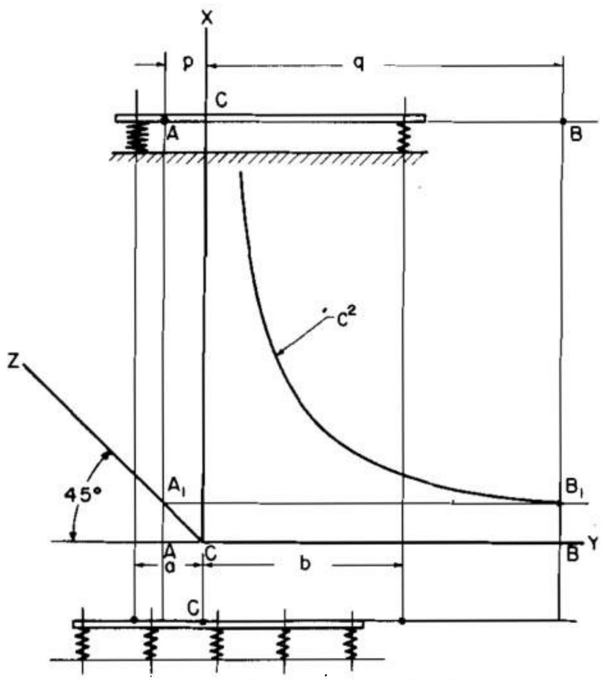


Figure 8-8. Elastically Conjugate Points

8-21 INERTIAL SYSTEM

A similar analysis can be made with respect to *dynamically conjugate* points. Figure 8-9 shows a

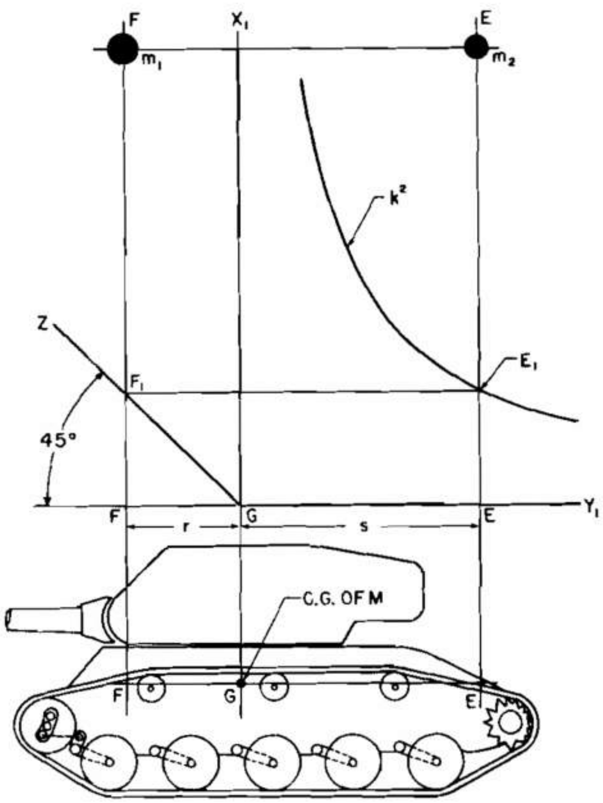


Figure 8-9. Dynamically Conjugate Points

typical vehicular mass system M. A dynamically equivalent mass system can be selected to replace the original system. This second system must possess the same mass, moment of inertia, and position of its mass center as the original mass system. Thus a body consisting of two masses m_1 and m_2 (with $m_1 + m_2 = M$) concentrated at points E and F as shown and connected by a weightless bar can be substituted for the original mass provided that the essential dynamical characteristics are the same.

The moment of inertia about its mass center of the original mass can be expressed as Mk^2. If $m_1 r = m_2 s$ and $m_1 r^2 + m_2 s^2 = Mk^2$, then $rs = k^2$.

Since the dimension r (or s) can be chosen arbitrarily, it follows that there is an infinite number of such pairs of points. The geometrical construction shown in Figure 8-9 enables the pairs of dynamically conjugate points to be determined graphically in a manner similar to that used for the elastically conjugate points.

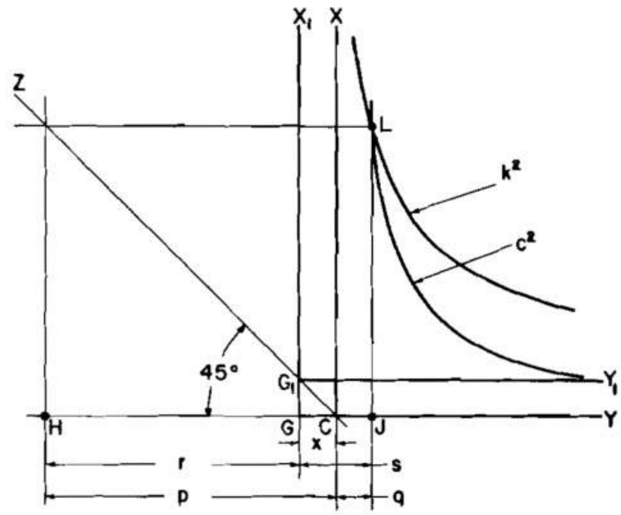


Figure 8-10. Doubly Conjugate Points

8-22 COMBINED SYSTEM

The elastically conjugate system can be combined with the dynamically conjugate system as shown in Figure 8-10. For a given mass-elastic system, the two hyperbolas will, in general, intersect at one point L. Then the points H and J, which are projected from L, are points which are both elastically and dynamically conjugate and are termed *doubly conjugate* points.

If the position of the mass center G, in Figure 8-10, approaches the spring center C, then the point of intersection of the hyperbolas will move to the right; and when G arrives at C, the point J will be at infinity and the point H will coincide with the common point CG. Under these conditions, and if c^2 does not equal k^2, the suspended body will oscillate with two independent motions, i.e., translation in the vertical plane, and angular displacement about the point G. The system now acts as if the entire mass is concentrated at H but possesses a moment of inertia about H equal to Mk^2.

An algebraic method to determine the doubly conjugate points can be presented. By referring to Figures 8-8 and 8-9, the following equations can be written

$$pq = ab = c^2 \qquad (8\text{-}33)$$

$$rs = k^2 \qquad (8\text{-}34)$$

$$p = r + x \tag{8-35}$$

$$q = s - x \tag{8-36}$$

For a given system the quantities c^2, k^2, and x are normally known; and p, q, r and s are to be determined. Substituting for p and q in Equation 8-33 results in

$$(r + x)\left(\frac{k^2}{r} - x\right) = c^2 \tag{8-37}$$

or

$$r^2 x + r(x^2 + c^2 - k^2) - k^2 x = 0 \tag{8-38}$$

Solving for r

$$r = \frac{(k^2 - x^2 - c^2)}{2x} \pm \frac{[(k^2 - x^2 - c^2)^2 + 4k^2 x^2]^{\frac{1}{2}}}{2x} \tag{8-39}$$

Given r and Equations 8-34, 8-35, and 8-36 then determine s, p, and q. Equation 8-39 gives two values for r and, hence, two values for s. However, it will be found that the negative value of r equals the positive value of s and vice versa, so that only two points, H and J, are determined.

A special case exists when points C and G coincide and $c^2 = k^2$. Under these conditions, the two hyperbolas coincide, and there is an infinite number of pairs of double conjugate points related to the system since any point on the common hyperbola gives such a pair.

It has been stated that forces applied to one of a pair of elastically conjugate points produce no motion at the other point; therefore, inertia forces developed at one of a pair of double conjugate points will produce no motion at the other point. Thus vibrations of masses located at the double conjugate points are independent and no coupling exists between them even though the mass center does not coincide with the spring center. These independent motions will be angular oscillations about fixed points.

If the system is deflected by a force which acts at a point other than a double conjugate point, then both reference masses will be deflected. When the force is removed, the motion of each double conjugate point will be a simple harmonic motion while the motion of every other point will be compounded of the two simple motions.

The oscillating motions about the double conjugate points (or lateral axes) are the primary vibrations of the suspended mass. The frequencies of the primary vibrations can be calculated by considering the relationship of the angular rate and the moment of inertia about the particular axis.

8-23 NATURAL FREQUENCY

The natural frequency f_n of the primary vibrations about an oscillation axis, in cpm, can be expressed as

$$f_n = \frac{60}{2\pi}\left[\frac{K_t}{J}\right]^{\frac{1}{2}} \tag{8-40}$$

where

J = mass moment of inertia of the basic sprung mass about the selected axis, in.-lb-sec^2

K_t = angular spring rate of the sprung mass about the same axis as J, in.-lb/rad

Each primary vibration is characteristically a simple harmonic motion. Usually the vibrations about two separate double conjugate points have different frequencies and, as a result, there is a continuous change of phase relationship. When the two vibrations exist simultaneously, the motion at any location remote from axes P and Q tends to be irregular and spasmodic.

The foregoing discussion indicates that fire control for a given vehicle would be aided by arranging the oscillation axes so that one of them is under the gun mounting. In addition, the discussion indicates that the proposed ideal system of having the elastic system center coincident with the mass system center, and the mass constant k^2 equal to the elastic constant c^2, would possess the advantage of simplifying the vibrational behavior of the vehicle resulting from a terrain disturbance.

On the basis of quite limited information, current combat vehicles appear to have vertical displacement frequencies of the order of 110 to 120 cpm while the pitch frequencies are considerably lower. An increase in the angular spring rate of the elastic support system of these vehicles would increase the energy absorption capacity of the suspension systems. It is anticipated that such an increase in angular spring rate would improve stability and maneuverability, and permit higher speed over rough terrain with slight, if any, sacrifice of personnel comfort.

SECTION VII LATERAL VIBRATIONS (Ref. 8)

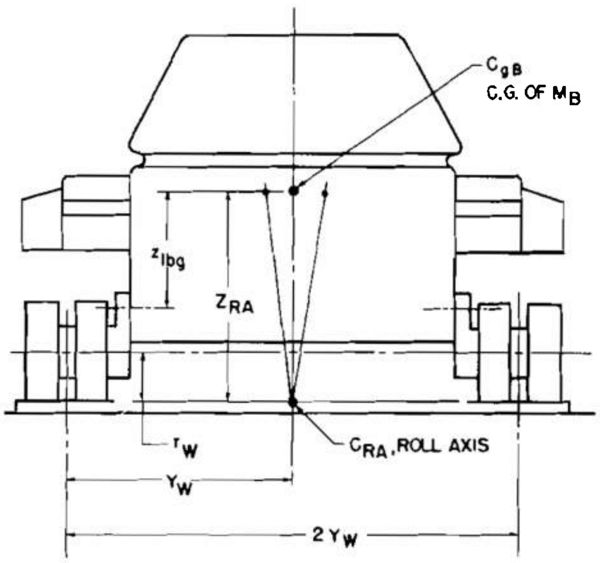


Figure 8-11. Roll Axis of Track-laying Vehicle

The lateral vibratory characteristics of a vehicle about a longitudinal axis are called its roll properties. Generally, the mass components are located so that the center of gravity is approximately coincident with the geometric center of the vehicle. In addition, the suspension components of the opposite sides of the vehicle are usually alike and equidistant from the vehicle's geometric center, thus, making the spring center coincident with the geometric center. Accordingly, the roll axis passes longitudinally through the geometric center of the vehicle.

For an unconventional vehicle having the lateral center of gravity considerably offset sideways from the spring center, the lateral vibration characteristics may be determined by a procedure comparable to that presented for the longitudinal vibrations.

Figure 8-11 is a schematic transverse section of a typical track-laying vehicle. The center of gravity of the sprung mass is located the distance Z_{RA} about the position C_{RA} which designates the roll axis. As shown, the horizontal plane containing the roll axis is assumed to be that which is flush with the upper surface of the track lying on the terrain. This position is determined by the geometry of the suspension linkage. A detailed method of determining the position of the roll axis for various suspension designs is presented later.

8-24 LATERAL PROPERTIES OF THE BASIC MASS SYSTEM ABOUT THE ROLL AXIS

The effective angular spring rate of the basic elástic system for angular displacement about the roll axis can be expressed as

$$\Gamma = \Lambda_B (Y_W)^2 + 2n_W \lambda_t (y_t)^2 - W_B Z_{RA} \qquad (8\text{-}41)$$

where

Γ = effective angular spring rate of the basic elastic system, in.-lb/rad

Λ_B = vertical spring rate of the basic elastic system, lb/in.

Y_W = transverse distance from the geometric center of the vehicle to the lateral center of the road wheels on each side of the vehicle, in.

n_W = number of road wheels on each side of the vehicle

λ_t = vertical spring rate of tires of one wheel resting on the track, lb/in.

y_t = axial offset of each tread centerline from the wheel lateral center, in.

W_B = weight of the sprung mass, lb

Z_{RA} = vertical distance of the longitudinal roll axis below the center of gravity of the sprung mass, in.

8-25 FREQUENCY OF THE LATERAL VIBRATION ABOUT THE ROLL AXIS

For the symmetrical vehicle, the natural frequency f_n of the primary vibrations about the roll axis, in cpm, can be expressed as

$$f_n = \frac{60}{2\pi}\left[\frac{\Gamma}{J}\right]^{\frac{1}{2}} \tag{8-42}$$

where

$J =$ mass moment of inertia of the basic sprung mass about the roll axis, in.-lb-sec^2

$\Gamma =$ angular spring rate of the basic elastic system about the same axis as J, in.-lb/rad

For the lateral motion of the sprung mass, two primary vibrations can be described: vertical displacement and roll. The former vibration resolves into the two longitudinal vibrations previously discussed. Therefore, the primary lateral motion is roll. However, there are certain yawing tendencies associated with roll, the trends of which are influenced by the longitudinal location of the center of gravity relative to the longitudinal location of the effective center of the terrain reactions.

The roll-producing couple results from forces acting on the sprung mass. The magnitude of this couple is given by the product of the disturbing force and the distance between its point of application and the roll axis of the sprung mass. These forces can be classified as (a) inertial forces, (b) wind forces, and (c) terrain induced forces.

The inertial forces are centrifugal forces developed when the vehicle moves in curvilinear motion. The wind forces, which can be considered as acting at the center of pressure of the vehicle, produce roll since the center of pressure is normally above the roll axis. The terrain induced roll producing forces result when one side of a vehicle contacts a terrain irregularity or when the vehicle stands or operates on side slopes.

The geometry of the vehicle suspension system is the major determining factor with respect to the location of the roll axis for a given vehicle. This subject and related topics such as roll angles, roll moments, and antiroll devices are discussed in detail in Ref. 9.

SECTION VIII EFFECT OF OPERATIONAL REQUIREMENTS ON LOCATION OF VEHICLE CENTER OF GRAVITY (Ref. 10)

The relative locations of the mass system center and the elastic system center of a vehicle influence the performance and operational behavior of the vehicle. These cannot, however, be located arbitrarily to satisfy ride and mobility characteristics. The center of gravity is normally located so that traction, trench crossing, and obstacle climbing are maximized.

The location of the center of gravity which best satisfies those mobility requirements is one that is vertically as low as possible, laterally on the vehicle centerline, and longitudinally on or slightly forward of the center of the road wheelbase. If the vehicle is to have equal uphill and downhill traction characteristics and equal trench-crossing and obstacle-climbing characteristics in forward and reverse, the longitudinal location of the center of gravity should be on the center of the road wheelbase. If, however, the emphasis is to be on uphill and forward operation over obstacles and trenches, the center of gravity should be forward of the center of the road wheelbase.

8-26 LEVEL GROUND

On level ground, the centrally located center of gravity will result in a uniform loading of the road wheels under static conditions. However, when a track-laying vehicle is driven at a constant speed, there is an apparent weight shift in the longitudinal direction opposite to the direction of motion owing to the final drive torque reaction. Since forward motion of the vehicle predominates, a slightly forward center of gravity will compensate for the apparent weight shift and help maintain uniform loading of the road wheels and, therefore, maximum traction capabilities.

8-27 SLOPE OPERATIONS

When a vehicle is climbing a slope, there is a redistribution of the ground contact pressure resulting from three conditions (a) the actual horizontal movement of the vehicle center of gravity in relation to the track caused by the inclined position, (b) the actual horizontal movement of the center of gravity of the sprung mass in relation to the unsprung mass caused by the inclined position, and (c) the apparent weight shift caused by the propulsion torque reaction.

All three of these factors contribute to a weight shift to the rear and, as such, a forward center of gravity would tend to maintain a uniform load distribution along the track. Downhill motion also causes a redistribution of the ground pressure. The weight shift is in the direction of motion except for that caused by the driving torque reactions. The lower the center of gravity, for a given vehicle, the less will be the effect of the incline with respect to redistribution of the road wheel loading.

Side slope operation causes an increase in the loading on the lowest track (or wheels) and a corresponding decrease in the loading on the upper level track (or wheels). Since the roll producing couples varies directly with the distance between the center of gravity and the roll axis, the shift in loading between the tracks on side slopes can be minimized by making this distance as small as practicable.

8-28 TRENCH CROSSING

The location of the center of gravity of a vehicle is an important factor with respect to the width of the trench the vehicle can cross without dropping into the trench. On the assumption that the trench has shear nonyielding walls and a particular vehicle has a centrally located center of gravity, the forward end of the vehicle will extend over the trench until the center of gravity passes the edge. Then the front portion of the track will drop to the opposite bank and the vehicle will proceed across the trench until the rear portion of the track leaves the initial bank. If the center of gravity at this time has passed the edge of the far wall, the vehicle will proceed across the trench. However, if the center of gravity is not over the edge of the far bank when the rear track leaves the bank, the rear end of the vehicle will drop into the trench. A slightly forward center of gravity will aid the vehicle in trench crossing since the above conditions are improved under these circumstances.

8-29 CLIMBING VERTICAL OBSTACLE

When a typical track-laying vehicle attempts to climb a shear nonyielding embankment, the limiting factor is the ability to put the center of gravity over the point of ground contact. If the center of gravity is too far rearward or the obstacle is too high for the particular vehicle, it may be impossible for the vehicle to place its center of gravity over the edge of the embankment and rock to the horizontal position required for a successful negotiation of the obstable. A forward location of the center of gravity may assist in this respect. However, the center of gravity must not be so far forward that the vehicle tips with too steep an angle coming down off the embankment.

8-30 SUMMARY

The previous statements regarding the effects of the location of the center of gravity on the behavior and performance of typical track-laying military vehicles have indicated that a slightly forward center of gravity location is advantageous. A rigorous analysis of the vehicular system and the forces acting on it during various modes of operation will show the best positions for the center of gravity with respect to operation requirements.

SECTION IX APPLICATION OF ANALOG COMPUTER TECHNIQUES TO SUSPENSION DESIGN (Refs. 11, 12, 13)*

8-31 INTRODUCTION

8-31.1 GENERAL DISCUSSION

The analysis and evaluation of a vehicle suspension by manual methods has always been a difficult, if not impossible, task for vehicles in the concept or design state. Fortunately, mathematical analysis and analog computers provide high speed tools for such studies. By use of these techniques, suspensions may be more thoroughly analyzed with greater accuracy for geometric or real-life terrain conditions. In addition, an analog computer simulation yields the dynamic loads experienced by components that cannot be developed by simple manual calculations. This technical note describes some current techniques employed by the Systems Simulation Branch of the U. S. Army Tank-Automotive Center to evaluate and analyze complete vehicle suspensions.

8-31.2 BASIC CONCEPTS

The discussion which follows is a brief outline of the fundamental elements of the electronic differential analyzer and the basic techniques of analog computer communication. For a more thorough treatment of the subject see Refs. 14, 15, and 17.

A *direct* type of analog computation can be accomplished by means of a physical model which is different from the physical system under study but whose parameters are analogous with a one-to-one correspondence. For example, as shown in Figure 8-12, a direct electrical analog can be described for a particular mechanical system. In this case, inductance is analogous to mass; elastance (the reciprocal of capacitance) to spring rate; voltage to force; current to velocity; and resistance to energy dissipation (damping). The similarity of the differential equations of each system is evident.

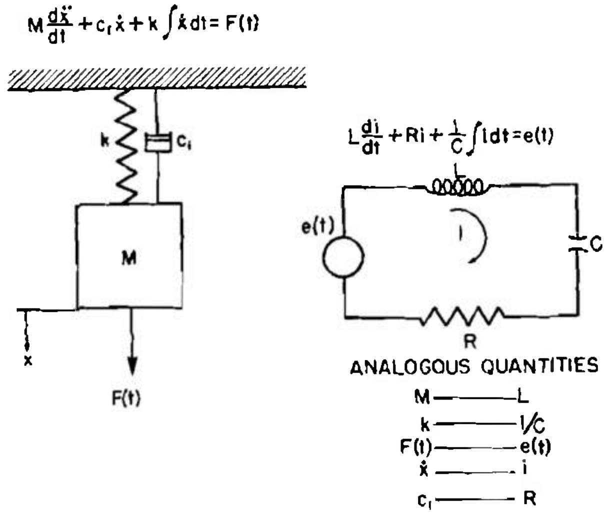


Figure 8-12. Electrical Analog of a Mechincal System

In the more versatile *indirect* type of analog computation, a mathematical model rather than a physical model is generated, and the mathematical model is solved by the proper selection and assemblage of electronic computing elements. A symbolic system is utilized to show a proper interconnection of computing elements required to solve the mathematical equations.

The electronic differential analyzer is the most frequently used general-purpose analog computer for the solution of ordinary differential equations and for real-time simulation. Dependent variables are represented in the computer by voltages and the independent variable by real time. Essentially, an electronic differential analyzer (analog computer) is an assembly of operational amplifiers and related equipment suitable for solving sets of ordinary, simultaneous, nonlinear differential equations. Operational amplifiers are capable of receiving several voltage inputs, multiplying them by a constant, and summing and/or integrating the results with respect to time. In the computer, they are usually associated with a stable reference voltage and power supply, a number of potentiometers for multiplying by fractions, multipliers for multiplying by variables, resolvers for changing coordinate systems, function generators for introducing empirical relations, switches, relays and associated components for controlling and

*Much of the material in this section was extracted freely from a report (Ref. 11) prepared by Sam F. Heal of the Systems Simulation Branch of the U. S. Army Tank-Automotive Center.

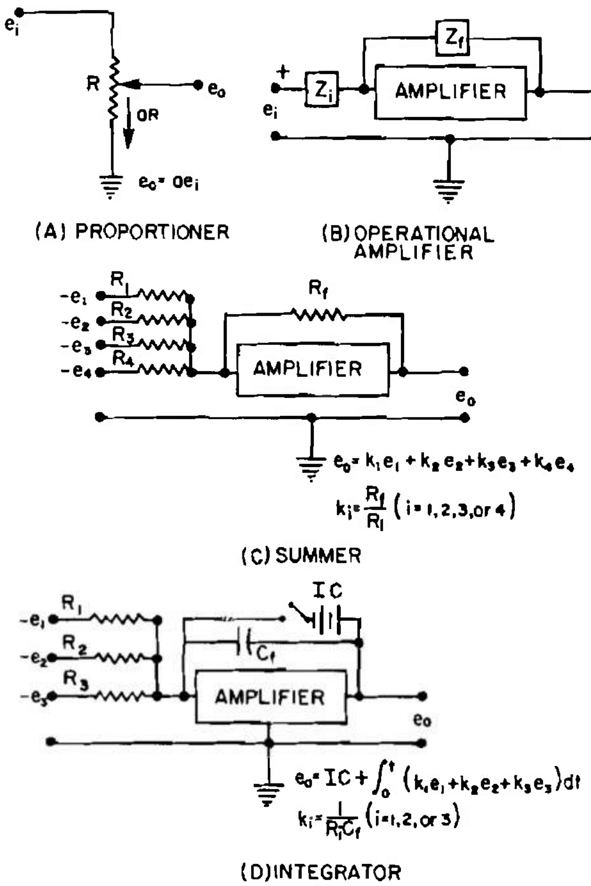


Figure 8-13. Linear Computing Devices

monitoring operation, and, to record results, recorders and plotting boards. All of these components operate on voltages which are analogous to the variables of the physical system and as such may be used to determine the dynamic behavior of the physical system.

The three basic elements of electronic differential analyzers can be characterized as linear computing devices, nonlinear computing devices, and recorders.

With *linear computing devices,* the output voltage is always proportional to the input voltage. The basic linear computing devices are (a) proportioners, (b) operational amplifiers, (c) summers, and (d) integrators. These are diagramed in Figure 8-13. *Proportioners* are precise potentiometers that are used to multiply a voltage by a constant coefficient a, where $0 \leqq a \leqq 1$. *Summers* employ a high gain direct coupled amplifier in conjunction with an input impedance Z_i and a feedback impedance Z_f to form an *operational amplifier*. The operation performed by a given amplifier depends on the nature of Z_i and Z_f. If both impedances are resistive ($Z_i = R_i$ and $Z_f = R_f$, then $e_o = -k_ie_i$ where $k_i = R_f/R_i$), then the operation performed is multiplication by a constant. The sign inversion is a result of the negative feedback of the amplifier. Since the operational amplifier is a linear device, the theorem of superposition applies for multiple inputs, resulting in a *summer* such that the output voltage is the sum of the imput voltages.

An *integrator* is produced when the feedback impedance of the amplifier is capacitive and the input impedance is resistive ($Z_i = R_i$ and $Z_f = C_f$, then $e_o = k_i \int_0^t e_i dt$ where $k_i = 1/R_iC_f$). The operation performed is integration and superposition applies for multiple inputs.

As shown in Figure 8-13, provision is made for introducing initial conditions (IC) into the integrator by charging the feedback capacitor to a prescribed initial voltage before a problem solution is initiated.

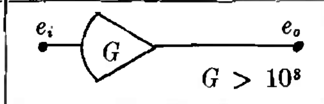
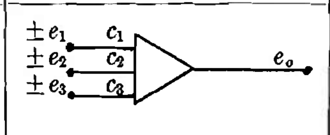
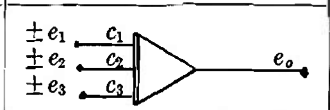
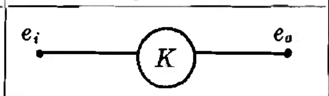
A *differentiator* results when the feedback impedance of the operational amplifier is resistive and the input impedance is capacitive ($Z_i = C_i$ and $Z_f = R_f$, then $e_o = -de_i/dt$). Differentiators are not practical for computer applications since the differentiation process inherently amplifies the error (noise) voltages instead of attenuating them as does the averaging process of integration.

When the input and/or feedback impedances of the operational amplifier are composed of combinations of resistors and capacitors, a wide variety of additional mathematical operations can be performed.

The techniques of simulation permit the determination of the dynamic behavior of a particular physical system once the behavior of the system has been described and formulated into mathematical statements.

Table 8-1 shows some of the basic devices and symbols used in the formulation of a simulation circuit. Figure 8-14 shows a simple simulation circuit for the solution of a given differential equation.

TABLE 8-1
BASIC SIMULATION SYMBOLS

Name	Symbol	Function	Description
High Gain Amplifier		$e_o = -Ge_i$	Operational Amplifier
Summer		$e_o = -(\pm c_1e_1 \pm c_2e_2 \pm c_3e_3)$	Amplifier Multiple Input
Integrator		$e_o = -\int_0^t (\pm c_1e_1 \pm c_2e_2 \pm c_3e_3)\,dt$	Amplifier Multiple Input
Coefficient Potentiometer		$e_o = Ke_1 \quad 0 < K < 1$	Manually Set Potentiometer

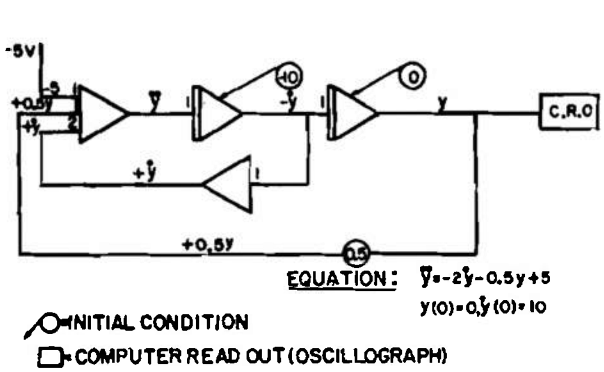


Figure 8-14. Simulation Circuit Diagram

Since many physical systems require the use of nonlinear differential equations to describe their behavior, nonlinear computing devices have been developed. With *nonlinear computing devices,* provision is made for performing operations in which the output voltage has a nonlinear relationship to the input voltage.

Some of the most frequently used nonlinear devices are voltage multipliers, resolvers, function generators, and limiters. Multipliers are required for calculating the products of two variable voltages. These units are either all electronic multiplying circuits (using either modulated square or trapezoidal waves or special diode function generators) or electromechanical devices. One electromechanical system (servomultiplier) utilizes a

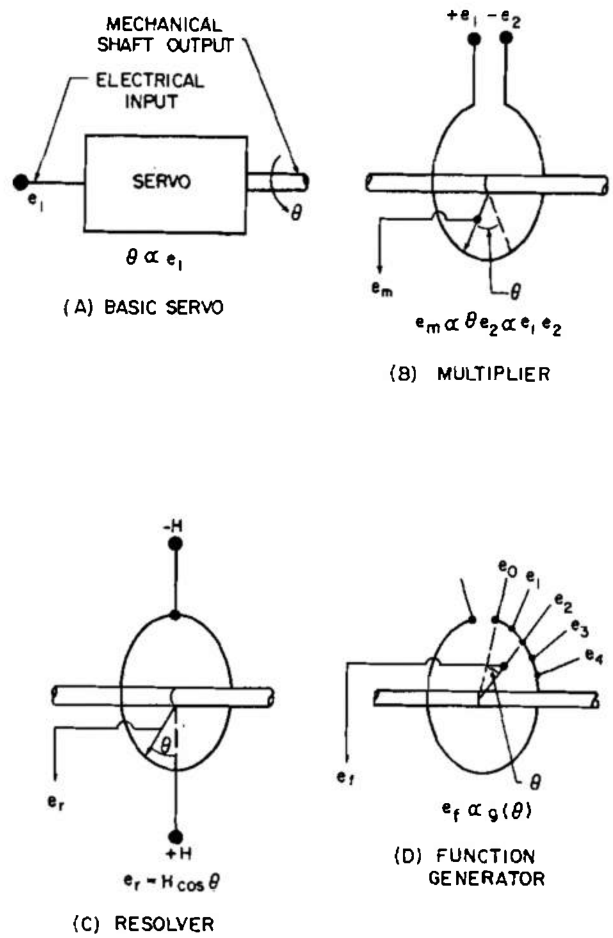


Figure 8-15. Typical Nonlinear Computing Devices

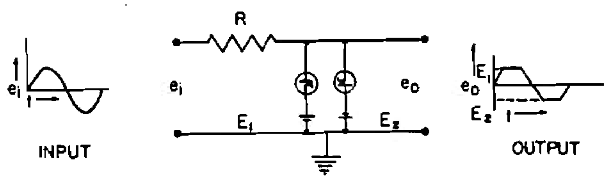


Figure 8-16. Limiter

coefficient potentiometer driven by a servo-motor, as shown in Figures 8-15 (A) & (B). The voltages e_1 to the servo and e_2 to the potentiometer are the variable inputs to be multiplied. The angular displacement of the servo shaft Θ is proportional to the input voltage e_1. The potentiometer wiper is driven by the servo-motor so that e_m, the output voltage, is proportional to Θ and e_2; hence, $e_m \propto e_1e_2$.

Resolvers are used to generate output voltages that vary as the sine and cosine of the input voltage. One type of resolver shown in Figure 8-15(C) uses a basic servo-motor and a resolving potentiometer which is wound so the $e_r = H \cos \Theta$. Another resolving wiper located 90° from the one shown would yield an output of $H \sin \Theta$.

Function generators have the capability of generating output voltages bearing definite functional relationships to the input voltage. A diode function generator can reproduce nearly any arbitrary function with respect to an input variable. One type of function generator is shown in Figure 8-15(D). This device employs the basic servo-motor and a function potentiometer which is linearly wound and tapped at a number of points. The output voltage e_f will be a linear interpolation between the voltages maintained at the taps. Any single-valued function $f(\Theta)$ can be approximated by maintaining tap voltages which are proportional to function ordinates.

Limiters are used in simulation circuits where physical variables have fixed limits. For example, the vertical displacement of a road wheel may be limited by shock absorber blow-off. Voltages representing such variables in a simulation circuit can be constrained or limited to define values by means of diodes. The output voltage of a circuit represented by Figure 8-16, is limited to the bias voltages in the diode circuits.

The readout devices or recorders used with electronic differential analyzers are essentially recording voltmeters which compare or plot a computer voltage against computer time or against another computer voltage. Oscilloscopes, oscillographs, digital voltmeters, and XY plotters are used most frequently for computer readout.

8-31.3 COMPUTER COMMUNICATIONS

In order to set up the computer for the solution of mathematical models, it is necessary to generate the proper simulation circuits or "roadmaps" and to select the proper scale factors.

The simulation circuits or roadmaps are diagrams such as Figures 8-18(A) and 8-18(B) which show the proper interconnection of computing elements required to solve the equations. The scale factors, as expressed by a set of scaling equations, define the proportionality between the mathematical variables of the model and the analogous computer variables. Voltage scale factors are chosen as large as possible so that the voltages representing the mathematical variables are significantly larger than the noise in the computer. Voltage magnitudes are limited, usually to ± 100 volts, by the linear range of operation for which the electronic components are designed.

A dependent variable, such as x, is related to the computer voltage X by the equation $X = bx$, in which b is the scale factor. The mathematical independent variable t is related to the computer independent variable T by the scaling equation $T = at$.

When $a = 1$, the computation is a real time simulation and the computer time corresponds to the actual problem time; when $a < 1$, the computer time required for solution will be less than the mathematical time. Conversely when $a > 1$, the computer time is greater than real time.

8-32 DESCRIPTION OF SYMBOLS USED IN SUSPENSION STUDY

W_S sprung weight of vehicle
J pitch moment of inertia
J_o 1/2 pitch moment of inertia
W_U unsprung weight of vehicle
N number of wheels
M_o 1/2 sprung mass

M_W 1/10 unsprung mass

l_1, l_2, l_3, l_4, l_5 distance from vertical centerline of wheels 1, 2, 3, 4, and 5, respectively, to center of gravity of sprung mass

K_1, K_2, K_3, K_4, K_5 spring constants of individual spring on one side of vehicle

D_1, D_5 damping constants of front and rear shock absorbers, respectively

K_B spring constant of bump stop

K_W spring constant of rubber on road wheel

D_W damping constant of rubber on road wheel

l_{b1}, l_{b2}, l_{b3}, l_{b4}, l_{b5} static to bump stop dimensions for each wheel

f_Y natural bounce frequency

f_Θ natural pitch frequency

f_W natural wheel frequency

D_{CW} critical damping for wheel

ζ_W damping coefficient for bounce

D_{CY} critical damping for bounce

ζ_Y damping coefficient for bounce

$D_{C\Theta}$ critical damping for pitch

ζ_Θ damping coefficient for pitch

$\ddot{Y}_o$ vertical acceleration of center of gravity

$\dot{Y}_o$ vertical velocity of center of gravity

Y_o vertical displacement of center of gravity

$\ddot{\Theta}_o$ pitch acceleration about center of gravity

$\dot{\Theta}_o$ pitch velocity about center of gravity

Θ_o pitch displacement about center of gravity

Y_1, Y_2, Y_3, Y_4, Y_5 vertical displacement of hull directly above wheels 1 through 5

Y_1, Y_5 vertical velocity of hull directly above wheel 1 and wheel 5

Y_{W1}, Y_{W2}, Y_{W3}, Y_{W4}, Y_{W5} vertical displacement of wheels 1, 2, 3, 4, and 5, respectively

$\dot{Y}_{W1}$, $\dot{Y}_{W2}$, $\dot{Y}_{W3}$, $\dot{Y}_{W4}$, $\dot{Y}_{W5}$ vertical velocity of wheels 1, 2, 3, 4, and 5, respectively

a_1, a_2, a_3, a_4, a_5 road inputs to wheels 1, 2, 3, 4, and 5, respectively

g acceleration of gravity

8-33 DISCUSSION

For analytical purposes, a vehicle suspension can be considered as a combination of masses, springs, and dampers. Thus, the vehicle hull or frame structure is represented as a large mass, each wheel as a small mass, the torsion bars and rubber of the roadwheel as springs, and the shock absorbers as dampers. Once a suspension is put into this form, it can be described by a set of differential equations derived from Newton's laws of motion. A typical five-wheel tracked vehicle in the 15- to 21-ton weight class is chosen for discussion here. Furthermore, only the more severe motions, vehicle bounce and pitch, are studied. Since the roll, yaw, and surge motions are not considered, the vehicle may be separated along the longitudinal axis. Hence, only the wheels, springs, and shock absorbers of one side need to be discussed. Figure 8-17 presents a block diagram of the vehicle configuration.

The simulation equations of motion are written with constant coefficients. This form permits ready manipulation. In general, nonlinearities enter via the variables of the equations rather than the coefficients. These nonlinearities would include bump stops, double acting blow-off type shock absorbers, wheels leaving the ground, etc. To reduce the quantity and type of equipment required for normal suspension system analysis, certain assumptions have been applied

(a) Sine of the angle is equal to the angle in radians for small angles.

(b) Hull, road arms, and wheel hubs are solid, nondeformable bodies.

(c) Track effect does not significantly affect the simulation.

(d) Springs are linear except at bump stop position.

(e) Vehicle forward speed is constant.

(f) Suspension system friction in negligible in comparison to shock absorber damping.

8-33.1 SIMULATION EQUATIONS

8-33.1.1 Vertical Motion of Hull

$$\ddot{Y}_o = -\frac{K_1}{M_o}(Y_1 - Y_{W1}) - \frac{K_2}{M_o}(Y_2 - Y_{W2}) - \frac{K_3}{M_o}(Y_3 - Y_{W3}) - \frac{K_4}{M_o}(Y_4 - Y_{w4})$$
$$- \frac{K_5}{M_o}(Y_5 - Y_{W5}) - \frac{D_1}{M_o}(\dot{Y}_1 - \dot{Y}_{W1}) - \frac{D_5}{M_o}(\dot{Y}_5 - \dot{Y}_{W5}) + g$$

8-33.1.2 Pitch Motion of Hull

$$\ddot{\Theta}_o = -\frac{K_1 l_1}{J_o}(Y_1 - Y_{W1}) - \frac{K_2 l_2}{J_o}(Y_2 - Y_{W2}) - \frac{K_3 l_3}{J_o}(Y_3 - Y_{W3}) + \frac{K_4 l_4}{J_o}(Y_4 - Y_{W4})$$
$$+ \frac{K_5 l_5}{J_o}(Y_5 - Y_{W5}) - \frac{D_1 l_1}{J_o}(\dot{Y}_1 - \dot{Y}_{W1}) + \frac{D_5 l_5}{J_o}(\dot{Y}_5 - \dot{Y}_{W5})$$

8-33.1.3 Vertical Motion of Wheels

$$\ddot{Y}_{W1} = -\frac{K_{W1}}{M_{W1}}(Y_{W1} - a_1) - \frac{D_{W1}}{M_{W1}}(\dot{Y}_{W1} - \dot{a}_1) + \frac{K_1}{M_{W1}}(Y_1 - Y_{W1}) + \frac{D_1}{M_{W1}}(\dot{Y}_1 - \dot{Y}_{W1}) + g$$

$$\ddot{Y}_{W2} = -\frac{K_{W2}}{M_{W2}}(Y_{W2} - a_2) - \frac{D_{W2}}{M_{W2}}(\dot{Y}_{W2} - \dot{a}_2) + \frac{K_2}{M_{W2}}(Y_2 - Y_{W2}) + g$$

$$\ddot{Y}_{W3} = -\frac{K_{W3}}{M_{W3}}(Y_{W3} - a_3) - \frac{D_{W3}}{M_{W3}}(\dot{Y}_{W3} - \dot{a}_3) + \frac{K_3}{M_{W3}}(Y_3 - Y_{W3}) + g$$

$$\ddot{Y}_{W4} = -\frac{K_{W4}}{M_{W4}}(Y_{W4} - a_4) - \frac{D_{W4}}{M_{W4}}(\dot{Y}_{W4} - \dot{a}_4) + \frac{K_4}{M_{W4}}(Y_4 - Y_{W4}) + g$$

$$\ddot{Y}_{W5} = -\frac{K_{W5}}{M_{W5}}(Y_{W5} - a_5) - \frac{D_{W5}}{M_{W5}}(\dot{Y}_{W5} - \dot{a}_5) + \frac{K_5}{M_{W5}}(Y_5 - Y_{W5}) + \frac{D_5}{M_{W5}}(\dot{Y}_5 - Y_{W5}) + g$$

8-33.1.4 Auxiliary Hull Equations

$$Y_1 = Y_o + l_1 \sin\Theta \qquad \dot{Y}_1 = \dot{Y}_o + (l_1 \cos\Theta)\dot{\Theta} = \dot{Y}_o + l_1\dot{\Theta}$$
$$Y_2 = Y_o + l_2 \sin\Theta$$
$$Y_3 = Y_o + l_3 \sin\Theta$$
$$Y_4 = Y_o - l_4 \sin\Theta$$
$$Y_5 = Y_o - l_5 \sin\Theta \qquad \dot{Y}_5 = \dot{Y}_o - (l_5 \cos\Theta)\dot{\Theta} = \dot{Y}_o - l_5\dot{\Theta}$$

For small angles assume:
$\sin\Theta = \Theta$ in radians
$\cos\Theta = l$

Prior to establishing the final equations of motion, it is necessary to collect data describing the vehicle to be simulated. These data provide such characteristics as spring rates, shock absorber constants, center of gravity location, weight, polar moment of inertia, and road wheel rubber characteristics. In addition, it is necessary to know the horizontal dimensions from each road wheel to the center of gravity of the sprung mass.

8-33.2 CONSTANTS AND NONLINEARITIES

For typical track-laying vehicles in the 15- to 21-ton weight class, the constants and nonlinearities listed in the following paragraph were chosen.

8-33.2.1 Vehicle Constants Used in the Simulation

W_S (Sprung Wt) = 28,000 lb
J (Pitch Inertia) = 27,000 slug-ft^2
$\frac{1}{2}J = J_o = 13{,}500$ slug-ft^2
W_U (Unsprung Wt) = 3000 lb
N (Number of Wheels) = 10
$g = 32.16$ fps^2

$$M_o = \frac{W_S}{2g} = 435.3 \text{ slugs}$$
$$M_W = \frac{W_U}{10g} = 9.328 \text{ slugs}$$

$l_1 = 6$ ft	$K_1 = 11{,}500$ lb/ft
$l_2 = 3$ ft	$K_2 = 7{,}000$ lb/ft
$l_3 = 1$ ft	$K_3 = 7{,}000$ lb/ft
$l_4 = 2$ ft	$K_4 = 7{,}000$ lb/ft

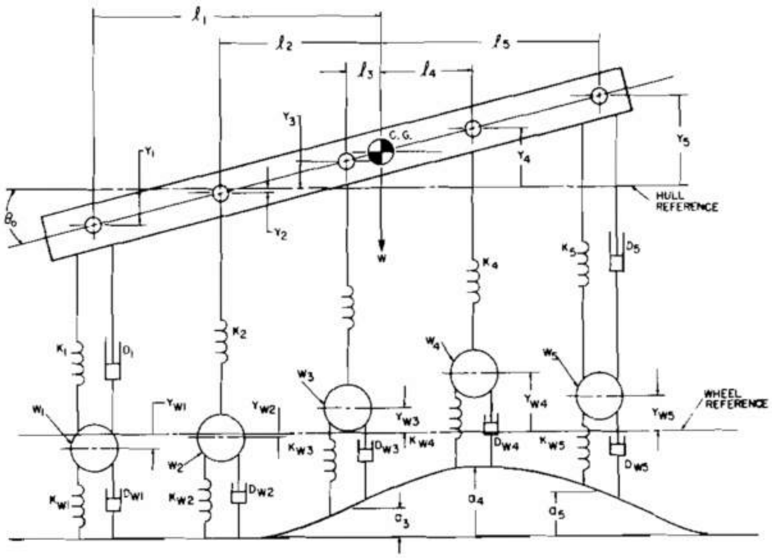


Figure 8-17. Block Diagram of Vehicle Suspension

$l_5 = 5$ ft $\qquad K_5 = 11{,}500$ lb/ft
$D_1 = 800$ lb-sec/ft (comp); 1600 (rebound)
$D_5 = 800$ lb-sec/ft (comp); 1600 (rebound)
$K_{B1}, K_{B5} = 1{,}000{,}000$ lb/ft
$K_{B2}, K_{B3}, K_{B4} = 500{,}000$ lb/ft
$D_W = 400$ lb-sec/ft
$K_W = 400{,}000$ lb/ft

Static to Bump Stop Dimensions

Wheel 1, $l_{B1} = 6.00$ in.
Wheel 2, $l_{B2} = 5.00$ in.
Wheel 3, $l_{B3} = 5.00$ in.
Wheel 4, $l_{B4} = 5.00$ in.
Wheel 5, $l_{B5} = 6.00$ in.

8-33.2.2 Nonlinearities Used in the Simulation

$K_1 = 11{,}500$ lb/ft for $(Y_1 - Y_{W1}) > -l_{b1}$
$K_1 = 1{,}000{,}000$ lb/ft for $(Y_1 - Y_{W1}) < -l_{b1}$
$K_2 = 7{,}000$ lb/ft for $(Y_2 - Y_{w2}) > -l_{b2}$
$K_2 = 500{,}000$ lb/ft for $(Y_2 - Y_{W2}) < -l_{b2}$
$K_3 = 7{,}000$ lb/ft for $(Y_3 - Y_{W3}) > -l_{b3}$
$K_3 = 500{,}000$ lb/ft for $(Y_3 - Y_{W3}) < -l_{b3}$
$K_4 = 7{,}000$ lb/ft for $(Y_4 - Y_{W4}) > -l_{b4}$
$K_4 = 500{,}000$ lb/ft for $(Y_5 - Y_{W5}) < -l_{b4}$
$K_5 = 11{,}500$ lb/ft for $(Y_1 - Y_{W1}) > -l_{b1}$
$K_5 = 1{,}000{,}000$ lb/ft for $(Y_1 - Y_{W1}) < -l_{b1}$
$D_1 = 800$ lb-sec/ft for $(\dot{Y}_1 - \dot{Y}_{W1}) < 0$
$D_1 = 1600$ lb-sec/ft for $(\dot{Y}_1 - \dot{Y}_{W1}) > 0$
$D_1 =$ Partially limited to 1000 for $(\dot{Y}_1 - \dot{Y}_{W1}) < 0$
$D_1 =$ Partially limited to 2000 for $(\dot{Y}_1 - \dot{Y}_{W1}) > 0$.
$D_5 = 800$ lb-sec/ft for $(\dot{Y}_5 - \dot{Y}_{W5}) < 0$
$D_5 = 1600$ lb-sec/ft for $(\dot{Y}_5 - \dot{Y}_{W5}) > 0$
$D_5 =$ Partially limited to 1000 for $(\dot{Y}_5 - \dot{Y}_W) < 0$
$D_5 =$ Partially limited to 2000 for $(\dot{Y}_5 - \dot{Y}_{W5}) > 0$
$K_{W1} = 400{,}000$ lb/ft for $(Y_{W1} - a_1) < 0$
$K_{W1} = 0$ for $(Y_{W1} - a_1) > 0$
$K_{W2} = 400{,}000$ lb/ft for $(Y_{W2} - a_2) < 0$
$K_{W2} = 0$ for $(Y_{W2} - a_2) > 0$
$K_{W3} = 400{,}000$ lb/ft for $(Y_{W3} - a_3) < 0$
$K_{W3} = 0$ for $(Y_{W3} - a_3) > 0$
$K_{W4} = 400{,}000$ lb/ft for $(Y_{W4} - a_4) < 0$
$K_{W4} = 0$ for $(Y_{W4} - a_4) > 0$
$K_{W5} = 400{,}000$ lb/ft for $(Y_{W5} - a_5) < 0$
$K_{W5} = 0$ for $(Y_{W5} - a_5) > 0$

8-33.3 CALCULATED COEFFICIENTS AND NECESSARY DYNAMIC CHARACTERISTICS

From the constants and nonlinearities above, the coefficients for the differential equations are calculated. Other dynamic characteristics of the suspension are determined for problem board checking. Calculated coefficients and necessary dynamic characteristics for the selected sample vehicle are listed in the paragraphs which follow.

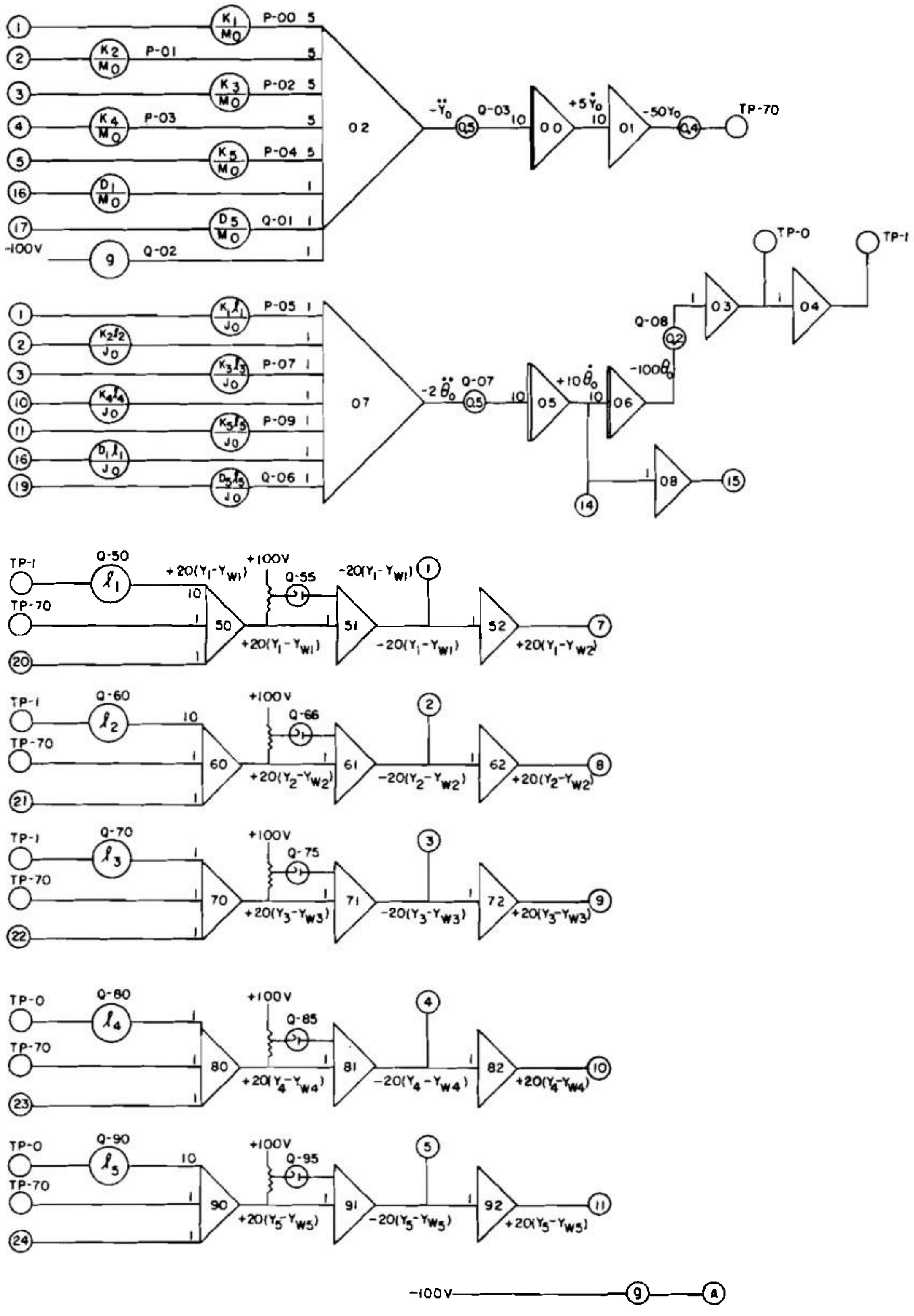


Figure 8-18 (A). Computer Circuit Diagram for Tracked Vehicle Suspension

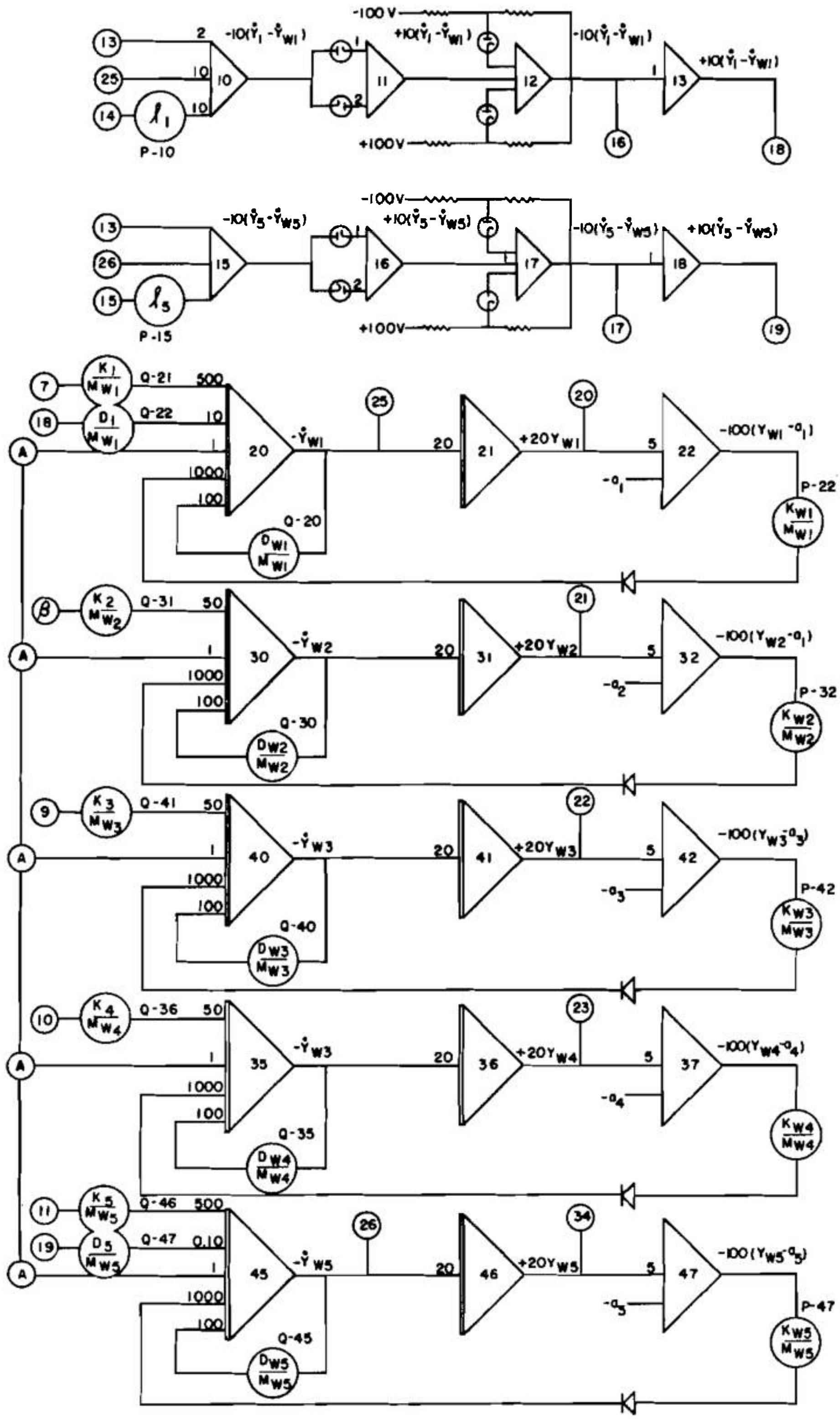


Figure 8-18 (B). Computer Circuit Diagram for Tracked Vehicle Suspension

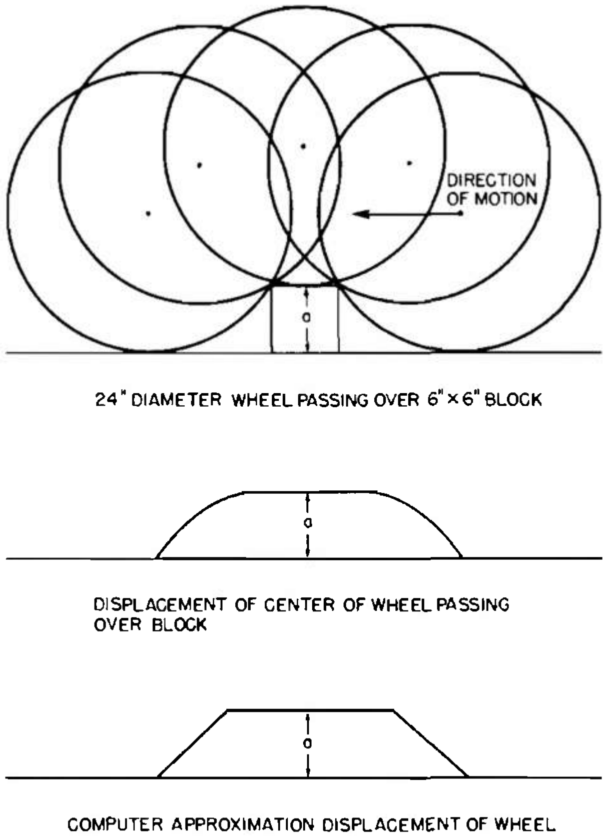


Figure 8-19. Wheel-Displacement Configuration

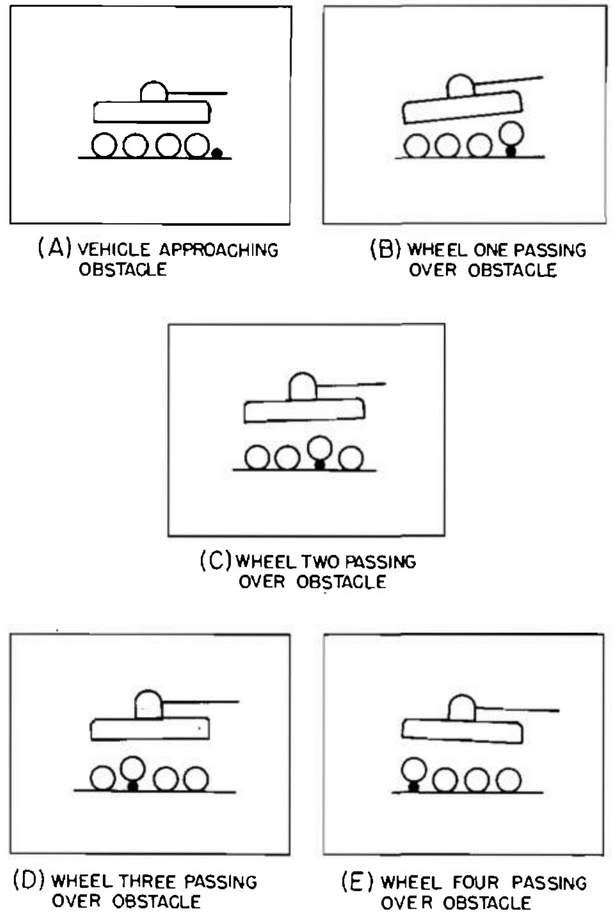


Figure 8-20. Simulscope Patterns

8-33.3.1 Calculation of Potentiometer Settings

$$\frac{K_1}{M_o} = 26.42 \qquad \frac{K_3}{M_W} = 750 \qquad \frac{D_1 l_1}{J_o} = 0.356$$

$$\frac{K_2}{M_o} = 16.08 \qquad \frac{K_4}{M_W} = 750 \qquad \frac{D_5 l_5}{J_o} = 0.296$$

$$\frac{K_3}{M_o} = 16.08 \qquad \frac{K_5}{M_W} = 1233 \qquad \frac{D_1}{M_W} = 85.8$$

$$\frac{K_4}{M_o} = 16.08 \qquad \frac{K_W}{M_W} = 42880 \qquad \frac{D_5}{M_W} = 85.8$$

$$\frac{K_5}{M_o} = 26.42 \qquad \frac{K_1 l_1}{J_o} = 5.111 \qquad \frac{D_W}{M_W} = 42.9$$

$$\frac{D_1}{M_o} = 1.838 \qquad \frac{K_2 l_2}{J_o} = 1.493$$

$$\frac{D_5}{M_o} = 1.838 \qquad \frac{K_3 l_3}{J_o} = 0.498$$

$$\frac{K_1}{M_W} = 1233 \qquad \frac{K_4 l_4}{J_o} = 0.996$$

$$\frac{K_2}{M_W} = 750 \qquad \frac{K_5 l_5}{J_o} = 4.259$$

8-33.3.2 Calculation of Natural Frequencies and Damping Coefficients

$$f_Y = \frac{1}{2\pi}\sqrt{\frac{K_1 + \ldots\ldots + K_5}{M_o}} = \frac{1}{2\pi}\sqrt{101.08}$$

$$f_Y = 1.601 \text{ cps}$$

$$f_\Theta = \frac{1}{2\pi}\sqrt{\frac{K_1 l_1^2 + \ldots\ldots + K_5 l_5^2}{J_o}} = \frac{1}{2\pi}\sqrt{58.930}$$

$$= 1.222 \text{ cps}$$

$$f_W = \frac{1}{2\pi}\sqrt{\frac{K_W}{M_W}} = \frac{1}{2\pi}\sqrt{42{,}880} = 32.95 \text{ cps}$$

$$D_{CW} = 2\sqrt{K_W M_W} = 2\sqrt{3{,}731{,}200}$$

$$= 3864 \text{ lb-sec/ft}$$

$$\zeta_W = \frac{D_W}{D_{CW}} = \frac{400}{3864} = 0.1035$$

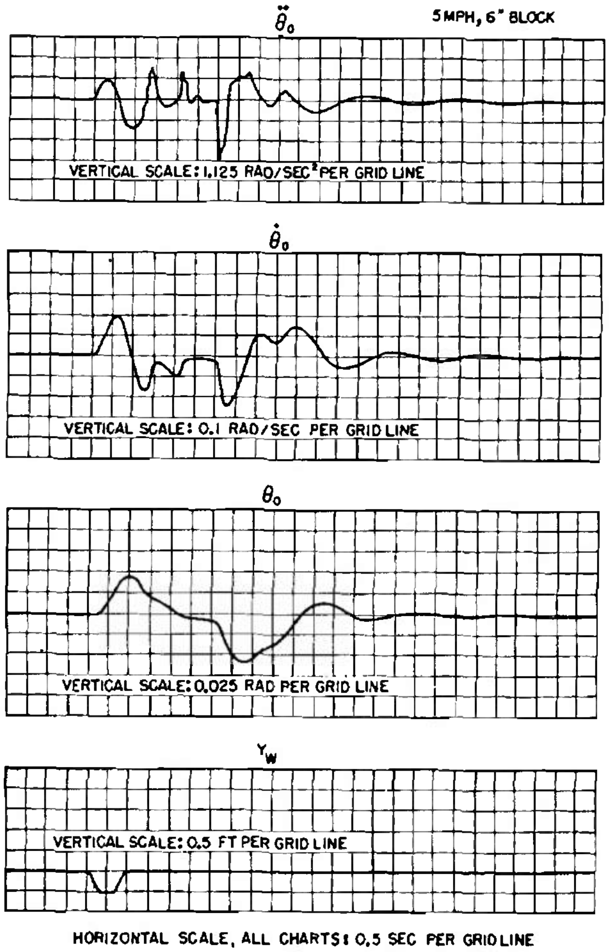


Figure 8-21 (A). Oscillograph Chart

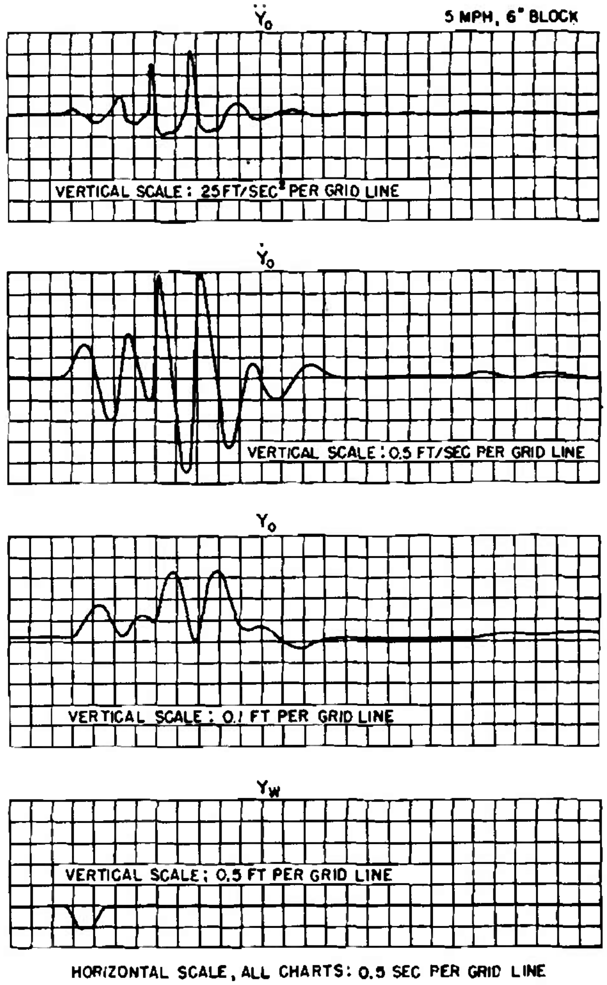


Figure 8-21 (B). Oscillograph Chart

$$D_{CY} = 2\sqrt{(K_1 + \ldots\ldots + K_5)\, M_o}$$
$$= 2\sqrt{18{,}774{,}600} = 8666 \text{ lb-sec/ft}$$

$$\zeta_Y = \frac{D_1 + D_5}{D_{CY}} = \frac{1600}{8666} = 0.1846$$

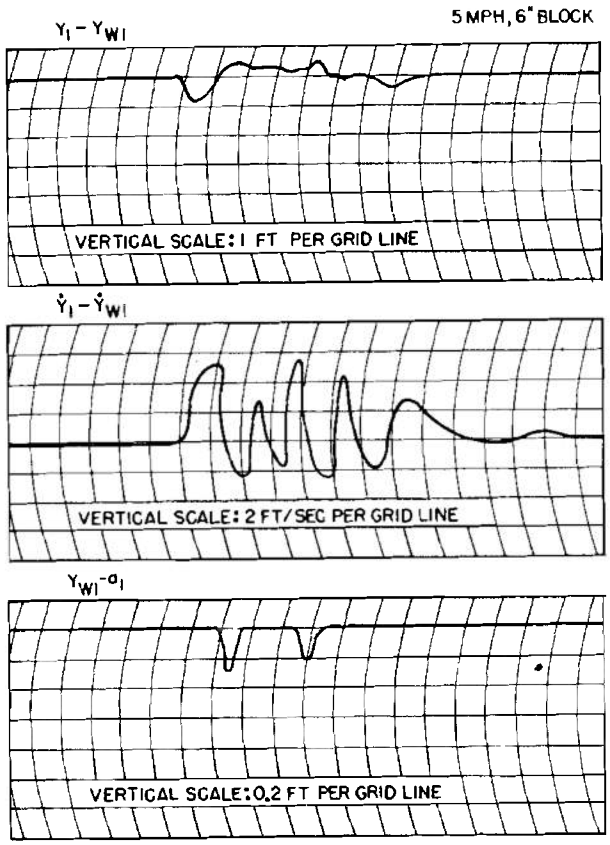
$$D_{C\Theta} = 2\sqrt{(K_1 l_1^2 + \ldots\ldots + K_5 l_5^2)\, J_o}$$
$$= 2\sqrt{10{,}793{,}250{,}000}$$
$$= 207{,}780 \text{ lb-sec/ft}$$

$$\zeta_\Theta = \frac{D_1 l_1^2 + D_5 l_5^2}{D_{C\Theta}} = \frac{48{,}800}{207{,}780} = 0.235$$

8-33.4 GENERAL PROCEDURE AND RESULTS

Once these constants are obtained, an analog computer board wiring diagram may be drawn. Such diagrams are commonly referred to as "roadmaps." Simulation roadmaps Figures 8-18(A) and (B) show in detail the vehicle constants, variables, and nonlinearities in terms of voltages through the use of amplifiers, potentiometers, and resistances. Thus, these functions must be scaled to obtain physical quantities such as loads, displacements, velocities, and accelerations. Figures 8-18 (A) and (B) are the roadmaps for the vehicle represented by Figure 8-17.

The computer board is wired directly from the roadmap; however, due to wiring complexity, it must be checked in detail. This involves the isolation of individual oscillatory systems to determine natural frequencies and damping ratios. The various nonlinearities are isolated from the balance of the computer circuitry during the checking process and are checked individually. Generally,

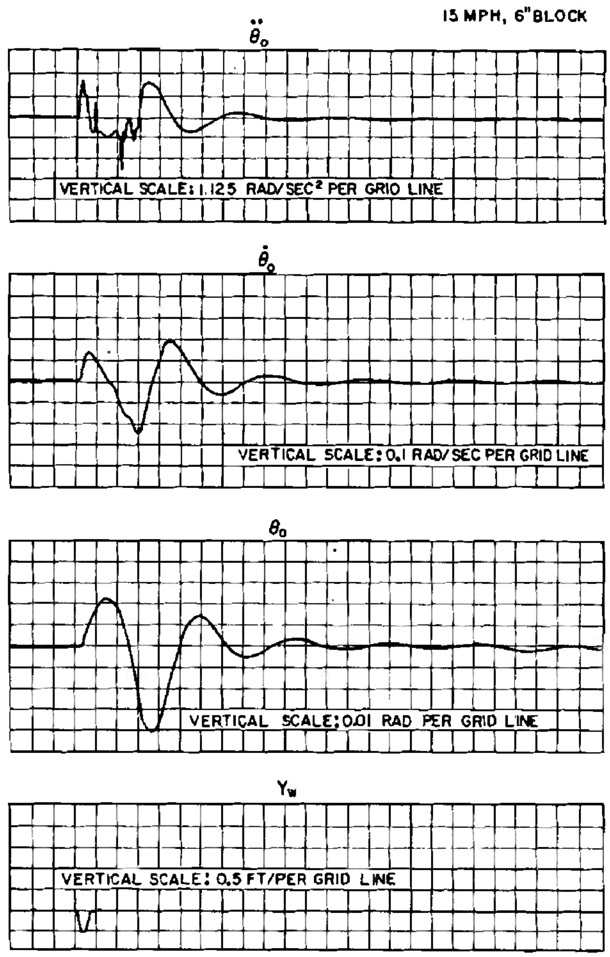


HORIZONTAL SCALE, ALL CHARTS: 0.5 SEC PER GRID LINE

Figure 8-21 (C). Oscillograph Chart

it is wise to perform certain static checks of the entire simulation prior to problem runs. For example, displace the front wheel and record the vehicle pitch attitude and C.G. movement against previously calculated values.

The Systems Simulation Branch of the U.S. Army Tank-Automotive Center has facilities for generating most terrain configurations encountered in a military environment. Special devices have been constructed to generate geometric bump inputs, road waveforms, and any pre-recorded random terrain countour in the laboratory. The latter device is used to simulate given terrain such as the Aberdeen Proving Ground Belgian block and the Camp Perrymann test courses. However, for purposes of illustration, a six-inch square block is used in this discussion. This particular road input adopts the technique of modifying the square block to correct for wheel diameter rather than enter the



HORIZONTAL SCALE, ALL CHARTS: 0.5 SEC PER GRID LINE

Figure 8-22 (A). Oscillograph Chart

wheel dimension into the vehicle simulation. This modification is shown in Figure 8-19.

The readout of computer results is by direct writing oscillographs. Other means are also available. The Systems Simulation Branch at ATAC is equipped with a special purpose cathode ray tube device, called a simulscope, which displays a pictorial representation of vehicle motion. Not only does the simulscope aid in vehicle suspension response studies, but serves as a useful tool in problem board checking. Typical simulscope presentations of a four-wheeled vehicle passing over a square block and are shown in Figure 8-20.

For the sample suspension study the following variables were recorded on the oscillograph:

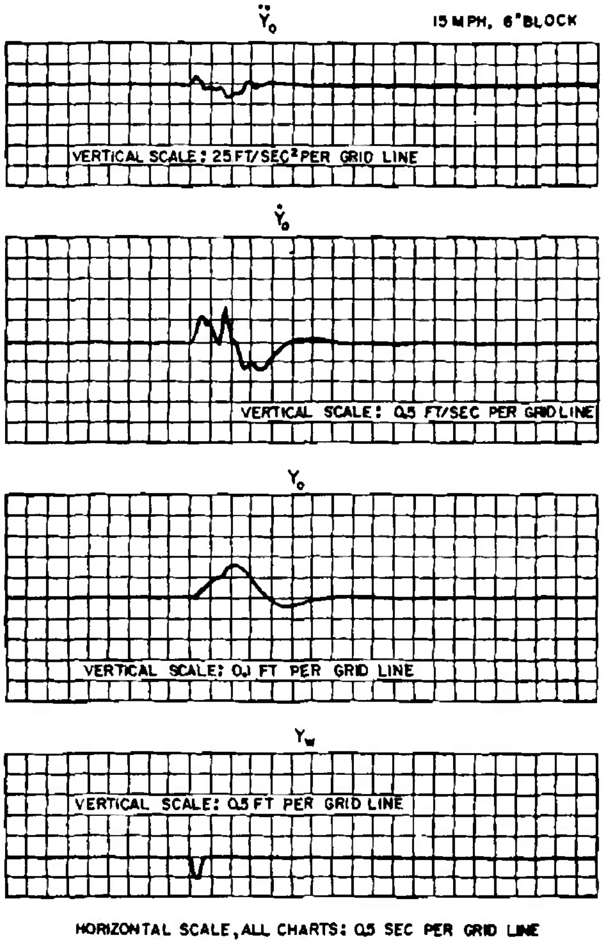


Figure 8-22 (B). Oscillograph Chart

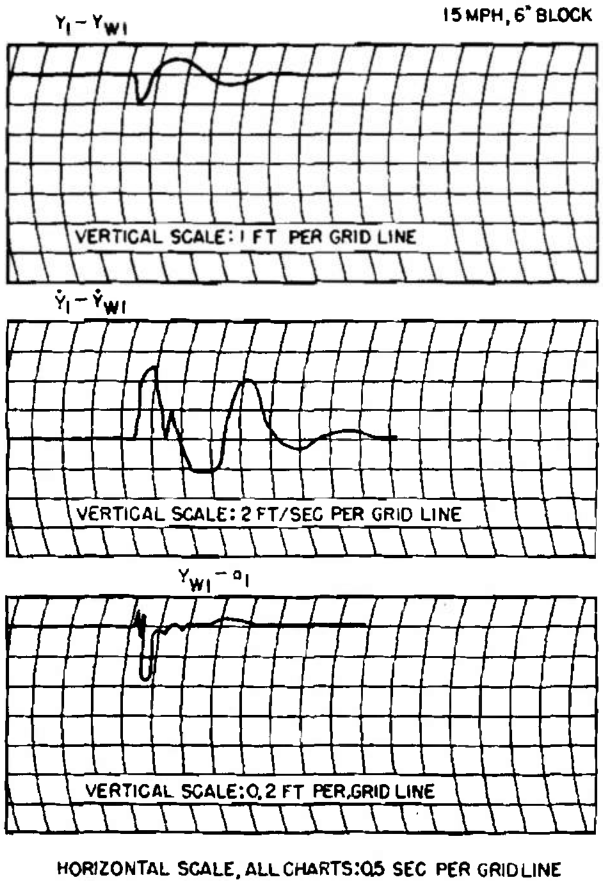


Figure 8-22 (C). Oscillograph Chart

$\ddot{Y}_o$ vertical acceleration of the center of gravity.
$\dot{Y}_o$ vertical velocity of C. G.
Y_o vertical displacement of C. G.
$\ddot{\Theta}_o$ pitch acceleration about C. G.
$\dot{\Theta}_o$ pitch velocity about C. G.
Θ_o pitch displacement at C. G.
$(Y_1 - Y_{W1})$ relative displacement between hull and wheel at wheel 1.
$(\dot{Y}_1 - \dot{Y}_{W1})$ relative velocity of the hull and wheel at wheel 1.
$(Y_{W1} - a_1)$ relative displacement between wheel and the input bump at wheel 1.

Typical recordings of these variables are shown for two different vehicle speeds (Figures 8-21 and 8-22). From the latter three variables the dynamic spring, shock absorber, and wheel loads for the front wheel may be calculated. The dynamic spring load is calculated by multiplying the displacement from the static position $(Y_1 - Y_{W1})$ by the spring constant K_1. Similarly, the shock absorber and wheel loads may be evaluated by multiplying $(\dot{Y}_1 - \dot{Y}_{W1})$ and $(Y_{W1} - a_1)$ by the damping constant D_1 (compression) and by the road wheel rubber spring constant K_{W1}, respectively.

The use of the analog computer for suspension analysis leads to a better understanding of specific suspension characteristics and design limitations. In a sense, the analog computer becomes the vehicle itself; thus, the analog computer simulation may lead to greater refinement of product design in prototype vehicle versions.

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CHAPTER 9

SUSPENSIONS FOR WHEELED VEHICLES*

SECTION I GENERAL DISCUSSION

9-1 INTRODUCTION (Ref. 1)

Suspension systems for wheeled vehicles are classified into two major categories; namely, nonindependent types and independent types, according to the mechanical interrelationship of the wheels. Nonindependent suspension systems are characterized by rigid transverse structural members connecting two or more of the wheels. With this arrangement, a vertical displacement of one wheel causes a displacement of the interconnected wheel or wheels. Figure 9-1 shows some typical nonindependent (or solid axle) suspension systems.

Independent suspension systems are characterized by the lack of rigid structural members interconnecting the individual wheels. With this arrangement, a vertical displacement of one wheel does not affect the remaining wheels directly. Typical suspension systems of this type are shown in Figures 9-3 and 9-6 to 9-8, inclusive.

The purposes and functions of both types of suspension systems are the same but each has some operational advantages over the other. These will be discussed in later sections of this chapter.

In addition to various arrangements of the mechanism, various spring and damping systems have been utilized for vehicular suspensions.

The types of elastic devices which have been used on wheeled vehicles include helical compression springs, leaf springs, torsion bars, pneumatic springs, hydraulic springs, hydropneumatic springs, torsion-elastic springs, volute and helical torsion springs. These spring types are discussed in detail in Chapter 10, Section VII.

Types of damping devices which have been used on experimental and standard military wheeled vehicles include hydraulic, friction, pneumatic, electromagnetic, dynamic, and inertial. These damping devices are discussed in detail in Chapter 10, Section VIII.

*Written by Nicholas R. Rome and Rudolph J. Zastera of the IIT Research Institute, Chicago, Ill.

9-2 DESIGN CONSIDERATIONS

9-2.1 GENERAL

Suspension systems for military vehicles must be very rugged to endure the high impact loading experienced when traveling over rough terrain. The anticipated loads experienced by the wheels are discussed in Chapter 2.

The high degree of refinement in ride quality found in civilian vehicles is not required for military vehicles but noise and high-frequency vibrations should be attenuated to minimize the detrimental effects on the crew and passengers. The comfort and ability of the crew to function properly are limited by the vibration frequency and amplitude of the body or hull. A detailed discussion of this subject is presented in Chapter 2.

The military suspension system must provide for sufficient vertical wheel displacement to allow for extreme terrain irregularities. Common practice is to design the suspension systems for multiwheeled vehicles so that any wheel will be capable of moving to any position from 12 inches above to 6 inches below its normal level standing position without increasing or decreasing the load it supports by more than 25 percent (Ref. 2). To prevent damage to the suspension system or other parts of the vehicle, bump stops must be provided.

Materials for the suspension components, particularly the springs, must be carefully selected for their resistance to fatigue. The design, heat treatment, and surface treatment must consider the significance of stress raisers and fatigue. For design purposes, it is suggested (Ref. 3) that material properties be taken from MIL-HDBK-5.

9-2.2 MAINTENANCE

All military vehicles must be capable of being maintained and serviced in the field, whether the vehicle is an experimental or a production model.

The experience gained from mock-ups, prototypes, and experimental vehicles is invaluable in reference to maintenance and servicing, as is the experience of the personnel who have been or are operating similar vehicles. Designing for maintenance is discussed in Chapter 2.

9-2.3 RELIABILITY (Ref. 4)

In general, *reliability* may be defined as the probability that an item will perform satisfactorily for a specified period of time when used in a manner and for the purpose intended. An item or system of suspension components is considered reliable if its actual performance satisfies the required performance with minimum down time for repairs. Consistent satisfactory performance is of greater value than occasional superior performance combined with unsatisfactory behavior.

The achievement of reliability is, to a great extent, a function of good design practice and the designer's recognition of the need and the importance of reliability. Therefore, it is essential for the designer to ascertain, as realistically and as early as possible, the specifications of the task, the environmental conditions, the human aspects, and the emergency conditions likely to be encountered. MIL-STD-1228 specifies a reliability criterion for wheeled vehicles that is to serve as a goal for designers of military vehicles. This criterion is that wheeled, tactical vehicles shall be capable of accomplishing 25,000 miles of operation in a military environment without field or depot maintenance (Chapter 2). Several basic considerations that will enhance the reliability of military equipment are discussed below.

9-2.3.1 Controlled Environment

In the event that the reliability of components is significantly affected by the environment in which it operates, the reliability can often be improved by controlling the environment. This can frequently be accomplished by employing devices such as shields, covers, or using additional components such as filters, air cleaners, and heaters. At times, the environment can be controlled by proper selection of component locations.

9-2.3.2 Selection of Materials and Construction Methods

Judicious selection of materials, construction methods, and designs will often provide increased reliability at the same or less cost. For example, the use of castings, forgings or weldments for various parts should be carefully considered.

9-2.3.3 Simplicity

In general, the reliability of a technical system decreases as the complexity increases. Making a design as simple as possible, commensurate with functional requirements, will tend to increase its reliability. In accomplishing this, the designer must avoid designing multiple functions into an item and should keep the parts of a system, as well as the interaction demanded of them, to an absolute minimum. The concept of simplicity should be also extended to the performance of the maintenance, service, and inspection functions.

9-2.3.4 Redundancy

Redundancy, in the technical sense, is the installing in a parallel arrangement more than one part, circuit, or component having the same or a satisfactory substitute function, for the purpose of increasing reliability. Two basic types of redundancy are frequently considered: standby and continuous.

Standby redundancy involves the use of parallel and similar units, circuits, or components which are automatically or manually brought into operation if the primary item fails.

Continuous redundancy involves the use of parallel and similar units, circuits, or components which are in continuous operation so that failure of the primary item will not terminate the performance but will result in not less than an acceptable level of performance.

The application of redundancy in design requires careful consideration of the consequences. In the case of simple component functions and for complete and easily detected failure, an inexpensive reliable detection and decision device may be used. However, for complex component functions, marginal failures or performance degradation, the detection and decision device may be so complex that the total reliability of the system is unsatisfactory.

The designer is cautioned to apply redundancy with discretion because of its impact on complexity, its space and weight requirements, and its economic import.

SECTION II NONINDEPENDENT SUSPENSIONS

9-3 GENERAL DISCUSSION

As stated in Section I, nonindependent suspension systems are characterized by rigid transverse structural members interconnecting at least two wheels. The simplest arrangement consists of a spring-mounted beam cross support connecting two wheels on opposite sides of the vehicle (Figure 9-1). Since this type of axle does not incorporate means for driving the wheels, it is called a *dead axle;* those which incorporate means for driving the wheels are called *live axles.* The term "live axle" applies to the entire axle assembly consisting of the housing containing the drive gears, differential system, and power transmitting shafts.

9-4 DEAD AXLE SUSPENSION SYSTEMS

9-4.1 SIMPLE BEAM TYPES

The simple beam type dead axle is most frequently used as a front axle for multi-axle vehicles or for nonpowered trailers. The dead front axle system supports the vehicle weight, supports torsional loading occurring during braking, supports the longitudinal, lateral, and vertical forces experienced during traveling, and maintains or restores wheel alignment under operating conditions. This type of axle is usually attached to the frame by means of relatively stiff semi-elliptic leaf springs which also perform the function of maintaining wheel alignment. These springs produce a harsh ride and limit the speed of a vehicle over irregular terrain. Lower rate springs may be employed, in which case the axle control and guidance functions must be accomplished by separate linkages or mechanisms.

To permit steering, the dead front axles are equipped with pivoting wheel spindles. The axles are usually I-sections of dropforged alloy steel. The unsprung mass may be reduced by using more expensive tubular axles of high alloy steel.

9-4.2 DeDION AXLE

Another type of dead axle suspension system is the DeDion type. The DeDion axle arrangement is also used with powered or driving wheels. The rigid connection between the two wheels is provided by a beam (usually a light tubular member) which passes either in front of, or to the rear of, the final drive (differential) case. The final drive case is mounted on the frame or body of the vehicle. The axle shafts between the wheels and the drive case have universal joints at each end.

The rigid member connecting the wheels is controlled by some form of linkage to maintain wheel alignment.

9-4.3 WHEEL SHIMMY

When one wheel of a solid axle suspension passes over an obstacle, the axle executes an angular movement in the vertical plane and both wheels are simultaneously angularly displaced by the same amount. Since the rotating wheels act as gyroscopes, a forced gyroscopic procession occurs which tends to make the steerable wheels swing about their kingpins. When the axle returns to the horizontal following a displacement, the gyroscopic forces are reversed, thereby causing the wheels to oscillate, or shimmy, about the kingpins. When the angular deflection of the axle occurs at a frequency near the natural frequency of torsional vibration of the vehicle body, the wheel shimmy is self-sustaining and can be eliminated only by a drastic reduction in vehicular speed. Another factor that tends to perpetuate the oscillation of the wheels is the self-aligning force produced by the caster angle of the kingpins.

Wheel shimmy resulting from terrain irregularity can be virtually eliminated by the use of independent front suspension. When one wheel of an independent suspension passes over an obstacle, both wheels will most likely be deflected but not simultaneously nor in the same amount. With certain types of independent suspension the plane of the wheel during vertical movement remains substantially parallel, thereby eliminating the gyroscopic effects which produce wheel shimmy.

9-4.4 AXLE TRAMP

Axle tramp is a vibration of the axle in a vertical plane. It occurs in rigid axle suspensions when negotiating irregular terrain and under certain conditions of braking. The mechanics of axle tramp related to irregular terrain and wheel shimmy are discussed in paragraph 9-4.3. Axle tramp occurs during braking when the braking effect of the two front wheels is unequal and is a result of employing leaf springs to maintain wheel and axle alignment. When braking hard, with the typical rigid axle suspension, the leaf springs wind up as a result of transmitting the braking reaction torque. As long as the braking of both wheels is equal, the windup of the springs is equal and the wheel alignment is unaffected. If, however, one wheel loses its braking adhesion, the spring on the same side unwinds and the axle rotates about a vertical axis so that it is no longer at a right angle to the longitudinal centerline of the vehicle. A gyroscopic precession is produced which lifts the opposite wheel, thereby increasing the load on the first wheel. The first wheel then recovers braking effort and the second loses braking effort, reversing the force conditions. The cycle continues and is self-sustaining until the brakes are released. This axle tramp can be eliminated in rigid axle suspensions by providing separate rigid guiding linkages that relieve the springs of braking torques. Axle tramp cannot occur in independent suspensions because the wheels are not rigidly interconnected.

9-5 LIVE AXLE SUSPENSION SYSTEMS

9-5.1 DESCRIPTION AND FUNCTION

The term "live axle" is used for a nonindependent integral axle assembly consisting of the rigid member connecting the wheels, the housing containing the drive gears and differential elements, and the power transmitting shafts.

Various types of springs have been used with live axle systems; these are discussed below.

9-5.2 MAJOR TYPES

9-5.2.1 Hotchkiss Drive

The Hotchkiss drive is one of the most commonly used live axle suspension systems used on wheeled military vehicles. In this system, a propeller shaft (drive shaft) is employed with two universal joints and a sliding splined joint as shown in Figure 9-1(D). The torque reactions, driving thrusts, and road induced forces are transmitted to the springs which must be rigid enough to maintain reasonable wheel and axle alignment under these conditions. For this reason, relatively stiff leaf springs are used. These springs are mounted longitudinally to the frame (or body) with one end pivoted and the other shackled, and the spring seats are fixed to the rigid beam which connects the wheels.

Because of the torque and force transmission requirements, the Hotchkiss drive has been criticized as providing inferior riding qualities, thereby, limiting vehicle mobility. However, this drive system is used extensively because of simplicity, low cost, and ruggedness. An additional advantage of the Hotchkiss drive is that the flexible connection between axle and frame acts as cushion for the drive train elements.

9-5.2.2 Torque Tube Drive

The torque tube drive, while not common on heavy military vehicles, is used on a number of passenger and light commercial vehicles. In this type of drive, the propeller shaft (drive shaft) is housed in a steel tube, the torque tube (Figure 9-1(C)). The rear end of the torque tube is bolted rigidly to the rear axle housing by means of a flange; its front end is connected to the transmission or a frame cross member by means of a special joint which transmits the driving thrust and torque, yet permits the required angular and longitudinal motions. A single universal joint is used at the forward end of the propeller shaft.

Normally, longitudinal leaf springs are used with the torque tube drive, and two radius rods connect the ends of the axle with the forward end of the torque tube. These elements maintain the axle housing alignment and support lateral loads. Since the springs are not required to transmit

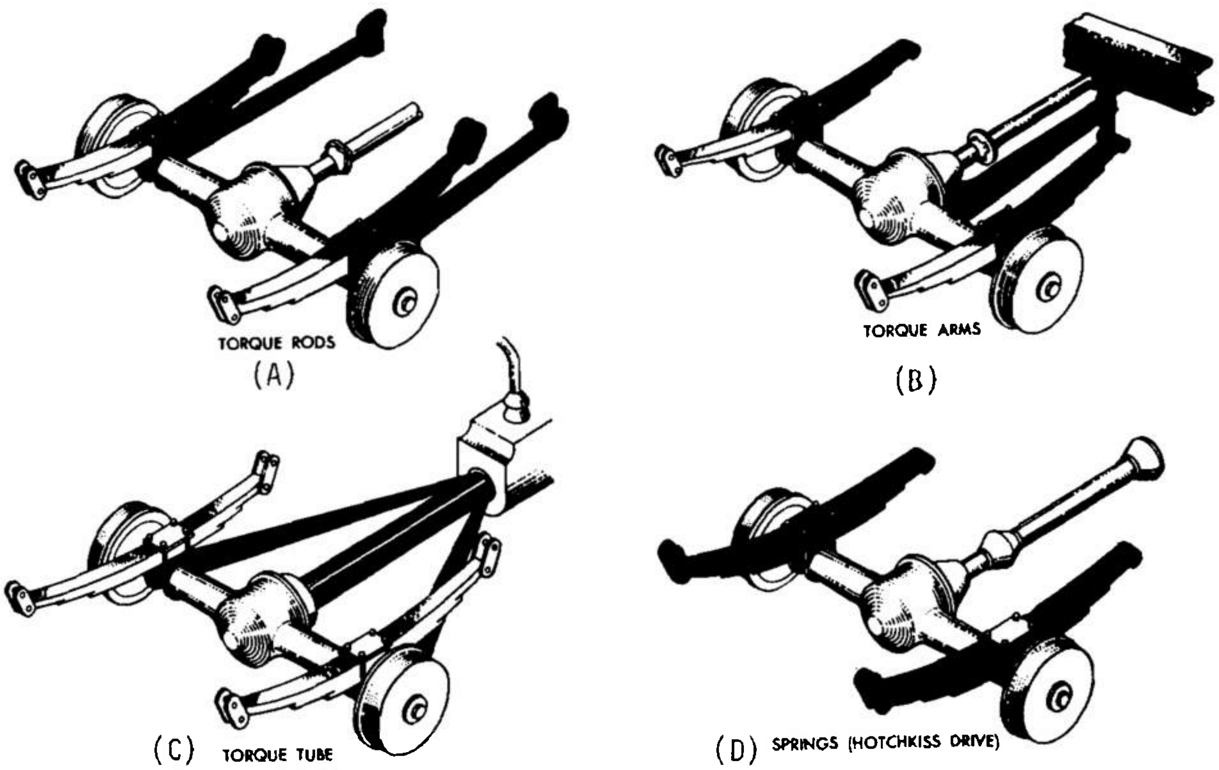


Figure 9-1. Typical Live Rear Axle Suspension Systems

driving or braking forces, they usually have a lower spring rate than those used in a similar vehicle having a Hotchkiss system.

9-5.2.3 Torque Arm Drive

The torque arm drive is seldom used on military vehicles. It consists of a solid or tubular arm, rigidly connected to the driving axle housing at its rear end and to a frame cross member at its front end through a ball-and-socket joint or spring bracket, as shown in Figure 9-1(B). An open propeller shaft is employed. The torque arm drive is similar to the torque tube drive, the principal difference being that the former employs an open propeller shaft running parallel to the torque arm instead of a drive shaft housed within a torque tube.

9-5.2.4 Radius Rod Drive

In the radius rod drive (Figure 9-1(A)), two radius or torque rods are used to transmit the driving thrust to the frame and to maintain the alignment of the driving axle assembly. The radius rods are connected to both the axle housing and to the frame by jointed connections which permit full vertical, and sometimes lateral, motion of the axle housing relative to the frame. The torque reactions of driving and braking are transmitted to the frame or body by means of the suspension springs, as in the Hotchkiss drive system. The radius rod drive usually employs an open propeller shaft with two universal joints. This type of drive is used to a very limited extent in military vehicles.

9-5.2.5 Bogie Suspensions for Wheeled Vehicles

Multi-wheeled suspension systems are employed on heavy military vehicles to maintain the wheel and tire loadings within safe limits. The most common arrangement to accommodate the additional wheels is the bogie suspension system. Bogies are generally used only at the rear of the vehicle; although some unusual vehicles, such as the Teracruzer, have also employed them at the front of the vehicle. The bogie suspension systems used on wheeled military vehicles are live axle systems, except when used on trailers.

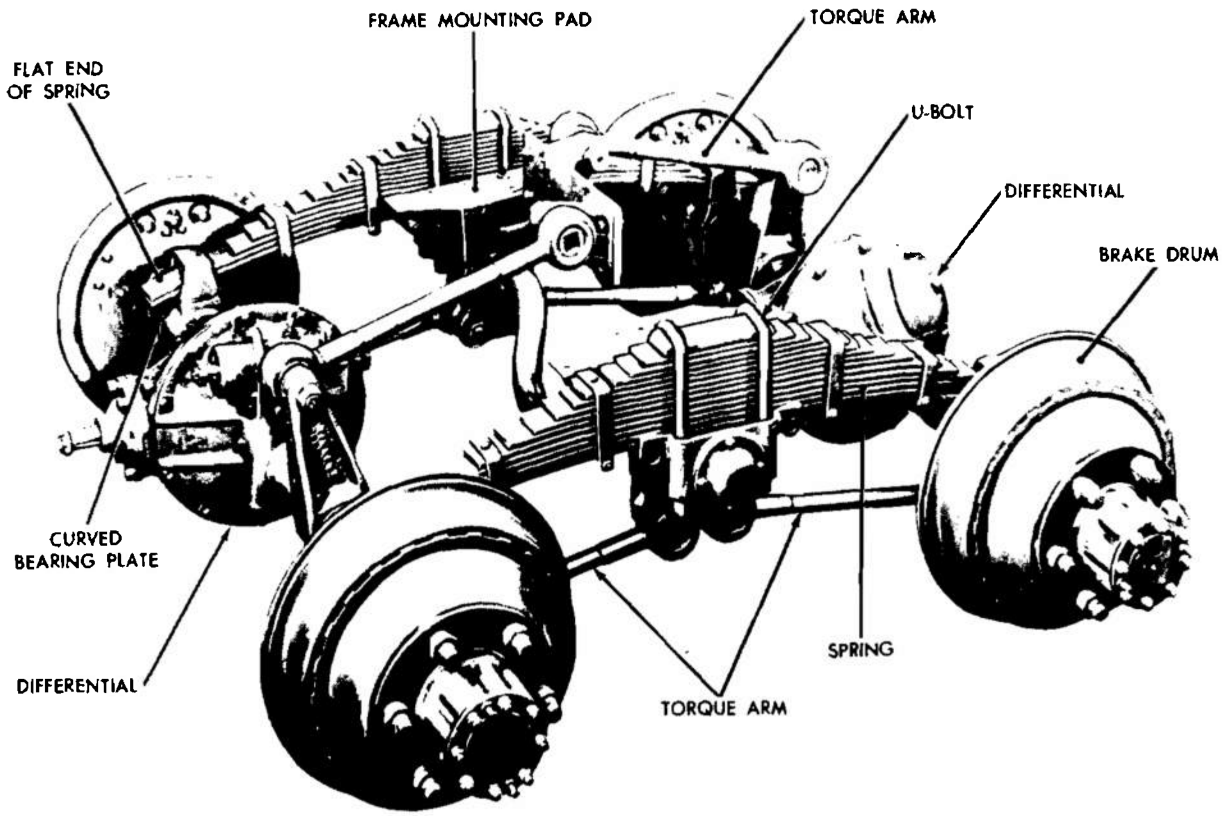


Figure 9-2. Bogie Suspension System for Wheeled Vehicle

The automotive bogie is a suspension assembly that enables tandem axles to function together as load-carrying and driving axles. The usual arrangement consists of tandem axles longitudinally interconnected by a pair of walking beams. These are joined, usually at their midpoints, by a single cross support (trunnion axle) which serves as the pivot point for the entire unit. Most frequently, leaf springs are employed as the primary flexible member because they can also serve as the walking beam. A typical military bogie is shown in Figure 9-2.

In the unit shown, the elastic members are leaf springs which are fastened at their midpoints to a spring seat which, in turn, is secured to the vehicle frame. The outer ends of the springs rest on hardened steel bearing plates on the tandem axle housings. Both spring seats are mounted on spindles at the ends of the trunnion axle. Tapered roller bearings are incorporated into the spindles to allow them to rotate freely despite side thrusts. Torque arms are employed to prevent the driving and braking torques from producing springs windup which would result in unequal axle loadings. These arms also maintain wheel alignment since the spring ends can float relative to the axle bearing plate. Because the springs (walking beams) are pivoted, they can distribute half of the load to each axle. As a result, the load is equally distributed over four wheels and, as such, permits full utilization of safe tire loadings.

The bogie suspension, as used on wheeled vehicles, has good obstacle-climbing and ditch-crossing capabilities because the pivoting-walking beam adjusts to terrain irregularities in such a manner that uniform ground pressure and full traction are maintained within design limits. The effect is somewhat similar to that produced by a single, larger diameter wheel.

Bogie suspension systems for wheeled vehicles can be designed with various amounts of displacement between the suspension components and the vehicle superstructure. Experience has established that a bogie design for wheeled vehicles should permit any one axle to rotate through an angle of 30° about the longitudinal axis of the vehicle without interference. Any wheel should be capable of moving to any position from 12 in. above to 6 in. below its normal level standing position (Ref. 1). Military characteristics for tactical vehicles may specify more severe requirements of this type. This will tend to increase the mobility of the vehicle as well as the cost.

Since the walking beam arrangement requires a substantial vertical clearance between the wheel and frame (or body) to accommodate the angular motions, high-rate springs are normally used to prevent excessive bottoming of the suspension. The effect of the stiff springs is moderated somewhat by the pivoting feature. At low and moderate speeds—when one wheel of a bogie suspension is deflected vertically by an obstacle—the spring pivots, deflecting both ends, thereby reducing the shock transmitted to the sprung mass. At high speeds, this effect is not always fully realized and the transmitted shock may be greater than that of an independently sprung wheel.

Consideration should be given in bogie design to minimize the resistance to changing course (steering). The tandem axle centers should be as close together as tire diameters plus reasonable clearances permit. Some bogie suspensions provide a small amount of free play of the axles so that they can adjust during course changing to reduce the steering resistance. In bogies, where the semi-elliptic leaf springs are attached to the axles, a natural steering effect is produced (roll steer) by the spring deflections caused by the centrifugal forces acting on the sprung mass during cornering (Ref. 1).

SECTION III INDEPENDENT SUSPENSION SYSTEMS

9-6 GENERAL DISCUSSION

The term "independent suspension" is applied to vehicle elastic support systems in which each wheel supports part of the total vehicle load with no intermediary connection between the wheels. Each wheel is free to oscillate independently of the other wheels. For wheeled vehicles, either the front or rear wheels, or both, may be independently suspended. They may be driven or free wheeling.

9-7 FREE WHEELING TYPE

A typical independent front suspension for a light vehicle is shown in Figure 9-3. In this design, each wheel is held in alignment by a pair of rigid control arms (upper and lower wishbones). The vehicle weight is transferred from the frame to the rigidly attached cross member, to the coil spring and the lower wishbone, and finally, to the wheel.

For the system shown, the shock absorbers are integrated with the upper control arms. Frequently, direct-acting hydraulic shock absorbers are used. With this type of unit, the one end of the damper is pivotally fastened to the lower control arm while the opposite end is pivotally mounted to the frame or body of the vehicle.

9-7.1 ROLL AXIS AND MOMENT

The sprung mass of a vehicle will rotate (or tend to rotate) about some longitudinal axis if a pure couple or moment is generated in a lateral plane of the vehicle. This axis is the roll axis of the suspension system. For a conventional wheeled vehicle, such as a four-wheeled tactical vehicle, there will be separate roll centers (points on the roll axis) for the front and rear suspension systems. The location of a specific roll center is determined by the configuration and geometry of the suspension system. The subject of roll center and roll axis location for wheeled vehicles is treated comprehensively in Refs. 6 and 7. The basic method of determining the roll center for a given system will be presented. Figure 9-4 shows a diagram of an independent suspension system having converging upper and lower control arms (wishbones or A-frames). For the system shown, the position of the roll center is determined by considering the

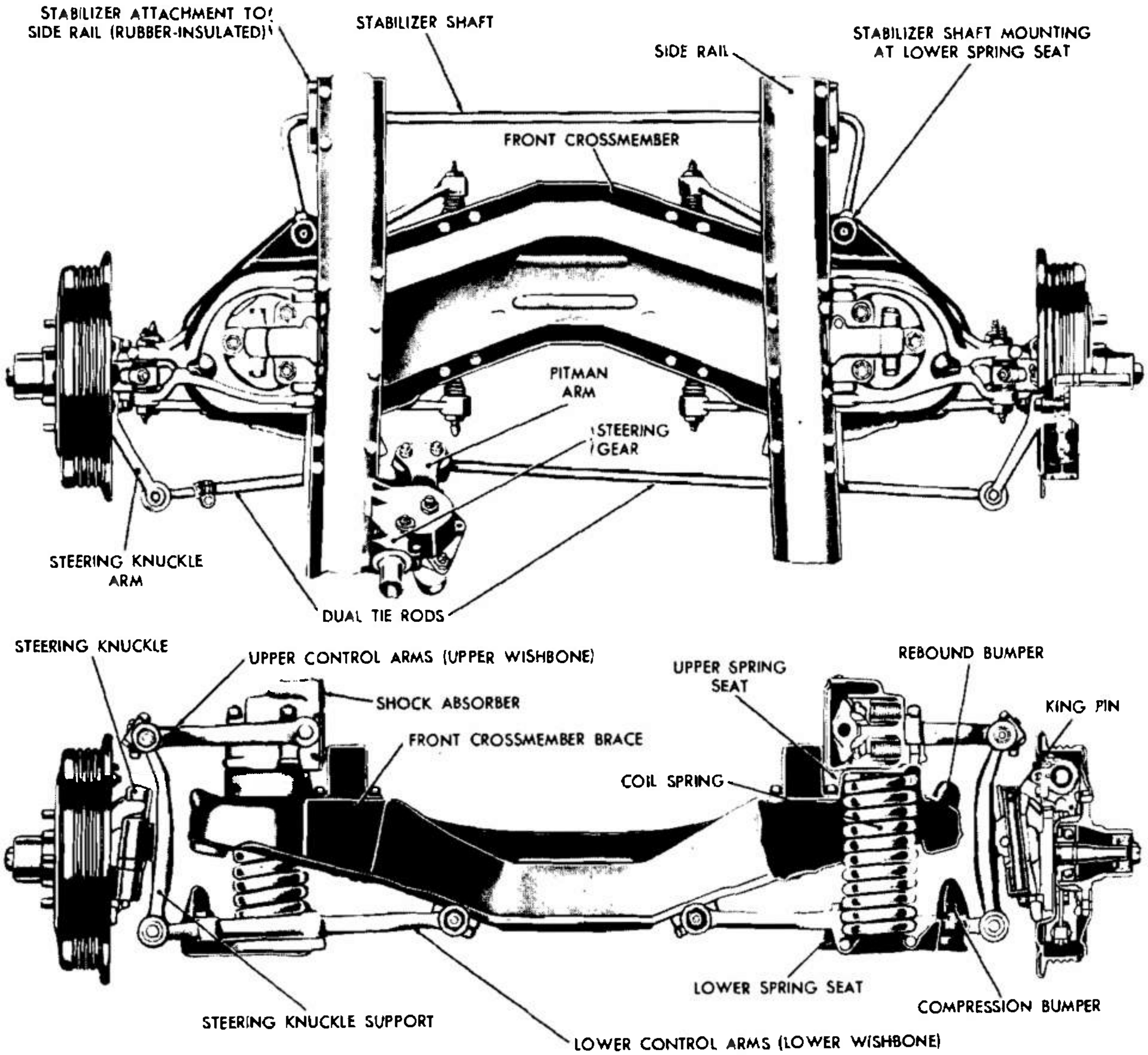


Figure 9-3. Typical Independent Front Suspension System for Wheeled Vehicle

instantaneous centers of the total system during rotation of the sprung mass relative to the ground.

The instantaneous center of points 1 and 2 relative to the sprung mass M is found by extending the lines connecting points $1 - 4$ and $2 - 5$ until they intersect at 0. Point 3, at the point of tire contact, may be considered to be a point on the ground and on the tire; hence, it is an instantaneous center in the wheel-ground-sprung mass system. The three instantaneous centers of this system must be on the extended line $0 - 3$. If we assume a symmetrical suspension, the roll center M_o will be located in the central longitudinal plane of the vehicle.

It is obvious that the vertical position of the roll center can be changed by changing the angle between the control arms. For example, if the arms are parallel to each other and to the ground, the

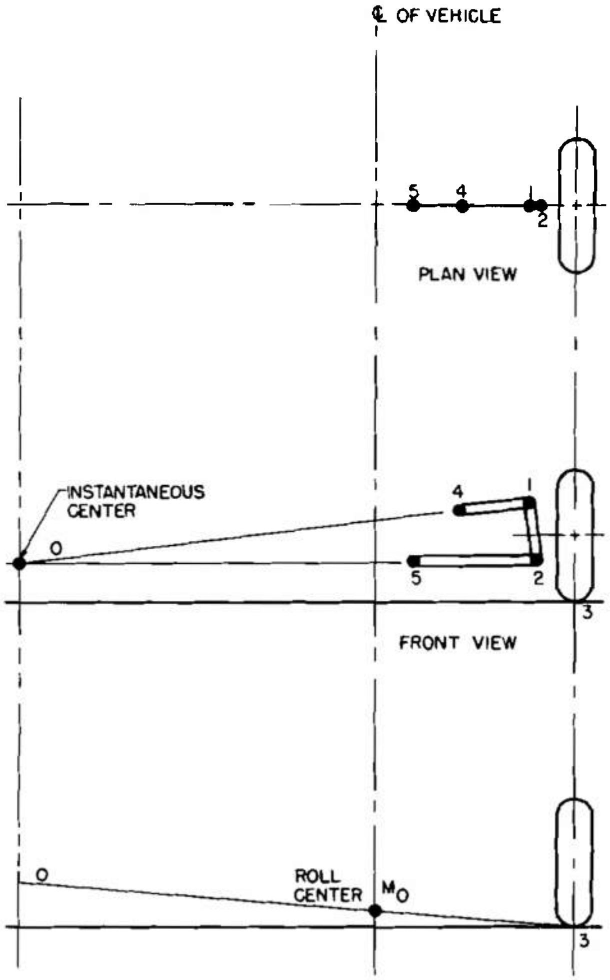


Figure 9-4. Graphical Determination of Roll Center

instantaneous center 0 of the links lies at infinity, and the roll center M_o of the sprung mass is at ground level.

The location of the roll axis for a given vehicle is an important factor with respect to the dynamic stability of the vehicle since the behavior of a vehicle as it travels along a curved path is influenced by the changes in tire loading and attitude resulting from roll.

9-7.2 WHEEL CONTROL

The kinetic behavior of the wheels of a vehicle during vertical displacement relative to the body is influenced by the type of suspension system. For example, neglecting the effects of tire deflection, the top of a wheel controlled by the double transverse link suspension arrangement (Figure 9-3) will move away from the center of curvature as a vehicle rolls during a turning maneuver. These changes in the camber angles of the wheels reduce the cornering ability of the tires.

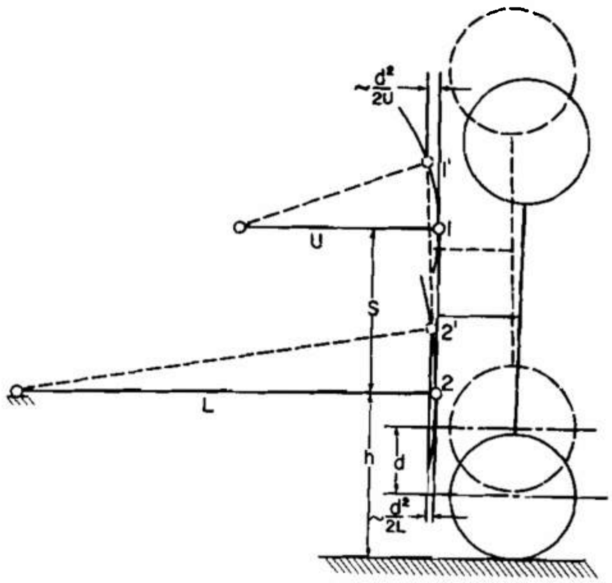


Figure 9-5. Geometry of Linkage of Independent Sprung Wheel

The vertical displacement of a wheel may generate gyroscopic or inertia torques, depending on the particular arrangement of the control members that determine the path of the wheel. In addition, changes in camber and/or track during vertical movement may be inherent for particular suspension arrangements. For the typical front wheel independent suspension shown in Figure 9-3, it is usually desirable to minimize the amount of lateral movement (scrub) of the tire at the ground when the wheel moves upward from its normal loaded position. For a system composed of parallel, unequal length wishbones, this can be accomplished by choosing the proper ratio for the lengths of the upper and lower control links.

Figure 9-5 is a diagram of the kinematics of the system under consideration (Ref. 8). For a vertical displacement d of the wheel, points 1 and 2 move to 1′ and 2′, respectively. For small values of d, the lateral displacements of points 1 and 2 are approximately $d^2/2U$ and $d^2/2L$. A simple geomet-

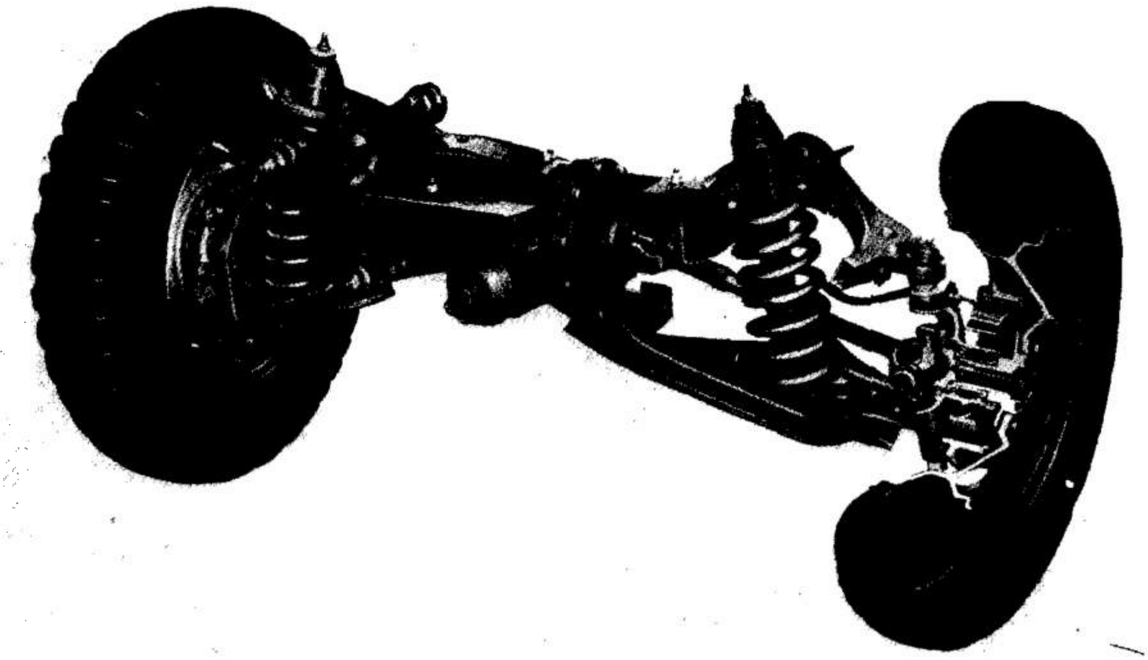


Figure 9-6. Typical Driven Independent Suspension System for Wheeled Vehicle

ric construction reveals, for the conditions of no lateral movement of the tire at the ground, the ratio of:

$$\frac{U}{L} = \frac{h}{h + S} \tag{9-1}$$

Similar kinematic studies and additional kinetic investigations should be made for any proposed independent suspension layouts to incorporate the following desirable characteristics into the system (a) minimum gyroscopic or inertia torques with vertical displacement, (b) minimum disturbance of the steering geometry with vertical displacement, and (c) favorable changes in wheel attitude, e.g., camber, during deflection caused by roll.

9-8 DRIVEN TYPES

Driven, or powered, independent suspension systems can be utilized in the front or rear of wheeled vehicles. In the former case, a steering system must be incorporated into the total assembly in addition to the power transmission system. A driven, steerable independent suspension utilizing pivoted control links is shown in Figure 9-6. Typical of powered independent suspension systems is the location of the gear or differential case shown in Figure 9-6. The power is transmitted to each wheel through half-shafts from the gear case which is attached to the sprung mass of the vehicle. Since the gear box is fastened to the vehicle body or frame, the driving torque reactions do not affect the suspension springs as they do with the powered rigid (nonindependent) axle suspension system.

Another important characteristic of the driven independent suspension system is the reduction in unsprung weight with respect to a comparable rigid axle system. Figures 9-7 and 9-8 show other arrangements of driven independent suspension systems.

9-9 FUNCTIONAL CHARACTERISTICS AND OTHER ATTRIBUTES

The development of independent suspension systems for wheeled vehicles was motivated primarily to reduce the unsprung mass in civilian vehicles, thereby improving the handling and traction characteristics under high-speed operation. However, the demand for greater speed and mobility for military vehicles warrants the use of independent suspensions on such vehicles. In addition to the improved performance associated

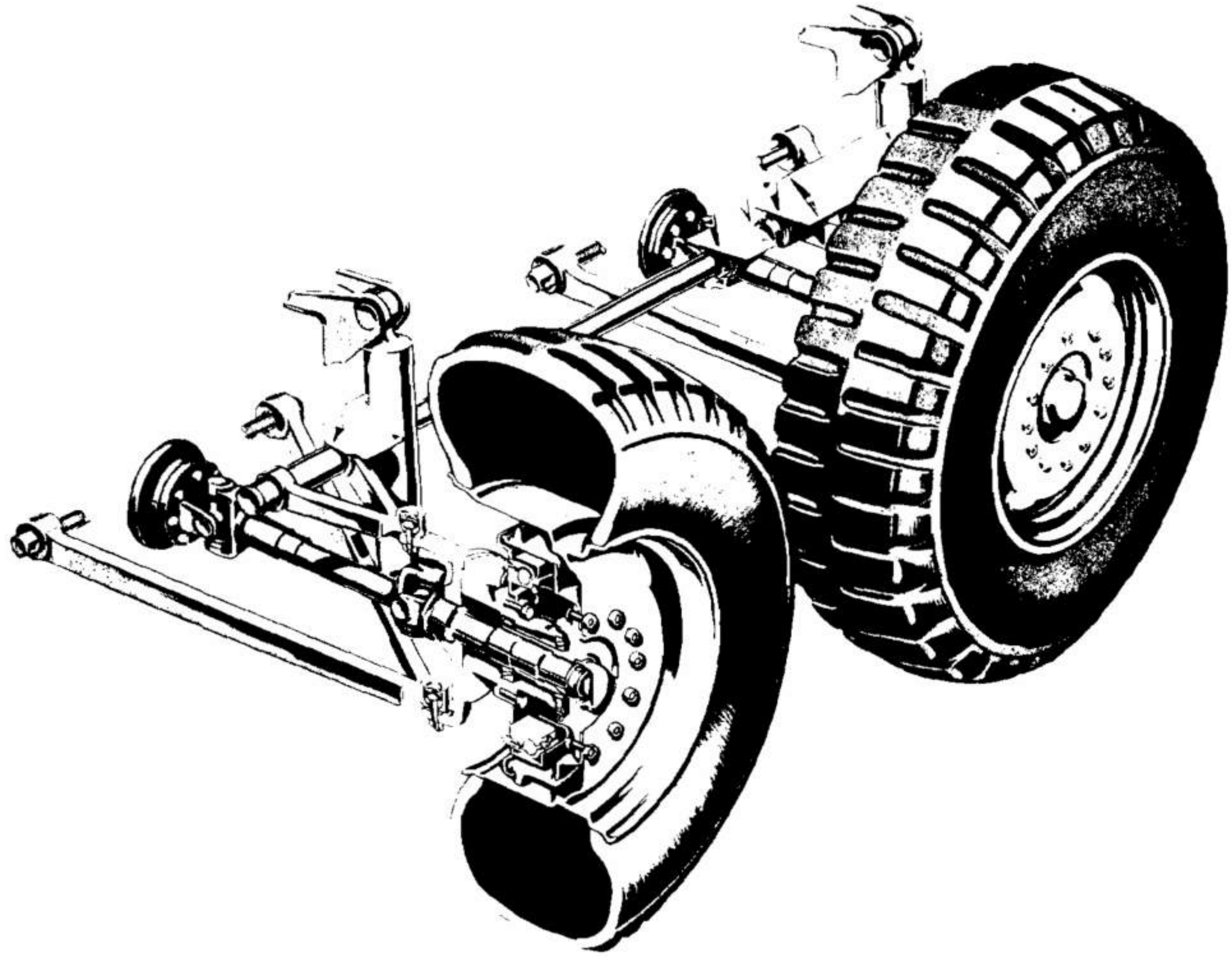


Figure 9-7. Typical Driven Independent Suspension System for a Heavy Multi-Wheeled Vehicle

with reduced unsprung mass, the use of independent suspensions in place of nonindependent systems increases the speed and mobility of the military vehicle by

(a) Reducing vibrations such as wheel shimmy and axle tramp
(b) Permitting the use of lower rate springs in many cases
(c) Providing more ground clearance
(d) Permitting improved wheel spacing

The means by which these advantages are affected are discussed in the paragraphs which follow.

9-9.1 REDUCTION OF UNSPRUNG MASS

When one of the wheels of a conventional nonindependent suspension system experiences an impact, the entire suspension assembly is affected. Under these circumstances, the large unsprung mass is responsible for the development of high forces on components such as tires, wheels, and bearings. The ride quality is adversely affected by the subsequent wheel and axle disturbances, thereby limiting the speed at which the operator will operate the vehicle. Equally high forces are experienced by the terrain which, under marginal conditions, may fail and impair vehicle mobility.

The wheel suspension system, utilizing pneu-

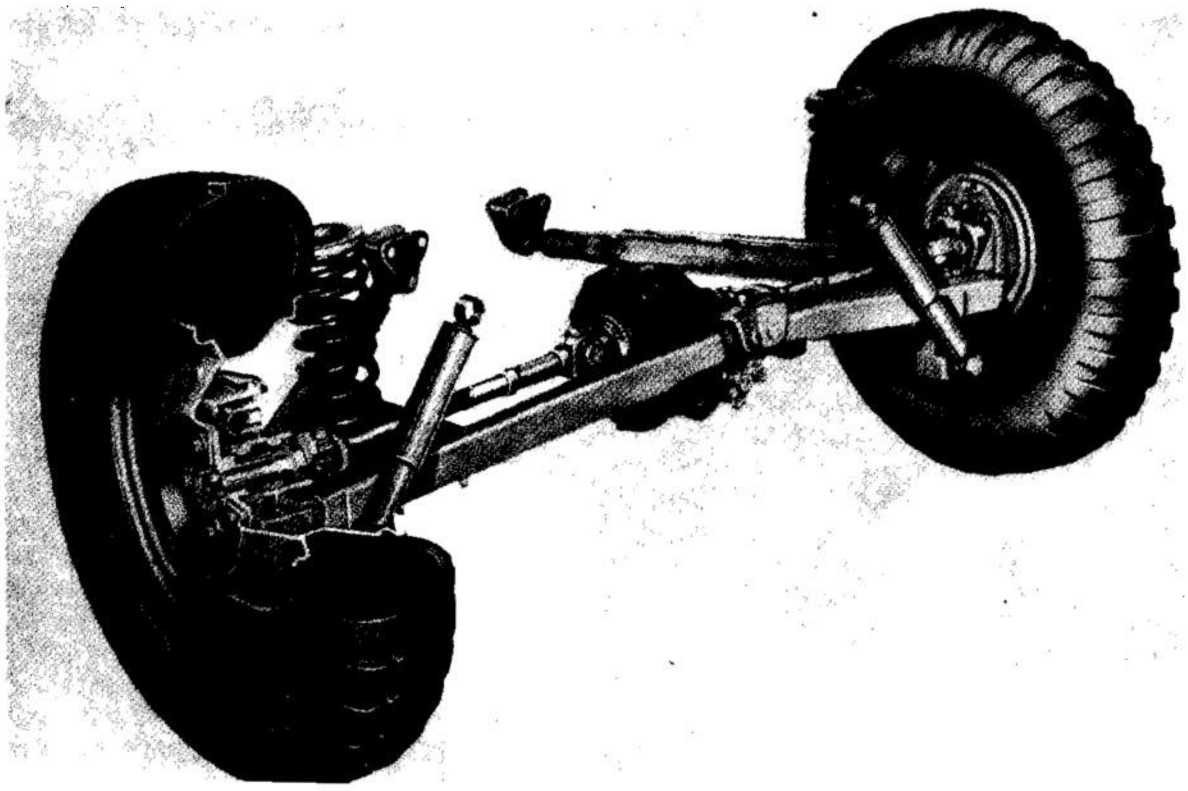


Figure 9-8. Swing Arm Independent Suspension System

matic tires, can be considered a two-degree-of-freedom spring-mass system consisting of the sprung mass, the primary elastic medium (spring), the unsprung mass, and the elastic tire. When the tire maintains contact with the ground, the two masses have separate and different natural frequencies of oscillation. When the tire leaves the ground, the unsprung mass responds to the action of the suspension spring at a natural frequency different from that occurring during ground contact. It is this latter frequency that determines the maximum speed at which the vehicle can negotiate irregular terrain and still maintain continuous tire contact. It can be demonstrated that a smaller unsprung mass will maintain continuous ground contact over irregular terrain at higher vehicular speeds than will larger masses with the same basic system.

From the fundamental principles of vibration, the following equations can be written. if the effects of damping are neglected.

$$T_m = 2\pi\sqrt{\frac{m}{k}} \tag{9-2}$$

$$T_M = 2\pi\sqrt{\frac{M}{k}} \tag{9-3}$$

where

T_m = period (sec) of free vibration of the unsprung mass, M, (lb-sec^2/in.)

T_M = period (sec) of free vibration of the sprung bass, m, (lb-sec^2/in.)

k = spring rate, lb/in.

Then

$$T_m = T_M\sqrt{\frac{m}{M}} \tag{9-4}$$

As shown by Equation 9-4, reducing the unsprung mass with no change in the sprung mass reduces the period T_m. The shorter period permits the unsprung mass to follow a particular ground wave or terrain irregularity at a higher vehicle speed. Continuous ground contact of the wheels of a vehicle is essential to provide directional stability, steering response, and traction.

9-9.2 REDUCTION OF WHEEL SHIMMY AND AXLE TRAMP

The phenomena of wheel shimmy and axle tramp are discussed in paragraphs 9-4.3 and 9-4.4.

Wheel shimmy has been eliminated by the use of independent suspension systems and axle tramp does not occur in independent suspensions because the wheels are not rigidly interconnected.

9-9.3 SPRING CHARACTERISTICS

Most of the nonindependent suspension systems in current use on military wheeled vehicles employ rigid axles and leaf springs. Since the leaf springs act as wheel control links as well as elastic members, they are usually relatively stiff. Independent suspension systems usually provide separate rigid members to maintain wheel alignment, and to transmit driving and braking torques to the main mass of the vehicle. This permits the best compromise with respect to the selection of front and rear suspension natural frequencies. For example, the tendency to pitch of a four-wheeled vehicle can be minimized if the elastic suport systems are chosen such that the front suspension frequency is somewhat less than the rear suspension frequency.

9-9.4 IMPROVED GROUND CLEARANCE

Another advantage of independent suspension systems for wheeled vehicles is the increased ground clearance. With the conventional rigid driven axle suspension, the gearbox is located at the wheel center height; and the gearbox normally extends below the axle center line. Furthermore, substantial clearance is provided above the gearbox. With the independent suspension, the final drive gear case is attached to the frame (or body in unitized vehicles), thereby providing substantially greater ground clearance than that of a comparable nonindependent system.

9-9.5 WHEEL SPACING

Many of the current military wheeled vehicles utilize the bogie suspension shown in Figure 9-2. The bogie configuration dictates the wheel spacing which may not be the optimum with respect to mobility. The use of independently suspended wheels allows greater flexibility in locating the wheels along the length of the vehicle.

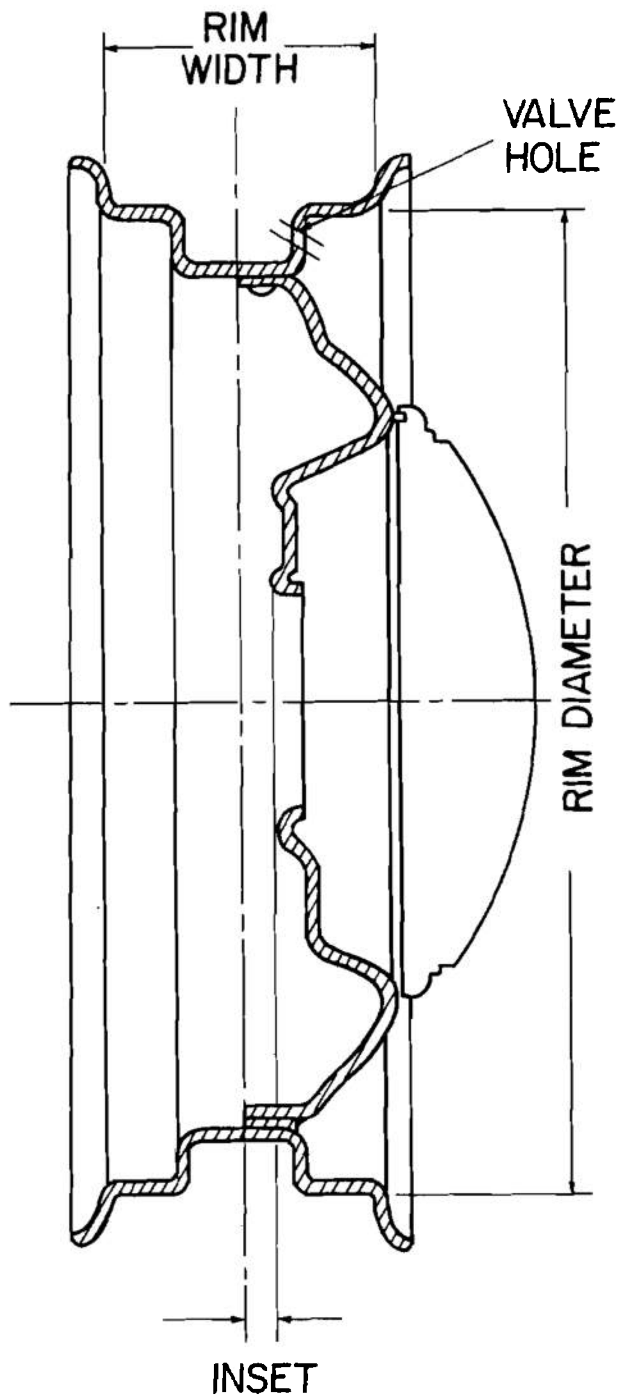


Figure 9-9. Light Disc-Type Wheel

9-10 DISADVANTAGES OF INDEPENDENT SUSPENSION SYSTEMS

The disadvantages of independent suspension systems for military vehicles are primarily those of cost and maintenance. Compared to the solid axle leaf-spring suspensions, the independent systems require increased design effort, somewhat more expensive machining, and increased service to maintain proper wheel alignment. Continuing development of the independent systems for military wheeled vehicles will minimize these disadvantages.

SECTION IV WHEELS AND TIRES

9-11 PURPOSE AND FUNCTION

The primary purpose of wheels on military vehicles is to minimize resistance to vehicle motion while supporting the vehicles on the terrain. Although wheels possess certain shortcomings and present many problems to the military vehicle, they are the lightest, simplest, and most extensively employed method of accomplishing the above function.

In addition to the primary function, wheels and tires are required to accomplish the following secondary functions:

(a) Transmit the drive and braking torques between the vehicle and the terrain
(b) Provide a means of steering the vehicle
(c) Reduce the road shocks transmitted to the rest of the suspension system
(d) Provide traction for the vehicle
(e) Reduce the effects of terrain irregularities
(f) Provide lateral stability for the vehicle.

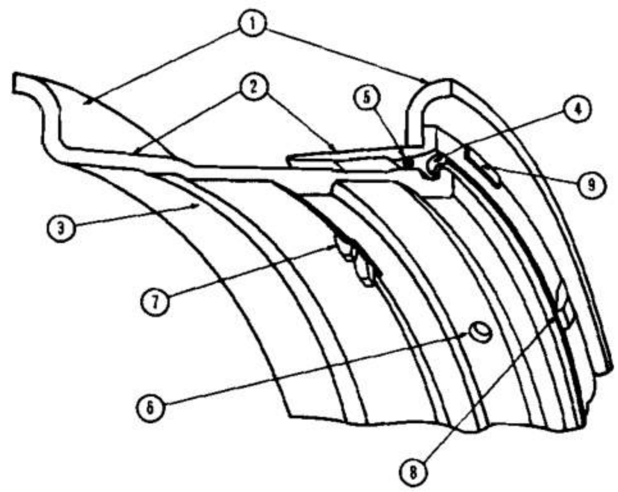
9-12 CONSTRUCTION OF WHEELS FOR MILITARY VEHICLES

9-12.1 GENERAL DESCRIPTION

The wheels in general use for light military vehicles are pressed steel disc types. These are bolted, detachable units which are mounted on the brake drum-hub assemblies. In some cases, for heavier wheeled vehicles, the spokes are cast integral with the hub and a demountable rim is used.

9-12.2 COMPONENT PARTS

The basic components of a disc-type wheel are shown in Figure 9-9. In addition, two of the basic dimensions (rim width and rim diameter) are indicated. The rim is a highly developed component which may be one piece for the lighter vehicles but is more likely to consist of several component parts. A typical heavy duty rim assembly is shown in Figure 9-10. The *rim base* has an intergral inner rim flange and inner bead seat. The outer *bead seat band* and the outer *rim flange* are installed after the tire is mounted on the rim base as is the *lock ring*. The basic standard



LEGEND

1. SIDE FLANGES
2. TAPERED BEAD SEATS
3. RIM BASE
4. LOCK RING
5. SEALING RING
6. VALVE HOLE
7. LOCATING LUGS
8. BEAD SEAT BAND DRIVER
9. PRY BAR POCKETS

Figure 9-10. Typical Large, Heavy Duty Rim Assembly for Tubeless Tire

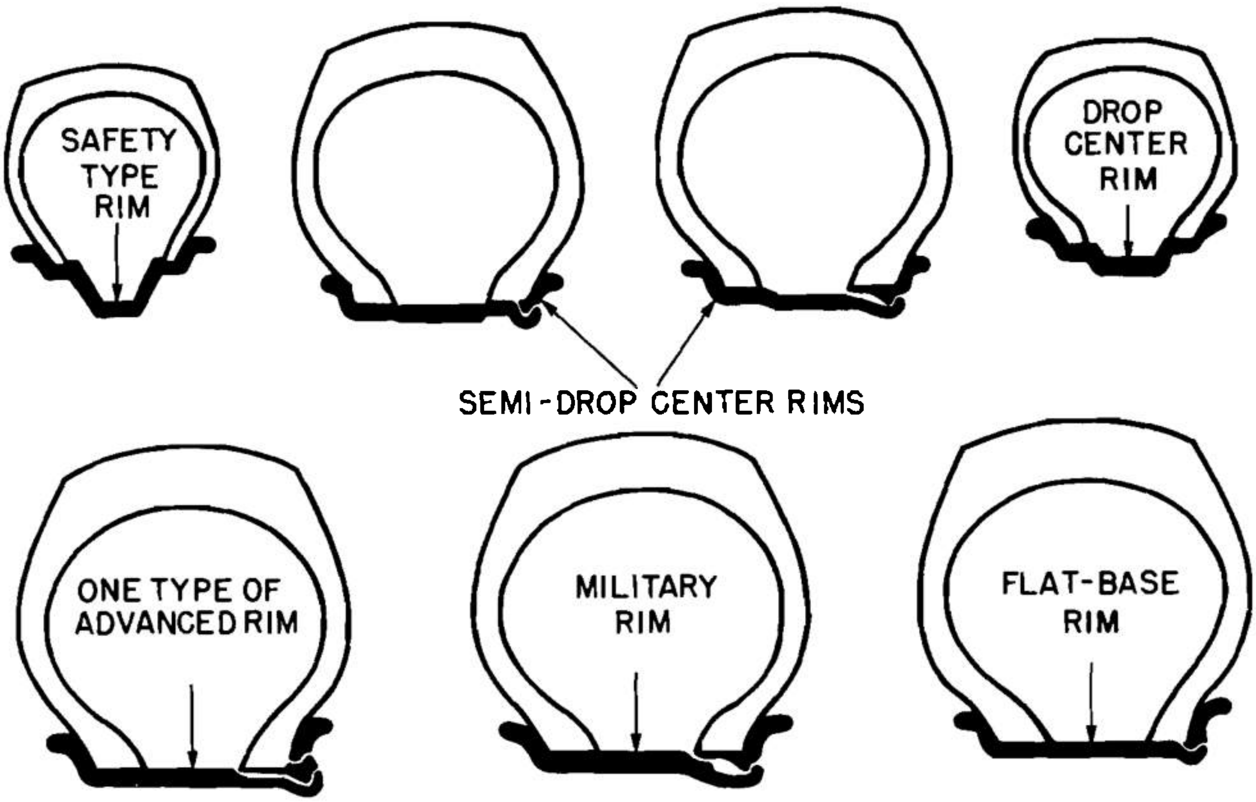


Figure 9-11. Typical Rim Outlines

wheel rims are discussed in the paragraphs which follow; however, the discussion is for general information only. For reference purposes, the latest Tire and Rim Association Inc. Year Books and manufacturers' current data books should be used.

9-13 STANDARD WHEEL RIMS (Ref. 9)

9-13.1 DROP CENTER RIMS

The drop center rim, as shown in Figure 9-11, is made in one piece and is permanently fastened to a disc, thus, forming a wheel. The distinctive feature of this rim is the center well or depression which permits mounting and demounting the tire. The bead seats of the rim have a 5 deg taper to match a corresponding taper on the beads of the tire. The taper is such that, as the tire is inflated, a wedging fit results. Drop center rims are generally used on light vehicles such as passenger vehicles and ¼-ton trucks.

9-13.2 SAFETY DROP CENTER RIMS

Safety drop center rims are similar to standard drop center rims with the exception of having a slight hump at the edge of the bead seat ledges which tend to hold the beads in place when the tire is deflated.

9-13.3 SEMI-DROP CENTER RIMS

The semi-drop center rim (Figure 9-11), is also permanently fastened to the wheel disc. It has a shallow well, tapered bead seats, and a demountable flange which fits into a groove or gutter on the outside edge of the rim and hold the tire in place.

9-13.4 FLAT-BASE RIMS

The flat-base rims are obsolete; however, a large number were used on World War II vehicles. It has a flat seat for the beads of the tire which must be correspondingly flat. There is no drop center or

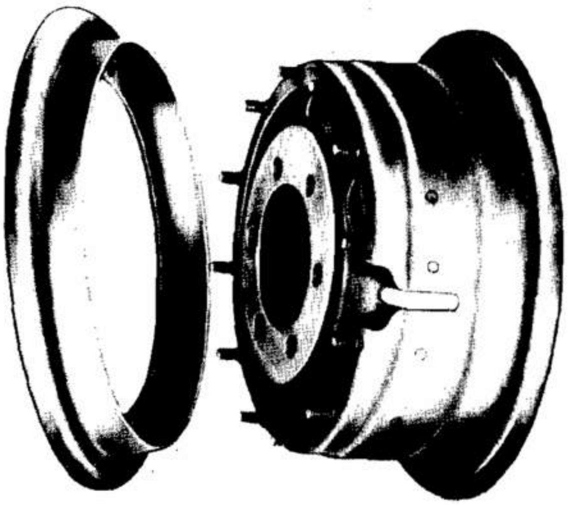


Figure 9-12. Divided Rim for Heavy Service

well to the rim, a detachable side flange permits mounting and demounting the tire.

9-13.5 MILITARY RIMS

9-13.5.1 Military Standard

The military rim (Figure 9-11) is similar to the two-piece flat-base rim, except that it has a 5 deg taper on each bead seat to prevent slippage. The military standard rim permits complete interchangeability among the rim bases and side rings (for given sizes) supplied by different manufacturers. The types of rims referred to as *advanced rims* are of this general configuration, except that they differ in the design of their rim base gutter areas among the various manufacturers. This prevents tire interchangeability.

9-13.5.2 Divided Rims

Wheels for heavy vehicles under the severe service encountered in tactical conditions are of the divided rim type as shown in Figure 9-12. In this type, the two parts are held together by bolts or studs and nuts. Divided rims are designed for use with combat tires, but standard military tires can also be used. With the divided rim, the tire is held in position by a hinged or a segmental beadlock (see paragraph 9-13.5.3).

9-13.5.3 Beadlocks

A beadlock is a metal device that fits between the beads of the tire so that pressure can be applied by tightening the rim flanges against the outside of the bead. The beadlock is slightly wider than the space between the tire beads when mounted on the rim. Thus, a compression fit is obtained that locks the beads into place so they will not slip on the rim and will hold the tire in position. In combat, this is necessary in order to offer support when pressure is lost.

9-14 COMBAT WHEELS

The typical combat wheel in current military service is the divided rim type discussed in paragraph 9-13.5.2. Although most of these are of steel disc construction, the demand for lightweight vehicles has lead to the development of wheels made of lightweight materials.

9-15 LIGHTWEIGHT WHEELS (Refs. 10-12)

Research programs have shown that wheels constructed of aluminum or magnesium alloy and of glass fiber reinforced plastics have high potential for military applications.

The major advantage gained by the use of these materials is the weight saving which can approach 60 percent based on standard steel wheels.

Experimental wheels of cast aluminum alloy, cast magnesium alloy, and forged magnesium alloy have been developed. A detailed account of a development and testing procedure for experimental wheels is presented in Ref. 12 and is summarized in Chapter 10.

Plastic wheels using glass-fiber reinforced epoxy and polyester resins have been developed for both wheeled and tracked vehicles. These units indicate that weight savings of 62.5 percent with no sacrifice in strength or durability is achievable.

9-16 UNUSUAL WHEELS (Ref. 13)

In order to achieve some function normally provided by other parts of the suspension system or to eliminate the pneumatic tire, unusual wheel designs have been tested.

In one such design, two rubber discs replaced part of the metal wheel disc. This was done in an attempt to improve the ride quality of a vehicle

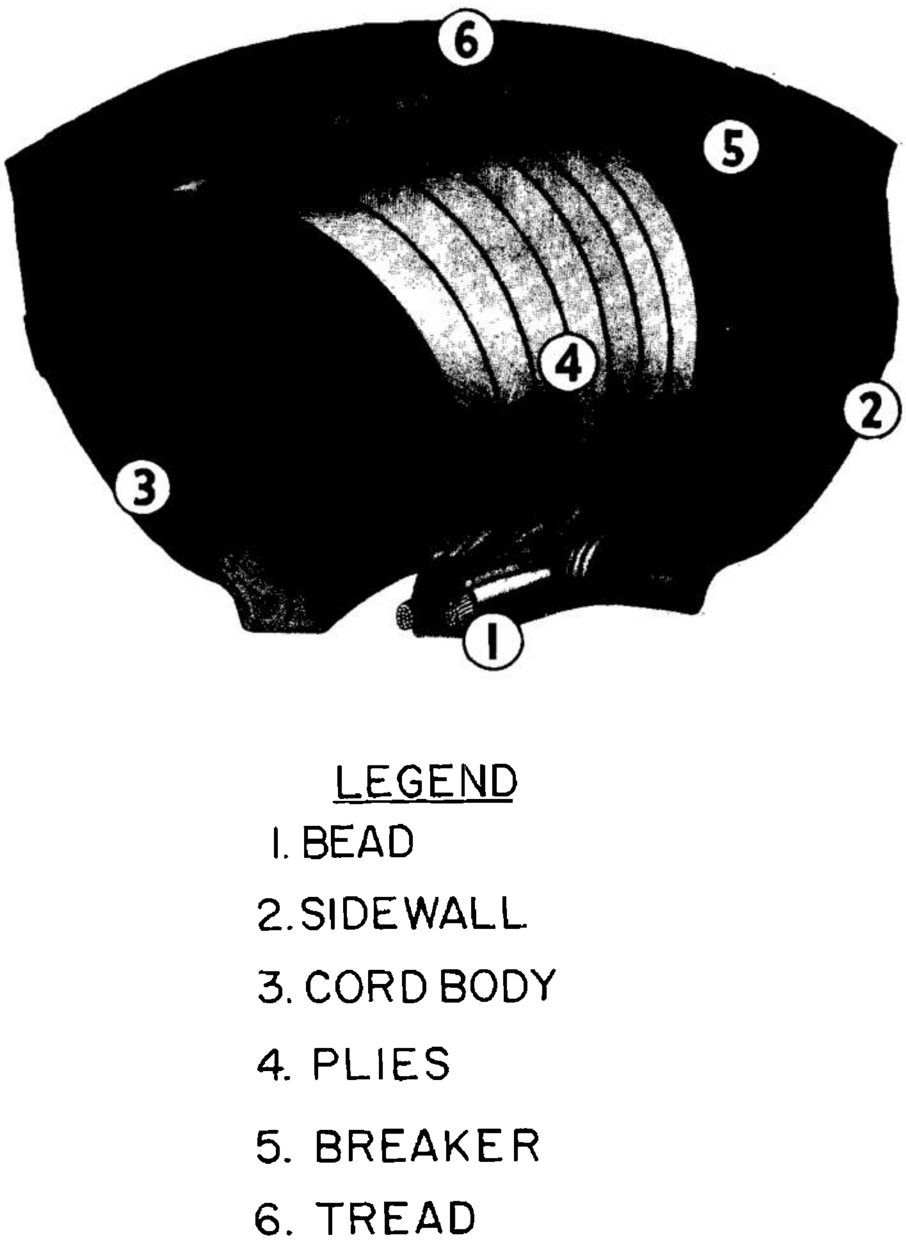


Figure 9-13. Pneumatic Tire Construction

that has no separate suspension springs and depends solely on the tires to absorb the road shocks. While this design did permit slightly higher speeds over the rough terrain, lateral stability, steering, and durability were poor. Another unusual wheel was the Martin wheel which consisted of rubber spokes connected to a hickory-reinforced non-pneumatic rubber tire. Its purpose was to eliminate some of the shortcomings and vulnerability of the pneumatic tire. This wheel, while comparing favorably with a pneumatic tired wheel in some respects, proved totally inadequate to cope with severe road impacts. The lateral stability, soft-soil performance, and obstacle-climbing ability were inferior to the pneumatic-tired wheel.

The elastomer foam-filled tire has been proposed many times and experimental tires based on this concept have been built and tested. The relatively large hysteresis of these materials and the resulting heating have been the major problems preventing the development of a satisfactory foam-filled tire.

9-17 TIRES

9-17.1 INTRODUCTION

Current military wheeled vehicles are equipped exclusively with pneumatic tires. Various sizes and types of tires have been selected and standardized for military service. These standardized units depend not only on the size and type of vehicle but also on the intended operational service.

9-17.2 TIRE STANDARDS (Ref. 14)

The tire manufacturers have standardized tire and tube sizes and types. The Tire and Rim Association has established additional standards for military tires and tubes. These standards are given in MS-35388 through MS-35342, found in Ref. 14.

9-18 TIRE CONSTRUCTION AND COMPONENT PARTS (Ref. 15)

The structural parts of a pneumatic tire are the tread, sidewalls, breakers, cushion, plies, and bead. These component parts are assembled to support each other and obtain the end for which the tire was designed. While practically all types of modern pneumatic tires contain these general components, each type of tire is built for a specific service and requires the design of its parts for the performance of its specific duty. For example, the angle at which the cords of the piles repose influences the behavior of the tire under lateral loading; also, the thickness and compound of the treads influences the performance and wearing behavior of the tire.

Each part of the tire (Figure 9-13) serves a definite purpose as described in the paragraphs that follow.

9-18.1 TREAD

The tread is the layer of rubber on the outside circumference of the tire and is the part which contacts the terrain. It must provide traction and resist wear. The tread also protects the plies from cuts, bruises, and moisture. Rubber extends down over the side wall for protection.

9-18.2 SIDEWALLS

The sidewalls are a protective covering of flexible rubber compound on the sides of the tire. They are designed to flex and bend without failing when subjected to the normal continuous deflection or extreme, sudden shock.

9-18.3 BREAKER AND CUSHION

The breakers is located between the tread and the main cord body. It insures union between these parts and protects the cord body from road shocks. Heavy layers of resilient, but tough, rubber compounds completely surround the strong cords of the breaker strips.

The cushion is a layer of soft, heat-resisting rubber which bonds the plies and breaker together.

9-18.4 PLIES (CORD BODY)

The plies are the successive layers forming the cord body. Each cord in the ply is completely surrounded by rubber compound and each ply is protected from the next by a cushion.

The number of cords per inch, as well as their diameter and the angle at which they lay with respect to the tire center line, are the fundamental factors with respect to ply strength. The number and composition of the plies determine the ultimate strength of the tire.

9-18.5 BEAD

The tire beads anchor the tire to the rim. Each bead consists of two or more cables of steel wire. The cables are impregnated with a hard rubber compound and all plies are tied-in to them preventing any change of shape or fit of the tire.

9-19 TUBELESS TIRES

Although tubless tires are used extensively for civilian passenger automobiles, and have recently been extended to normal truck sizes and earthmover applications, they are not used for military vehicles at present.

The tubeless tire eleminates the tube, replacing it with either an impervious liner that covers the whole inside surface of the tire or with an impervious treatment of the plies themselves. The rim must be air tight and must be smooth in the bead seat areas to affect a seal with the tire. The valve is installed in the rim of the wheel.

The major advantages of tubless tires with respect to the tubed tire are: (a) a reduction in the heat generated, (b) greater tire flexibility, (c) greater tolerance to small object punctures, (d) a vastly greater tolerance to slippage on the rim, and (e) reduced weight.

The major disadvantages are: (a) the need for better rim finish and condition, (b) less tolerance to severe side loading without pressure loss, (c) field repair problems, and (d) the need for sealing rings in rims having more than one part.

9-20 VERY LOW PRESSURE TIRES (Ref. 16)

Very low pressure tires, called rolligons or terra tires, have been recently introduced for use on military vehicles as an attempt to improve traction, flotation, and obstacle performance. These tires have a high width-to-diameter ratio and an operating pressure of 3 to 16 psi. The elastic and energy-absorbing properties of these tires are such that no suspension spring system is used. Early field tests of vehicles using these very low pressure tires have indicated that under conditions of small rock-strewn terrain or plowed ground, their performance was particularly successful but was deficient under other off-the-road conditions compared to other wheeled vehicles.

Land locomotion studies indicate that a narrow, low-pressure tire is superior to a wide tire of the same diameter, with respect to cross-country mobility (Ref. 1). An application of tires in this category is found in the Goer family of military vehicles which employs large diameter tires of the type used on earth-moving equipment. In general their performance is better than that of comparable smaller wheeled vehicles but does not equal that of tracked vehicles. The large diameter of the Goer type wheel is a disadvantage in some military operations.

9-21 UNUSUAL TIRES AND COMBAT TIRES

The vulnerability of the pneumatic tire has resulted in a demand for a nondeflatable tire that will have equal performance characteristics. Many concepts have been proposed and tried including solid rubber tires, elastomer foam-filled tires, elastic disc, and wire bristle tires. None of these concepts have met the requirements.

Combat tires are of the same basic construction as standard tires but are designed to operate without air pressure for limited distances in emergency situations. Compared to a standard tire, a combat tire is of much heavier construction, has more rigid sidewalls, and its heavily cushioned plies are spaced farther apart. The inside of the tire is lined with heavy rubber. A beadlock fits between the beads to prevent the tire from slipping on the rim and to hold it in position if deflated while supporting the load.

9-22 TUBES AND FLAPS

Three types of tubes are in general use on military vehicles, these are:

(a) Standard tubes constructed of single-layer rubber and intended for standard (non-combat) tires

(b) Combat tubes constructed of single-layer rubber with a smaller cross section for a given rim size to fit the smaller air space of combat tires

(c) Bullet-resisting tubes of heavy laminated construction which tend to seal bullet punctures

Most modern tubes are made from butyl rubber which has air retention properties superior to those of natural rubber.

Standard military tubes are listed in Reference 14.

Flaps are continuous strips of rubber (flat or contoured) that are placed between the tube and the rim to protect the tube from rim and bead injury.

SECTION V VERY LARGE DIAMETER TIRES (Ref. 17)*

9-23 GENERAL DISCUSSION

The off-road flotation tire is of thin wall, flexible carcass construction with rounded tread profile, thin to moderate tread thickness, and subdued tread pattern. These tires will withstand a much greater degree of flexing than heavier wall, heavier tread tires of the same size and, therefore, can be operated at greater than normal deflections and a correspondingly greater ground-contact area resulting in lower ground bearing pressures.

When operated within the load and inflation limits recommended, off-road flotation tires exhibit high tractive effort and low rolling resistance in soft soils, sand, mud, ice, and snow. A vehicle equipped with the optimum size flotation tire will have superior cross-country mobility compared to the same vehicle with normal combat tires but at the expense of reduced tire life.

The highly deflected tire lays down a flat tread pattern which has a tendency to compact the soil, sand, mud, or snow instead of digging into it. Vehicles so equipped can make more repeated passages through such soils before the ruts become deep enough to impede the vehicle than can the same vehicle equipped with the ordinary heavy wall, heavy tread aggressive pattern tires.

Tire chains may be used when necessary on flotation tires since they materially increase the tractive effort in some types of very slippery mud, on ice, and on compacted snow. Chains should not be used in sand or in any but the very worst slippery mud. They add nothing to the tractive effort in sand, and they promote digging and wheel trenching. They promote trenching in all soft soils and muds.

Even when equipped with chains, flotation tires do not damage roads or trails in soft soils and mud terrain as much as do highly inflated tires, heavy wall tires, or tires with an aggresive tread pattern.

These tires are not recommended for vehicles that are confined exclusively to highway or hard surface operations. They perform excellently on such surfaces and do minimum damage to the roads and highways but, because of their light construction and thin tread rubber, they have a relatively short life.

*The material presented in Section V was extracted freely from a comprehensive report (Ref. 17) by R. C. Kerr, then of the Arabian American Oil Co., formerly of the U.S. Army Engineer Board Transportation Section; and currently of the Office of the Director of Research and Development, U.S. Army Materiel Command.

9-24 THE TIRE AND RIM ASSOCIATION METHOD OF CALCULATING LOAD AND INFLATION SCHEDULES

The Tire and Rim Association (T. and R.A.) published load and inflation schedules for highway type tires are based on the following empirical equation

$$L = 0.425\,(S_1)^{1.39}\,(P)^{0.585}\,(D + S_1) \qquad (9\text{-}5)$$

where

L = optimum economic tire load, lb

$S_1 = \dfrac{(S_w - 0.4\,W)}{0.75}$, adjusted tire section, in. (9-6)

S_w = tire cross section diameter when mounted on the rim, in.

W = width of the selected rim, in.

P = inflation pressure, psi

D = bead diameter, in.

The tire loading schedules that are based upon the above equation have been proven by the test of time, and countless millions of tire miles of operation, to be entirely satisfactory for typical highway and paved road operations. Under these operating conditions, the tire maintains a relatively full section height. Tire deflections due to load are relatively small and the physical dimensions of the tire vary over rather narrow limits as the loaded wheels rotate. The variations that do take place seem to be adequately accounted for by the basic equation (Eq. 9-5).

However, off-road tires are allowed to deflect considerably under load so as to exert a lower and a more uniform ground pressure, and develop maximum tractive effort. Excessive flexing is recognized as being destructive to tires and the off-road tire is no exception. Since the off-road tire is of comparatively lighter construction, the side walls tolerate an extraordinary amount of flexing without appreciable damage. The abnormal flexing is necessary to achieve the desired mobility and, due to the lighter tire construction, this mobility can be attained with an *acceptable tire life*. Emphasis is placed on the term "acceptable tire life" because a distinction is being made with ***maximum tire life***. Higher inflation pressures, heavier tread stock, more plies, and more breaker strips would prolong the life of the tire, but at the expense of off-road mobility.

A re-examination of the basic T. and R.A. equation reveals several inadequacies when it is applied to off-road tires. For instance, only the width of the tire section is considered as a factor in the equation; the height of the tire above the flange is not. This seems to be a definite shortcoming of the equation when it is applied to tires that are greatly flattened in their profile during operations. Furthermore, the general dimensions, profile shapes, taper beads, and off-road rims are often radically different from tires for paved roads. The wider-than-normal rims of some more recent off-road tires cast doubt on the adequacy of the $(S_w - 0.4W)/0.75$ term.

Shortcomings of the T. and R.A. equation are recognized; various modifications have been applied. Perhaps the most noteworthy have been the efforts of Colonel Karl F. Eklund* and Mr. R. C. Kerr, then of the Arabian American Oil Company (ARAMCO).

9-25 THE WORK OF COLONEL KARL F. EKLUND

In 1945, Colonel Karl F. Eklund conducted an extensive study of all of the tire loading schedules published up to that time. This effort lead to the presentation of the following modified T. and R.A. equation

$$L = A_e (S_w)^{1.602} (D)^{0.7515} (P)^{0.585} \qquad (9\text{-}7)$$

where

L = optimum economic tire load, lb

S_w = cross sectional diameter of the mounted tire, in.

D = rim diameter, in.

P = inflation pressure, psi

A_e = speed-of-operation factor for Eklund's modified equation, dimensionless = 0.827, for T. and R.A. schedule TB-1A, Highway Operations. (Values for other schedules were also presented.)

*Colonel Karl F. Eklund, Corps of Engineers, U.S. Army, Director of Technical Division III of the Engineer Board, Fort Belvoir, Virginia, 1945.

Equation 9-7 yields load values acceptably close to those calculated by the basic T. and R.A. equation. While Equation 9-7 does provide for variations in rim widths, it does not take into consideration variations in height of the tire cross section. Nevertheless, with agreement from the Ordnance Corps and the Ordnance Pneumatic Tire Advisory Committee, Colonel Eklund proposed four compromise speed schedules for military vehicle operations which were an average of all the available data. He accepted the T. and R.A. Schedule TB-1A (Truck and Bus Highway Schedule) as the standard and rated the other schedules as percentages of TB-1A. His intent was to establish off-road operating schedules that could be used on the then existing military vehicles to enable them to achieve greatly increased off-road mobility by utilizing reduced tire inflation pressures. Four schedules were proposed which Col. Eklund designated as follows: (The corresponding ARAMCO schedules are given in parentheses.)

Military Highway—MH-1 (ARAMCO A1-50) This schedule was identical with T. and R.A. Schedule TB-1A and was to be followed for all high-speed hard-road operation.

Military Tactical—MT-1 (ARAMCO A1-25) This schedule utilized 75 percent of the inflation pressures indicated in MH-1 and TB-1A and speeds were restricted to 25 mph sustained and 35 mph intermittent in off-road operation.

Military Limited—ML-1 (ARAMCO A1-10) This schedule utilized 50 percent of MH-1 inflation pressures and was to be restricted to emergency operation in the 10 to 15 mph range of speeds for off-road service.

Military Emergency—ME-1 (ARAMCO A1-2) This was a minimum allowable genuine emergency schedule at 25 percent of MH-1 inflation pressure and speeds were restricted to 2 to 3 mph. It was to be used to permit vehicles to extricate themselves from sand or mud traps, or to permit traversing otherwise impassable soft soil terrains but was never intended to be used for long distance work.

These schedules were very easy to interpret because of the simple interrelationship steps in the inflation schedule. Knowledge of the MH-1 or TB-1A ratings, and the 3/4, 1/2 and 1/4 factors was all that was required to permit the operator to take full advantage of the greatly increased off-road mobility attainable through the utilization of these reduced inflation pressures.

The schedules were proposed after observation of extensive field testing and actual off-road operations. These schedules have not been widely accepted by industry but they have been used by the Arabian American Oil Company since that time, and are considered amply proven from the point of view of the operator.

The MH-1 and MT-1 schedules were intended to be used for both design and operation but the ML-1 and ME-1 schedules were intended to be used only by the operator to improve the mobility of existing vehicles whenever operating conditions so demanded; the schedules were not intended to be used as a basis in the design of new vehicles. This original intent was promptly forgotten, however, and for several years schedules ML-1 and ME-1 were used for design purposes, also. Since 1950, however, it has become increasingly apparent that schedules ML-1 and ME-1 are somewhat drastic from the design point of view. They are of value to the operator and use of them permit him to increase the mobility of his existing vehicles. However, utilization of the ML-1 and ME-1 inflation values does subject the tire to abnormal abuse. Although this is justified where accomplishment of the mission is paramount and tire life is of secondary importance, these schedules should be modified for design purposes where optimum tire life is desired.

9-25.1 EKLUND'S SCHEDULE MH-1 AND T. AND R.A. TB-1A

A study of all available highway service schedules published by T. and R.A., or submitted by manufacturers since that time, revealed only minor departures from the 1944 data. There seems to be no reason for modifying them for highway design. Therefore, factor A has been accepted at the T. and R.A. value of 0.425 for highway tires in highway operations.

The load ratings of a considerable number of highway tires were recalculated using the modified equation (Eq. 9-7) with factor $A = 0.425$, and discrepancies were found from as high as 5% overload to 7½ percent underload, with an average discrepancy of about 1 percent underload. These deviations were caused by variation in rim width, flange height, and thickness of tread. In several instances, the same tire deviated from about 1 percent overload rating of a wide rim to 1 percent underload rating for a narrow rim. It was concluded, therefore, that the modified equation (Eq. 9-7) with factor $A = 0.425$ is close enough to the T. and R.A. calculation (Eq. 9-5) for highway tires to justify its acceptanee.

Comparison calculations were made on a complete series of off-road tires ranging in size from 9.00×13 to 48.00×68, and depending on the degree of flattening of the profile, width of the rim, and the height of the flange—variations were found in load ratings from 2 percent overload to 30 percent underload in the TB-1A range. The very flat profile tires suffered the greatest underload variation and the full round profile tires the least. Tires mounted on narrow rims are underrated compared to the same tire on a wide rim. In the normal profile off-road tires, the new equation underrates the load on the average about 5 percent. The maximum underrate occurred with the 48.00×68 tire which has a sectional width of 48.00 in. and a sectional height above the flange of only 24.5 in. Calculation by the modified equation (Eq. 9-7) gives load values of only 70 percent of those calculated by the regular T. and R.A. equation (Eq. 9-5). Observations of actual tire performances tend to corroborate these load ratings. It was, therefore, concluded that a value of 0.425 for factor A was acceptahle for design purposes for the entire range of tires at highway speeds.

9-25.2 EKLUND'S SCHEDULES MT-1, ML-1, AND ME-1

Since MT-1, ML-1, and ME-1 were based on TB-1A, the basic factor 0.425 remained constant, and conversion to other speeds was accomplished as follows:

$$A_o = \frac{0.425 \times (50)^{0.25}}{(V)^{0.25}} = \frac{1.13}{(V)^{0.25}} \qquad (9\text{-}8)$$

TABLE 9-1
EKLUND'S MILITARY LOAD AND INFLATION SCHEDULES
(For operational use on existing vehicles)

Schedule	Speed of Operation, mph	Inflation, % of TB-1A, Fixed Load	Load, % of TB-1A, Fixed Inflation	Elkund's Operations Factor A_o
MH-1	50–70	100	100	0.425
MT-1	25–35	75	118	0.505
ML-1	10–15	50	150	0.635
A1-5	5–7	37	180	0.756
ME-1	2–3	25	225	0.952

NOTE: A1-5 Special schedule used by ARAMCO only.
MH-1 is identical with T. and R.A. Schedule TB-1A.

where

A_o = Eklund operations factor, dimensionless
V = sustained speed, mph
50 = basic sustained speed assumed for MH-1, mph

Eklund's proposed schedules and factor A_o values are shown in Table 9-1.

9-25.3 MODIFIED EKLUND'S SCHEDULES

Eklund's operating schedules have been subsequently recalculated using the ARAMCO modified equation (see paragraph 9-26) instead of the basic T. and R.A. equation. These calculations indicate lower recommended loads for flattened profile tires than did the older series. The revised schedules were given the following new designations in order to distinguish them from the older series. (Corresponding ARAMCO schedules are given for additional reference):

(a) *Military Highway Operations—MH-2 (ARAMCO A2-H and AR-H).* This schedule is indentical with the new design schedule A2-H and operating schedule AR-H. For full round section tires it indicates loads that are very slightly lower than Eklund's MH-1, T. and R. A. TB-1A and ARAMCO A1-50; but as the tire profile flattens out, the difference in load rating becomes more pronounced.

(b) *Military Tactical Operations—MT-2 (ARAMCO AR-25).* Speeds 25 to 35 mph. The preceding remarks, made with reference to the MH-2 schedule, apply to the correspondence with Eklund's MT-1 and ARAMCO A1-25.

(c) *Military Limited Operation—ML-2 (ARAMCO AR-10).* Speeds 10 to 15 mph. The remarks, made in the preceding discussion of the MH-2 schedule, apply to correspondence with Eklund's ML-1 and ARAMCO A1-10.

(d) *Military Emergency Operation—ME-2 (ARAMCO AR-E).* Speeds 2 to 3 mph. The remarks, made in the preceding discussion of the MH-2 schedule, apply to correspondence with Eklund's ME-1 and ARAMCO A1-2.

These operating schedules are presented for study and comparison purposes only. They are no longer recommended for operations (Table 9-2).

9-26 THE ARAMCO MODIFIED EQUATION FOR OFF-ROAD TIRES

The modification presented below of the basic T. and R. A. equation (Eq. 9-5) has been developed by ARAMCO after repeated tests of many other equations and has been checked by applying it to a wide range of off-road tires. It appears to be realistic and to possess flexibility in adaptation to flat profile tires and variations in rim width, bead seat dimensions, and flange height.

$$L = A(S_f)^{1.39}(P)^{0.585}(D_f + S_f) \qquad (9\text{-}9)$$

where

L = optimum economic load, lb

$$A = \frac{0.425\,K(50)^x}{(V)^x}, \qquad (9\text{-}10)$$

ARAMCO type-of-operations factor, dimensionless

TABLE 9-2
MODIFIED EKLUND SCHEDULES

Schedule		Speed of Operation, mph	Inflation, % of MH-2 and A2-H, Fixed Load	Load, % of MH-2 and A2-H, Fixed Inflation	Eklund's Operations Factor A_o
Eklund	ARAMCO				
MH-2	AR-H	50–70	100	100	0.425
MT-2	AR-25	25–35	75	118	0.505
ML-2	AR-10	10–15	50	150	0.635
	AR-5	5–7	37	180	0.756
ME-2	AR-E	2–3	25	225	0.952

K = tire life expectancy factor, dimensionless
V = vehicle design speed, sustained, mph
x = variable exponent expressing approach of actual load to maximum permissible load, dimensionless
$S_f = (S_w S_{hf})^{0.5}$ cross sectional factor, in. (9-11)

S_w = section width, in.
S_{hf} = tire sectional height above flange, in.
P = inflation pressure, psi
D_f = diameter of top of rim flange, in.

A discussion of the elements of Equation 9-9 is given in the paragraphs which follow.

This equation attempts to establish a load value that will yield an ''acceptable'' optimum economic tire life. It does not contemplate maximum possible tire life or attempt to set a certain number of miles of tire life. The operator who is forced to traverse rough, unkept roads, rocky areas, debris, and obstacles cannot possibly expect to achieve the same number of miles of tire life as the operator who confines his work to smooth, well-kept roadways free from obstacles and debris.

L, therefore, is an optimum quantitative value; the actual miles of tire life achieved depend on the speeds of the vehicles, the type of operation, the physical condition of the road or area to be traversed, servicing and maintenance control of both vehicle and tires, and, to a very great extent, on the operational control exercised by the operator.

L is simply a statement that if a tire carrying the selected load L is operated at the recommended inflation pressure P in a certain type of operation A, it will, on the average, yield satisfactory tire life.

9-27 THE OPERATIONS FACTOR A

The following deals with the development of factor A, the type-of-operations factor, used in Equations 9-9 and 9-10. This particular element of the equation has been the most difficult to evaluate and evaluation has been delayed for a long time because of failure to clearly differentiate between:

(a) the effect of speed on tire life when inflation and load are constant.

(b) the effect of speed on the load rating where inflation is held constant and the load is variable.

Particular emphasis is placed on the need for the clear separation of permissible operating practices from the responsibility of the design engineer to keep within conservative design limits.

Factor A combines the evaluation of contemplated vehicle speed V and acceptable tire life expectancy K. The derivation of factor A as given by Equation 9-10 has been selected after tests of numerous other assumptions. The factors K, V, and x are discussed in subsequent paragraphs of this chapter.

The basic T. and R.A. equation (Eq. 9-5) is based on the expectation of highway operations at high speeds on hard surface roads. It does not include an operations factor for other types of conditions. Numerous schedules have been published by the T. and R. A. and other manufacturers for slow speeds and special types of operations which, in effect, are based on a factor similar to A. These schedules have been established for certain specific services such as earthmoving, low bed trailer service, agricultural and construction machinery operations, logging operations, etc. There are, however, wide variations in the load ratings for the various classes of service schedules published by the manufacturers. This is caused by the fact that each type of service has been evaluated in the light of the entire operational environment involved. Not only has speed been evaluated but also the severity of the service, the condition of the roads (particularly the presence or absence of rocks and debris), the propensity of the particular operators to overload or overspeed the equipment, etc. There is need, therefore, for a series of schedules at various speeds of operation, specifically intended for long distance cross-country and off-road transport operations.

9-28 THE EFFECT OF SPEED ON LOAD RATING

9-28.1 INFLUENCE OF SPEED AT CONSTANT LOAD AND INFLATION

It should be noted that the following considerations indicate the effect of speed on the variable load rating of tires. They do not represent the effect on tire life of variable speed at constant load and inflation. For many years the tire industry has warned that the destructive effects of speed (at constant load and inflation) varies about as the square of the speed. No data exist that justify adopting any other assumption at this time. Certainly the assumption appears to be acceptable in the range of speeds in which military vehicles operate.

It is logical to assume that the influence of speed is analogous to the influence of load, and an exponent that varies according to departure from optimum might well be the case. This departure from the fixed exponent assumption would very likely take the form of accelerated destruction rate at high speeds, the same as the load destruction influence appears to be accelerating at heavy loads. At this time, however, no modification of the accepted fixed exponent 2.0 is proposed for use in Equation 9-10 for operations involving constant load and constant inflation.

9-28.2 INFLUENCE OF SPEED AT VARIABLE LOAD AND CONSTANT INFLATION

The influence of speed on tire life at constant load and inflation is fundamentally different from the effect of speed on the tire load rating. At constant load and inflation, the tire deflection remains constant regardless of the speed. In the case of the tire load rating, the inflation is constant but the load is variable, thus, subjecting the tire to variable deflections.

If we use the constant load and constant inflation condition as a guide, it appears that the load ratings vary as the square of the velocity ratio. That is, if we reduce the speed by one half, we should be able to quadruple the load. A study of this assumption, however, reveals that preposterous loads could be possible with relatively small reductions in speed. A study of the known operating load limits of many tires substantiated Eklund's assumption that the rated load varied as the inverse of $(V)^{0.25}$. This, then, was accepted as a satisfactory value for the speed exponent, under conditions of variable load and constant inflation, to be applied to existing vehicles as a measure to improve the performance of their tires. ARAMCO, however, was of the opinion that this assumption should be modified *for design purposes.* As a consequence, many trial assumptions of new exponent values were made and evaluated. The most acceptable solution involves the development of factor K which is presented below before continuing with the discussion of the speed exponent.

9-29 TIRE LIFE EXPECTANCY FACTOR K

With all other elements of the operation (except load) constant, it is apparent that tire life TL bears some inverse relationship to load L. Furthermore, it appears that the relationship is exponential; so we may state:

$$TL = f\left(\frac{1}{L^y}\right) \tag{9-12}$$

Tire manufacturers have for many years warned that overload operation seriously reduces the life of tires. The statement has been repeatedly made that tire life varies inversely as the square of the load. The transportation industry generally operates tires in a relatively narrow range of loads and inflations and, within this range, the second power relationship appears to be realistic and serves as a most useful rule. However, there is a lack of accurate, controlled engineering test data in the overload, particularly in the extreme overload ranges; but field operations indicate that this simple relationship is not applicable to such conditions.

The Arabian American Oil Company has had many years of experience operating tires at high deflections under loads that are far in excess of the manufacturers' recommendations. This has permitted extensive field observation which indicates that tire life under high deflection conditions is reduced at a higher rate than would be expected from the second power relationship. When the tire is deflected enough to approach the buckle point, tire life is reduced very rapidly. When the buckle point is reached or exceeded, the tire is quickly ruined. It can be stated, therefore, that the tire must not be permitted to reach the buckle point except under extreme emergency conditions, when accomplishment of the mission is more important than preservation of the tire.

How close the buckling can be approached without suffering an unacceptable reduction in tire life depends on the individual time. In two tires of identical exterior dimensions but of differing sidewall thickness (such as the 14.00×20-8 ply and the 14.00×20-12 ply) the thick wall tire will buckle at a lower deflection than the other. Therefore, the thin wall flexible tire can operate at higher deflections than the heavy wall tire for comparable tire damage or comparable reduction in tire life.

Very thin wall extremely flexible tires, such as the swamp buggy type, will tolerate a most severe wrinkling and buckling without excessive damage or immediate failure. It is expected that this wrinkling and buckling will reduce the life of the tire. The effect, however, is so moderate when compared to the same degree of wrinkling or buckling in stiff carcass tires that it has not been adequately evaluated. It can only be said that wrinkling or buckling in these tires should not be deliberately planned by the designer. The equipment operator, however, may subject these tires to wrinkling and buckling when a critical situation so demands and he may do so without much risk of causing excessive damage or immediate tire failure.

The buckling point of a tire is determined by the relative flexibility of its construction. Hence, the number of ply and ply material in the carcass, the angle of the cords, the number of breaker strips, the thickness and profile of the tread, and the rubber compounds used all exert their influence. Each size and type of tire must be test-loaded and test-operated to determine its deflection at the point of buckling. This should be done at normal operating pressures and at several pressures in the overload range. Then, having determined the buckle point deflections, a line should be drawn across the load and inflation tables indicating the absolute deflection limits below which the tire must not be operated except in high risk emergency. This procedure demands a separate set of tabulations or curves for each size, type, make, and ply construction, but it appears to be the only solution if the tire is to be operated in a highly deflected condition.

Several conditions were considered when attempting to develop a simple relationship of load vs tire life expectancy. The assumption that TL varies inversely as L^2 results in load values that appear reasonable down to 70 percent of normal life expectancy but yields unrealistically high load values at 50 percent and unacceptably high values at 25 percent of normal life. The assumption that TL varies inversely as L^3 results in load values that appear to be too low at 50 and 25 percent of normal life expectancy but appear reasonable at 10 percent. Also, the assumption that TL varies inversely as L^4 results in load values that are much too low in all ranges down to 10% of normal life expectancy. One is, therefore, forced to the conclusion that the relationship is not a simple exponential and that the buckling point is analogous to the yield point in the deformation of elastic materials. The load exponent, y, (Equation 9-12) that is applicable down to the buckling point of the tire is apparently a variable which increases with the load. Furthermore, it has a value of 2.00 at normal loads and normal tire life expectancy but reaches higher values as the load is increased; particularly as the buckling point is approached.

In view of these considerations, exponent y was selected on the basis of the departure of the operating load from optimum load as

$$y = \frac{L_o + L_a}{L_o} = 1 + \frac{L_a}{100} \qquad (9\text{-}13)$$

where

L_o = optimum load expressed as a percent = 100%

L_a = actual operating load expressed as a percent of optimum

Equation 9-12 can be re-written as

$$TL = \frac{100\,(L_o)^y}{(L_a)^y} = \frac{(100)^{y+1}}{(L_a)^y} \qquad (9\text{-}14)$$

where

TL = tire life expectancy expressed as a percent of the optimum.

Equation 9-14 may be used to determine the expected tire life for any load as a percentage of the optimum life when operated at normal rated loads. Normally, however, the designer wishes to select a desired tire life and determine the loading that will give this life by applying a simple factor. The development of this factor was accomplished by writing Equation 9-14 as

$$(L_p)^y = \frac{100\,(L_o)^y}{(TL)_d} = \frac{(100)^{y+1}}{(TL)_d} \qquad (9\text{-}15)$$

where

L_p = permissible tire design load in percent of optimum

L_o = optimum tire load in percent = 100%

$(TL)_d$ = desired tire life expectancy expressed as a percent of optimum

$$y = \frac{L_o + L_p}{L_o} = 1 + \frac{L_p}{100} \qquad (9\text{-}16)$$

from which

$$K = \frac{L_p}{100} = \frac{100^y}{(TL)_d} \qquad (9\text{-}17)$$

and

$$L_p = KL_o \qquad (9\text{-}18)$$

If the actual value of the optimum tire load in pounds, as recommended in an appropriate tire schedule, is used for L_o in Equation 9-18, the resulting permissible design load L_p will also be in pounds instead of as a percent of optimum load.

Table 9-3 lists tire loads L_a as percentages of the optimum L_o and their corresponding values of y. Table 9-4 lists desired tire life expectancies $(TL)_d$ and their corresponding permissible design loads L_p, both as percentages of the optimum load L_o, and their corresponding K factors. These data may be useful in future tire calculations. The designer is again cautioned, however, that in all cases of tire overloading, the buckling point of the tire must not be exceeded. Furthermore, this critical buckling point is not taken into account in Equations 9-13 through 9-18 nor in Tables 9-3 and 9-4. This information must be obtained from experimental sources.

TABLE 9-3
SELECTED VALUES OF y

Actual Load Expressed As a % of Optimum	Value of y
25	1.25
50	1.50
75	1.75
100	2.00
125	2.25
150	2.50
175	2.75
200	3.00

TABLE 9-4
TIRE LIFE EXPECTANCY AND FACTOR K

Designed Tire Life Expectancy, % of Optimum	Permissible Design Load, % of Optimum	Factor K
200	66	0.66
150	79	0.79
100	100	1.00
75	115	1.15
50	135	1.35
25	170	1.70
12-1/2	200	2.00

9-30 THE ARAMCO SCHEDULES A2, A2/2, AND A2/4

The T. and R.A. schedules are generally based on the belief that the operator always desires maximum tire mileage or tire life. Therefore, no factor corresponding to K appears in the T. and

R.A. equations (original and modified, Eqs. 9-5 and 9-7, respectively). There are occasions, however, when the designer is justified in designing for an overload with the full knowledge that the tires will suffer abnormal abuse and that tire life will be shortened. Realizing this, the following design schedules have been established to give an acceptable tire life for a particular operation if the operator carefully observes all the limitations in the basic equations:

(a) *Schedule A2—Full Normal Tire Life Expectancy.* This schedule is to be used in the design of vehicles engaged in long distance hauling where a vehicle can be expected to wear out several operation has been designated as Schedule A2 and sets of tires in its operational life. This type of is based upon a factor $K = 1.00$ in the expectation of achieving 100 percent normal tire life.

(b) *Schedule A2/2—Half Normal Tire Life Expectancy.* There are many operations where reduced tire life can be tolerated. Certain specialized vehicles travel only limited distances and it is not necessary to equip them with tires that will outlive the vehicle or will fail from age before their normal life expectancy is reached. From the vehicle design point of view, many factors limit the tire size that can be used in the vehicle. Obviously, it is not necessary to select tires that will yield 25,000 miles of operation if the vehicle dimensions are limited and the vehicle will not be expected to exceed, say, 5,000 miles per year. By the time the normal mileage expectancy is reached, the tires will have deterioriated because of age.

From the operational point of view, an operator may be faced with the necessity of using equipment that is at hand because he cannot tolerate the delay or expense involved in waiting for new equipment with adequate tires. Under these conditions, he may be justified in overloading the tires, proceeding with the job, and accepting a reduced tire life. For these conditions, Schedule A2/2 is applicable. It is based upon a factor $K = 1.35$ and results in one half the normal tire life.

(c) *Schedule A2/4—Quarter Normal Tire Life Expectancy.* There are other classes of vehicles that will travel only a short distance in their entire life. Many types of construction equipment—cranes, mobile power plants, mobile shops, etc.—are such vehicles. Specialized vehicles such as amphibious craft, pneumatic tire drydocks, etc., will never travel more than a few hundred miles in their entire life. From the point of view of tire life—not considering mobility, low rolling resistance, high tractive effort, etc.—it is obviously unnecessary to equip such vehicles with tires that are capable of many thousands of miles of tire life in the particular operation. For such design and operation conditions, Schedule A2/4 is applicable based upon a life expectancy factor $K = 1.70$ giving a life expectancy of one quarter the normal life.

9-31 DEVELOPMENT OF SPEED EXPONENT x

As mentioned in paragraph 9-28.2, Eklund established a speed exponent of 0.25 for the operational schedules MH-1, MT-1, ML-1, and ME-1. This exponent has served, and still serves, a most useful purpose in operations; it has been retained in the calculation of the new operation schedules MH-2, MT-2, ML-2, ME-2, AR-H, AR-25, AR-10, AR-5, and AR-E. For several years, however, it has been apparent that the use of this exponent yields load values that subject the tire to abnormal abuse at the lower speeds. In developing ARAMCO design schedules A2, A2/2 and A2/4, it was immediately apparent that each schedule demanded an appropriate speed exponent; otherwise the calculated load values at slow speeds became fantastically high. Obviously, however, the slow speed load values must not reach the buckling point of the tires in any of these design schedules. This is not to be interpreted, however, as forbidding the operator to deflate to the buckling point in an emergency.

Exponent values were first approximated by graphic analysis based upon the values of K presented in the previous discussion and an overload terminal value of $A = 0.957$ as originally proposed by Eklund. It is believed that there is ample justification for this value.

After the graphic analysis, a number of trial calculations established the relationship that actual dynamic loading tests for each particular tire would presumably yield somewhat different values;

TABLE 9-5
FACTOR A-TYPE OF OPERATION FACTOR FOR DESIGN PURPOSES—
OFF-ROAD GENERAL TRANSPORT SERVICE

Design Speed, mph	Normal Tire Life		Half Tire Life		Quarter Tire Life		Eklund's Schedule	
	Schedule A2		Schedule A2/2		Schedule A2/4		M-Series	
Highway	A2-H	0.425	A2/2-H	0.574	A2/4-H	0.723	MH-2	0.425
25-35	A2-35	0.483	A2/2-35	0.630	A2/4-35	0.765	MT-2	0.505
10-15	A2-15	0.573	A2/2-15	0.708	A2/4-15	0.825	ML-2	0.635
5-7	A2-7	0.650	A2/2-7	0.780	A2/4-7	0.872	AR-5	0.756
2-3	A2-3	0.770	A2/2-3	0.880	A2/4-3	0.940	ME-2	0.952
Emerg.	0.957		0.957		0.957		0.957	

but for a typical thin wall off-road tire, the values calculated appear quite reasonable. The speed relationship was determined to be

$$x = \frac{L_m - L_s}{3L_m} \tag{9-19}$$

where

x = speed exponent (dimensionless)
L_m = maximum permissible load before buckling, in percent of Schedule A2
L_s = scheduled load for the desired tire life expectancy, in percent of Schedule A2

This analysis is based on the relative proximity of the scheduled loads to the limiting buckling load where tire damage occurs at a greatly accelerated rate.

Eklund, after study of all the data available to him at that time, established 225 percent of optimum load to be about the average of the absolute limits beyond which one should not go. Pending exact data on the buckling point of individual tires, it is recommended that this value be used as the maximum overload limit. This 225 percent value can be expressed as a factor 2.25; similarly, the percent load of any schedule can also be expressed as a factor and is, in fact, the factor K discussed previously.

Equation 9-19 can, therefore, be reduced to

$$x = \frac{2.25 - K}{6.75} \tag{9-20}$$

With this equation, values for exponent x were calculated as follows

DESIGN SCHEDULES

A2	$x = 0.185$	$TL = 100\%$	$K = 1.00$
A2/2	$x = 0.133$	$TL = 50\%$	$K = 1.35$
A2/4	$x = 0.0815$	$TL = 25\%$	$K = 1.70$

OPERATING SCHEDULES

M-1	$x = 0.25$	TL Variable	$K = 1.00$
AR	$x = 0.25$	TL Variable	$K = 1.00$

9-32 FACTOR *A* TABULATED

Equation 9-10 is repeated here for convenience.

$$A = \frac{0.425K(50)^x}{(V)^x}$$

The substitution of the appropriate values of K and x developed for the three ARAMCO schedules results in the following values for factor A:

(a) *Schedule A2—Normal Tire Life Expectancy*

$$A = \frac{0.425 \times 1.00 \times (50)^{0.185}}{(V)^{0.185}}$$

(b) *Schedule A2/2—Half the Life Expectancy*

$$A = \frac{0.425 \times 1.35 \times (50)^{0.133}}{(V)^{0.133}}$$

(c) *Schedule A2/4—Quarter Tire Life Expectancy*

$$A = \frac{0.425 \times 1.70 \times (50)^{0.0815}}{(V)^{0.0815}}$$

TABLE 9-6
CORRESPONDENCE BETWEEN SUSTAINED VERSUS INTERMITTENT SPEEDS

Sustained Speed, mph	Intermittent Speed, mph	Approximate Ratios Adopted for Design Listing	
		Sustained	Intermittent
50	70.75	50	70
25	35.35	25	35
10	14.15	10	15
5	7.08	5	7
2	2.83	2	3

The calculated values for these design schedules are presented in comparison with Eklund's operation schedules in Table 9-5.

9-33 RELATIONSHIP BETWEEN SUSTAINED AND INTERMITTENT SPEEDS

Tire operations involving intermittent high speeds with intervals of slower speeds are not as damaging to tires as are sustained speeds over the same distance at the listed high speed. An arithmetic average of the speeds does not lead to a satisfactory solution because, at constant load and constant inflation, tire damage is directly proportional to the square of the speed in the ordinary range of tire deflections.

Based upon a study conducted by Karl Eklund and Richard C. Kerr in 1944, the following relationship was determined

$$\frac{(V_i)^2}{(V_s)^2} = 2.0 \qquad (9\text{-}21)$$

where

V_i = intermittent speed, mph
V_s = sustained speed, mph

In the 1944 study, the desire was to establish guidance for the vehicle operator, and specifying maximum permissible sustained speed appeared to be a more conservative approach than the specification of maximum possible speeds, because operators have a tendency to assume that the listed values are satisfactory for sustained speed operation. The design engineer, however, should in fact use the intermittent speed values in the design.

Both speed values are presented in the prepared tire schedules, although the item is not included in the basic equation. This guidance has been prepared primarily for use by the design engineer and he must proceed on the assumption that the vehicle will very frequently be operated at maximum sustained speeds. Selected speed ratios calculated on the basis of velocities squared are given in Table 9-6.

9-34 TIRE CROSS SECTIONAL FACTORS

The original T. and R.A. equation presented a tire cross-sectional diameter factor S_1 (see Eq. 9-6) such that

$$S_1 = \frac{(S - 0.4W)}{0.75}$$

This factor has proven to be satisfactory for application to normal highway type tires of full sectional height when mounted on rims that are about 75 percent as wide as the cross sectional width of the tire, and for normal bead seat diameters and flange heights. It does not evaluate high angle taper bead seats with correspondingly low flange heights.

Many large size tires—particularly in the earth-mover, cross-country, high-flotation, and swamp-buggy types—are flattened out in profile and there is an appreciable difference between cross-sectional diameter and sectional height of the tire above the flange. It is obvious that a full, round-section tire of any cross-sectional diameter and of normal sectional height, deserves a higher load rating than a tire of equal cross-sectional diameter, but of reduced tire height. This is particularly important in some of the new giant-size tires that in one instance has a ratio of height to width of about 0.51.

It is known that the flattened profile tire must deflect more under a given load in order to develop an equivalent area of tread pattern and, thus, it suffers greater abuse from sidewall flexing. Actual field tests show that these low profile tires are more vulnerable to rock bruising and impact breaks than are the full round-section tires. In view of these factors, it seems reasonable to conclude that the sectional height of the tire must be considered in tire load calculations.

Furthermore, the T. and R.A. factor S_1 nearly cancels out the effect of changes in rim width W. Accurate test data on the effect of rim width variation are not available but it is believed that rim width does influence the load bearing capacity of tires. Industrial experience has indicated that the use of wide rims may prolong the life of tires operating under severe conditions and there is a general concurrence that very narrow rims are undesirable.

It has been shown that the tractive effort, sinkage, and rolling resistance of a tire depends upon the dimensions of its ground-contacting area. Tires of varying rim widths but exhibiting contact areas that are alike, both in size and shape, will have identical mobility characteristics. Despite the fact that the mobility characteristics remain unchanged, advantages lie with the wider rimmed tires in the form of improved tire life, less vulnerability to side wall breaks at points of concentrated flexing, and greater stability under side loads.

In the case of extreme rim widths, sidewall flexing is localized much more than it is in tires with normal rim widths. This increases side wall fatigue and leads to reduced tire life.

For full round-section off-road type tires, acceptable tire life has been obtained with tires having rim width to section width ratios of 0.50 to 0.75. However, the flattened profile tires need wider than normal rims.

A graphical analysis of the dimensions of available tires was made to summarize the available rim widths. The data points were widely scattered in the plot but, for whatever value it may have, the results are shown in Table 9-7 with 90 percent of all available rims falling within the band covered by the table. These values are presented as a general guide to the selection of rim widths.

TABLE 9-7
RATIO OF RIM WIDTH TO MOUNTED SECTIONAL WIDTH

Ratio $\frac{S_{hf}}{S_w}$	Ratio $\frac{W}{S_w}$	Rim Width, Percent of Mounted Section Width	
		Spread	Average
1.0	0.50 to 0.68	50 to 68	59
0.9	0.55 to 0.71	55 to 71	63
0.8	0.60 to 0.74	60 to 74	67
0.7	0.65 to 0.77	65 to 77	71
0.6	0.70 to 0.81	70 to 81	75
0.5	0.75 to 0.84	75 to 84	79
0.4	0.80 to 0.87	80 to 87	84

S_{hf} = sectional height above the flange
S_w = sectional width of mounted tire
W = width of rim

Extremely wide rims are not recommended for highly deflected tires but satisfactory results can be achieved over a very considerable range of width to section ratios. The effect of varying the rim width needs further evaluation but it is expected that, within the normal range of width ratios, its effect on the load rating will not be appreciable.

As the rim width decreases, the cross sectional diameter also generally decreases. The sectional factor S_f (Eq. 9-11) reflects this change more realistically than does the T. and R.A. factor S_1 (Eq. 9-6) which very nearly cancels out the effect.

9-35 TAPERED BEAD SEAT REFERENCE POINTS

The use of the advanced rim with tapered bead seats complicates calculations somewhat because the tire size designation for this rim is customarily given in odd numbers of inches for the bead seat diameter. Otherwise, the exterior dimensions of the tire are unchanged, and there is general agreement that the load rating need not be increased despite the fact that the term $(D + S_1)$ is larger.

9-36 TOP OF THE FLANGE REFERENCE POINT

The flange diameter of the tapered bead seat rims remains unchanged and, thus, is a useful reference point. Aside from the convenience of using the top of the flange as a reference point, it appears logical that it should be used in load capacity calculations because the part of the tire below the top of the flange is rigidly confined and cannot participate in flexing or yielding.

9-37 TIRE CROSS SECTIONAL FACTOR, S_f

Many combinations of sectional width S_w and sectional height above the flange S_{hf} were studied. The dimensions of all the available off-road tires were examined in an attempt to establish a relationship between S_w, S_{hf}, and tire O.D. based upon the shape and area of the tread pattern. Tread patterns are approximately eliptical in shape—elongated in the plane of the tire. The transverse dimension is influenced primarily by S_w, and the greater longitudinal dimension by S_{hf} and tire O.D.

A relationship that is consistent throughout the normal range of off-road tire deflections has not been established but a relationship does seem to exist for certain deflections. The study, however, has not been encouraging.

The cross-sectional area of the tire appears to be a possible index of its load-carrying capacity. The relationship was studied in considerable detail with some degree of success.

Studies of various exponents of the sectional area resulted in the selection of the following relationship as the most realistic

$$S_f = C(S_w S_{hf})^{0.5} \tag{9-22}$$

where

S_f = cross sectional factor, in.
S_w = sectional width, in.
S_{hf} = sectional height above the flange, in.
C = a constant

This relationship yields values that are very close to the T. and R.A. values resulting from Equation 9-6 when constant $C = 1.01$. This constant has been reduced to 1.00, thus lowering the load rating of off-road tires by 1%, which is not undesirable. This relationship has, therefore, been adopted and included in the ARAMCO modified equation (Equation 9-9).

The influence of the sectional factor on the load rating is exponential in character. Investigations conducted by R. C. Kerr to evaluate this exponential relationship substantiated the T. and R.A. value of 1.39 as a reasonable value for the sectional factor exponent.

9-38 INFLATION PRESSURE AND INFLATION EXPONENT

The T. and R.A. equation (Equation 9-5) presents the term $(P)^{0.585}$. The investigative efforts of R. C. Kerr while at ARAMCO have justified this exponential relationship. An exponent that yields constant deflection regardless of load, results in load ratings that are too high in the heavy load ranges and too low in the light load ranges. Inflation to the $(P)^{0.585}$ relationship decreases the deflection in the higher load ranges. The continued use of this relationship is, therefore, recommended.

9-39 DIAMETER OF THE RIM FLANGE

The T. and R.A. Equation 9-5 evaluates the influence of overall tire diameter by the use of the term $(D + S_1)$, where D is the diameter of the bead seat and S_1 is the sectional width factor. A term that is more directly related to the radius, or the rolling radius, of the tire is preferred. Many combinations have been studied by ARAMCO and the term $(D_f + S_f)$ has been adopted, where D_f is the diameter of the rim flange in inches and S_f is the cross section factor.

The evaluation of this term is important because of the wide divergence in rim diameter to sectional height ratios of many specialized tires from the ordinary highway-type tires. The Atlas 16.00 × 16, for instance, has a very small diameter rim for the sectional height of the tire; while the Firestone 48.00 × 68 tire has a very large diameter of rim compared with the sectional height of the tire. Thus, the simple modification $(D_f + S_f)$ to the T. and R.A. Equation 9-5 yields values for both of these tires that are reasonable. This modification has been incorporated into the ARAMCO equation (Equation 9-9).

9-40 COMPARISON OF RESULTS

A considerable number of check calculations were performed on the highway classification of tires using the ARAMCO modified equation (Eq. 9-9). On the average, resulting load values were slightly lower than those given in the T. and R.A. Schedule TB-1A for narrow rims and slightly higher than those given for wide rims. When applied to the normal shape, round-section earthmover and off-road tires, the modified equation yielded slightly lower load ratings than did the T. and R.A. equation. In the case of the flat profile tires, the new equation indicated considerably less load capacity than did the T. and R.A. equation. These differences were anticipated and the load values determined by the ARAMCO method are considered to be more realistic to off-road operations.

9-41 LOAD RATINGS FOR DUAL TIRES

9-41.1 DISCUSSION

Dual tire combinations subject tires to more than normal wear. The individual tires of a dual pair are subject to all the following abnormal conditions.

(a) Rocks and other debris become wedged between the tires.

(b) Impacts on obstacles by one tire of a pair subjects it to shock at higher than normal load.

(c) Crowned roads cause greater than normal loads to bear on the inside tires of a pair.

(d) Trough shaped off-road trails cause abnormal loads on the outside tires of a pair.

(e) Awkward locations of the inflation stems which discourage checking and adjustment of inflation.

(f) Failure of an underinflated tire to show a warning bulge, because the load is carried by its companion; thus, it avoids detection by visual inspection.

(g) An abnormal load is carried by a more highly inflated tire of a dual pair if the two tires are not inflated to exactly the same pressure.

(h) The dual pair is subject to more frequent nail and trash puncture because it covers a wider track than an equivalent load capacity single tire.

(i) Tire wear is increased on curves due to unequal distances traveled by rigidly mounted wheels.

Furthermore, the severity of the wear and abuse to which dual mounted tires are subjected increases with the tire loading. When the loading approaches the buckling point of the tires, the individual tires of the pair cannot tolerate any unbalance of load between them. When an unbalance does occur, the overloaded tire immediately buckles and tire damage is greatly accelerated. However, it is obviously impractical to say that the load capacity of a pair of dual mounted tires at very heavy loads is only equal to the capacity of the single tire. Similarly, it is not justifiable to rate the carrying capacity of each tire in a dual pair at 100% of its scheduled load, even in smooth highway operations.

Load factors for design work on vehicles intended for secondary roads, trails, and cross-country operations have been established. Observation of many millions of miles of such operations has given ample justification to these factors. They are the results of the best efforts possible at the time they were established. In some cases, they may indicate dual wheel loads that are somewhat higher than may actually be practical but, lacking controlled test data, there is some reluctance to limit them unjustifiably.

9-41.2 DUAL TIRE LOAD FACTORS

Obviously, the dual mounted pair of tires have a greater load capacity than a single tire under similar conditions but the combination must be rated lower than that of two single tires. The exact percentage of full rating, however, is subject to variation with operating conditions and is particularly subject to the skill, competence, and degree of control exercised by the vehicle operator.

The "type-of-operations factor," factor A in Equation 9-9, is the most logical index of severity of probable abuse and is the term selected for manipulation in establishing a workable dual-tire load factor. The following equation expresses the relationship of factor A and the observed operational demand most realistically.

$$L_w = \frac{157.5}{A^{0.25}} \tag{9-23}$$

where

L_w = indicated load rating for a dual tire wheel expressed as a percent of the load rating of a single tire wheel operated at one-half the wheel load and at scheduled inflation

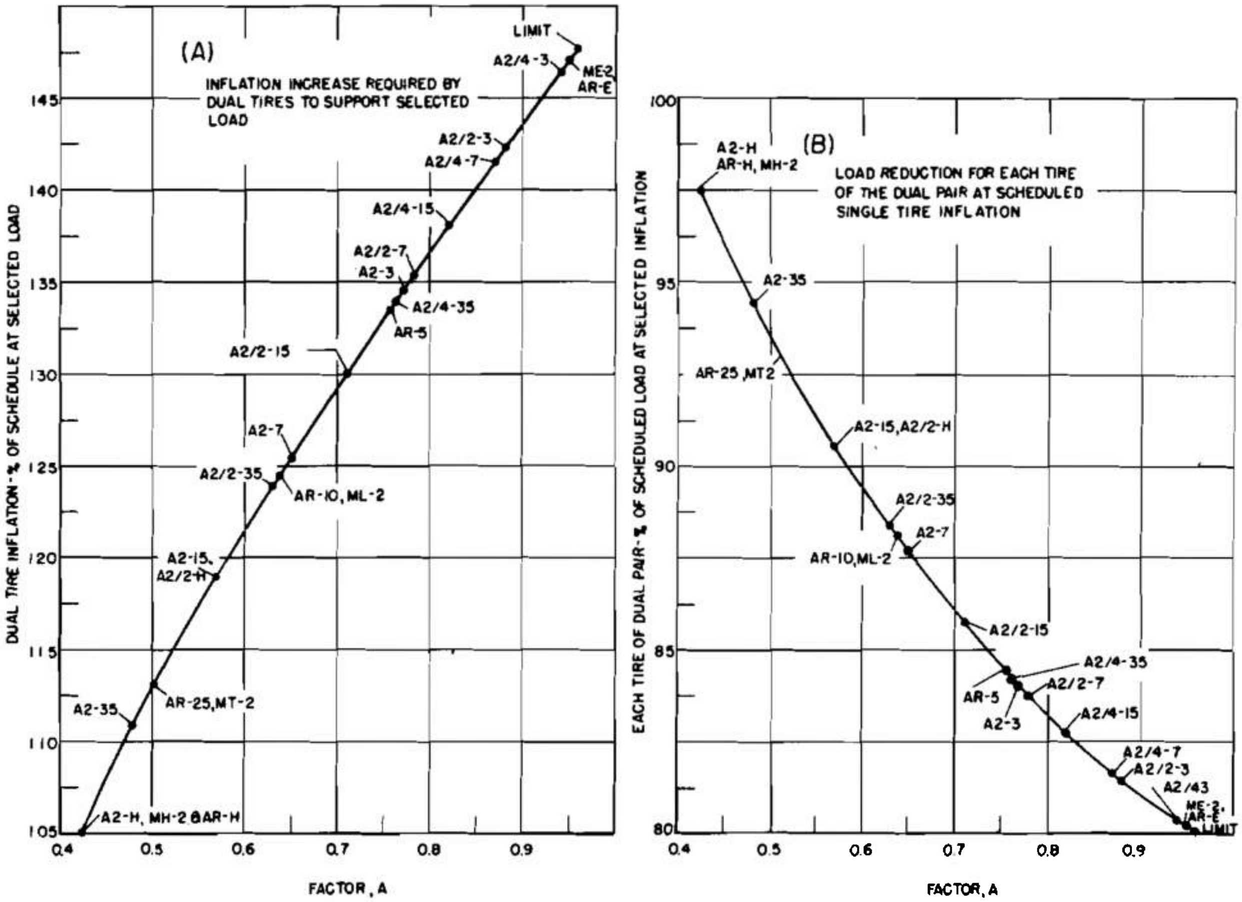


Figure 9-14. Dual-Tire Load and Inflation Adjustment Factors

A = the type-of-operations factor (see paragraph 9-27)

157.5 = constant selected to indicate a dual tire wheel load of 195 percent of single tire load rating at A2-H schedule inflation pressure for the single tire at half wheel load.

In practice, the actual load on each tire (half the wheel load) is determined and the inflation is increased above that scheduled for a single tire at that load, as shown in Table 9-8 and Figure 9-14(A). Such inflation gives the tire a fictitious, or apparent, load rating L_a that appears to be higher than a single tire load rating at that inflation. This difference is the dual tire load factor F_d.

Alternately, if it is necessary to carry a wheel load demanding that each tire carry full normal load, we must increase the inflation pressure above single tire normal.

The values given in Table 9-8 and Figure 9-14 may be used both in design and in operations. The approach is different in each case but the end result is the same. The basic concept is that the individual tires of a dual pair must be operated at a higher inflation pressure than is indicated by the appropriate load and inflation schedule for a single tire at half wheel load. The indicated higher inflation pressure increases the *apparent load capacity rating of the tire.* In actuality, however, the abnormal abuse to which the dual tire is subjected justifies decreasing this apparent load rating (Figure 9-14(B)) to the actual load values in an attempt to achieve normal life.

Table 9-9 presents the relationship of increasing load capacity with increasing inflation for any tire.

To determine the required inflation pressures for dual tires mounted on an existing vehicle, first determine the dual tire wheel load and divide by 2 to establish the load each tire is expected to carry. From the appropriate load and inflation schedules for the specific tire, determine the inflation that would be required to carry that load (half the wheel load) if the tire was to be operated on a single rather than a dual wheel. Then increase the inflation of each tire in order to achieve an apparent increased load capacity. Table 9-10 and Figure 9-14(A) indicate the required increase in inflation pressure for dual tire combinations at the various schedules.

Example:

Schedule A2/2 Half Normal Tire Life Expectancy
Existing vehicle tire size 14.00 × 20-8 Ply
Tire load 4500 lb
Wheel load 9000 lb

Speed of Operation, mph	Schedule No.	Sched. Infl. for Single Tire at 4500 lb Load, psi	Type of Operation Factor A	Infl. for Dual Tires, Percent of Single Schedule	Indicated Infl. for the Dual Tires, psi	Apparent Load Rating for Each Tire of Pair, lb
Highway	A2-H	35.0	0.425	105	37	4650
25-35	A2-35	28.0	0.483	111	31	4800
10-15	A2-15	20.5	0.573	119	25	5000
5-7	A2-7	16.5	0.650	125.5	21	5200
2-3	A2-3	12.5	0.770	134.5	17	5400
Emerg.	Limit	8.5	0.957	147	12.5	5600

Example:

Schedule A-2 Full Normal Tire Life Expectancy
Existing vehicle tire size 14.00 × 20-8 Ply
Tire load 4500 lb
Wheel load 9000 lb

Speed of Operation, mph	Schedule No.	Sched. Infl. for Single Tire at 4500 lb Load, psi	Type of Operation Factor A	Infl. for Dual Tires, Percent of Single Schedule	Indicated Infl. for the Dual Tires, psi	Apparent Load Rating for Each Tire of Pair, lb
Highway	A2/2-H	20	0.574	119.5	24	5000
25-35	A2/2-35	17	0.630	124.0	21	5100
10-15	A2/2-15	14	0.708	130.0	18	5200
5-7	A2/2-7	12	0.780	135.0	16	5300
2-3	A2/2-3	10	0.880	142.0	14	5500
Emerg.	Limit	8.5	0.957	147.0	12.5	[illegible]

TABLE 9-8
DUAL-TIRE WHEEL LOAD RATINGS

Factor A	Dual Wheel Load, Percent of Single Tire Schedule, L_w	Each Tire Load, Percent of Single Tire Schedule	Tire Inflation, Percent of Schedule for Half Wheel Load	Apparent Percent of Scheduled Load Capacity Each Tire, L_a	Dual Tire Load Factor F_d
0.425	195	97.5	105.0	103.0	1.030
0.500	187	93.5	113.0	107.5	1.075
0.600	179	89.5	121.5	112.5	1.125
0.700	172	86.0	129.5	116.5	1.165
0.800	167	83.5	136.5	120.0	1.200
0.900	162	81.0	143.5	123.5	1.235
0.957	160	80.0	147.5	125.0	1.250

The higher inflations may reduce the mobility of the vehicle in sand, mud, ice, and snow, and the operator may be forced to use lower inflation pressures. Tire life will, obviously, be reduced for the sake of mobility.

It is frequently possible to install oversize tires on existing vehicles. This is a practical solution when increased mobility is needed. The oversized tires can be operated at lower inflation pressures to improve mobility without overloading.

The designer should use large single tires of adequate capacity whenever possible in preference to using dual mounted tires. Vehicular mobility will be greatly improved, and the larger tire is invariably much stronger and more resistant to carcass breaks than either of the dual pair that must otherwise be used. There are times, however, where the designer is forced to use dual tires because of dimensional limits imposed by the operations. When modifying existing vehicles, it is frequently impossible to install a single tire of adequate size to carry the dual tire load. In such situations, the designer should try to select tire sizes that would ordinarily be considered oversized.

When designing a new vehicle, the procedure which follows may be used in determining the tires to be selected. Determine the wheel load to be carried. Divide by two to establish the actual load each tire is expected to carry. Decide the type of operations and establish factor A. Determine the degree of mobility required for the operation in which the vehicle is expected to be employed (par. 9-43). Enter the Off-Road Tire Selection Chart (Figures 9-15 to 9-19) for the appropriate schedule. Spot the individual tire load on the selected mobility factor line. This will indicate the inflation pressure required to provide the desired degree of mobility. From Tables 9-8 and 9-9, determine the increased load rating required L_a. Move to the right along the required inflation pressure line on the tire selection chart to the increased load L_a. Select the tire closest to this plotted point. When in doubt between two tires, select the larger of the two choices, provided the physical dimensions of the vehicle and the operational requirements permit. Operate this oversize tire at the inflation pressure scheduled for the *apparent load*.

TABLE 9-9
LOAD RATING VERSUS INFLATION

Inflation, Percent of Normal	Load, Percent of Normal
100	100.0
105	103.0
110	105.8
115	108.6
120	111.5
125	114.1
130	116.8
135	119.3
140	121.5
145	124.0
150	126.0

TABLE 9-10
INFLATION PRESSURES FOR DUAL MOUNTED TIRES

Speed of Operation, mph	Schedule A2			Schedule A2/2		
	Schedule No.	Factor A	Inflation, Percent of Schedule	Schedule No.	Factor A	Inflation, Percent of Schedule
Highway	A2-H	0.425	105.0	A2/2-H	0.574	119.5
25-35	A2-35	0.483	111.0	A2/2-35	0.630	124.0
10-15	A2-15	0.573	119.5	A2/2-15	0.708	130.0
5-7	A2-7	0.650	125.5	A2/2-7	0.780	135.0
2-3	A2-3	0.770	134.5	A2/2-3	0.880	142.5
Emerg.	Limit	0.957	147.0	Limit	0.957	147.0

Speed of Operation, mph	Schedule A2/4			Eklund's Operating Schedule			
	Schedule No.	Factor A	Inflation, Percent of Schedule	Schedule No.	Factor A	Inflation, Percent of Schedule	ARAMCO Oper. Schedule
Highway	A2/4-H	0.723	131.0	MH-2	0.425	105.0	AR-H
25-35	A2/4-35	0.765	134.	MT-2	0.505	113.0	AR-25
10-15	A2/4-15	0.825	138.	ML-2	0.635	124.5	AR-10
5-7	A2/4-7	0.872	141.5	AR-5	0.756	133.5	AR-5
2-3	A2/4-3	0.940	146.5	ME-2	0.952	147.0	AR-E
Emerg.	Limit	0.957	147.0	Limit	0.957	147.0	Limit

Example: Assume that there is a requirement to select tires for a vehicle currently in the design stage. The following specifications have already been established:

Wheel load 10,000 lb
Half wheel load 5000 lb
Service A2-15, consisting of:
  Normal tire life expectancy
  Speeds 10-15 mph
  Cross-country operations
Factor $A = 0.573$ (from Table 9-10; A2-15)
Dual tire actual load 5000 lb = 100%
Dual tire inflation increase required = 119.5% (from Table 9-10 and Figure 9-14)
Dual tire apparent load increase = 111.5% (from Table 9-9)
Dual tire apparent load $= 5000 \times 1.11$ $= 5550$ lb

From the Off-Road Tire Selection Chart A2-15 (Figure 9-17) at a tire load of 5550 lb, the following tires and inflations are indicated:

(a) 14.00 × 20-8 ply duals, inflated to 29 psi for a mobility factor of 90%
(b) 16.00 × 20-10 ply duals, inflated to 17 psi for a mobility factor of 114%

At a load of 10,000 lb, the following selections can be made:

(a) 18.00 × 25-12 ply singles, inflated to 28 psi for a mobility factor of 102%
(b) 21.00 × 25-16 ply singles, inflated to 16 psi for a mobility factor of 122%

It should be noted that, at any given mobility rating, the 18.00 × 25 tire is capable of carrying slightly more than twice the load of the 14.00 × 20

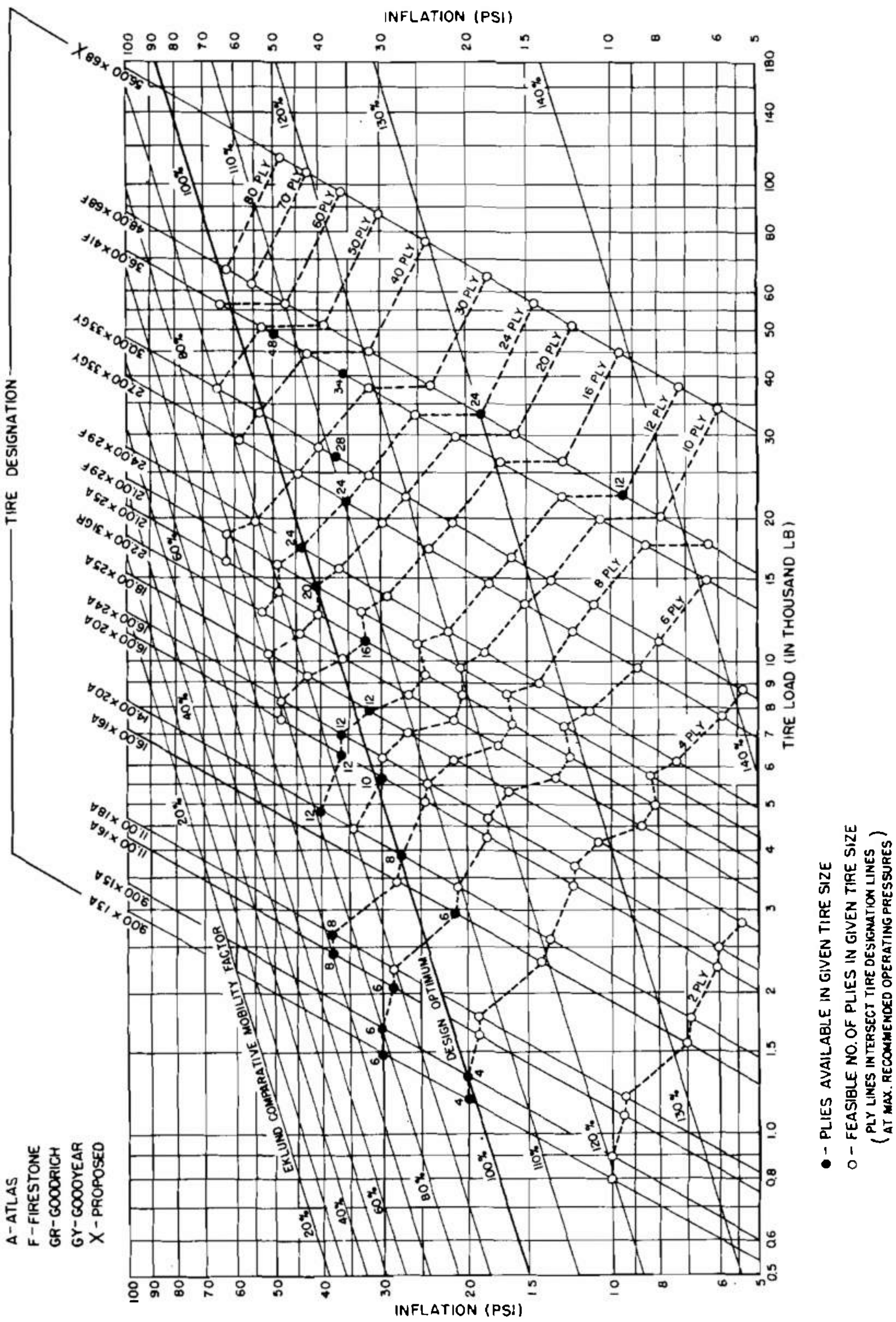
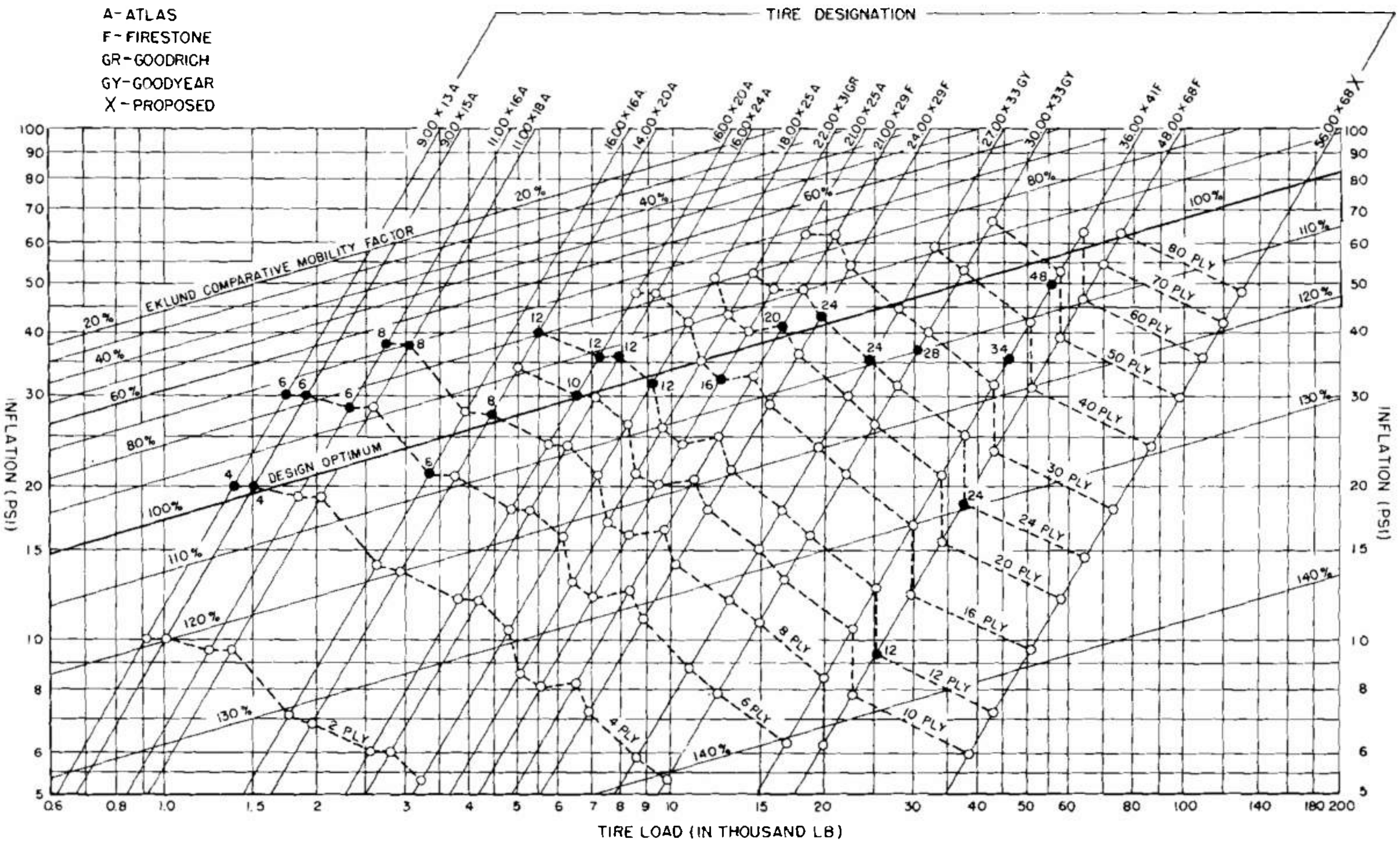


Figure 9-15. Off-Road Tire Selection Chart—Schedule A2-H, Normal Tire Life Expectancy (Speed, Above 35 mph-Single Tire only: For Dual Tires at indicated load increase inflat. to 105% or at indicated inflation decrease individual tire load to 97%. These tires are not suitable for high speed highway operation.)



● - PLIES AVAILABLE IN GIVEN TIRE SIZE

○ - FEASIBLE NO OF PLIES IN GIVEN TIRE SIZE (PLY LINES INTERSECT TIRE DESIGNATION LINES AT MAX RECOMMENDED OPERATING PRESSURES)

Figure 9-16. Off-Road Tire Selection Chart—Schedule A2-35, Normal Life Expectancy (Speed, 25-35 mph. Single Tire only. For Dual Tires at indicated load increase inflat. to 111% or at indicated inflation decrease individual tire load to 94%.)

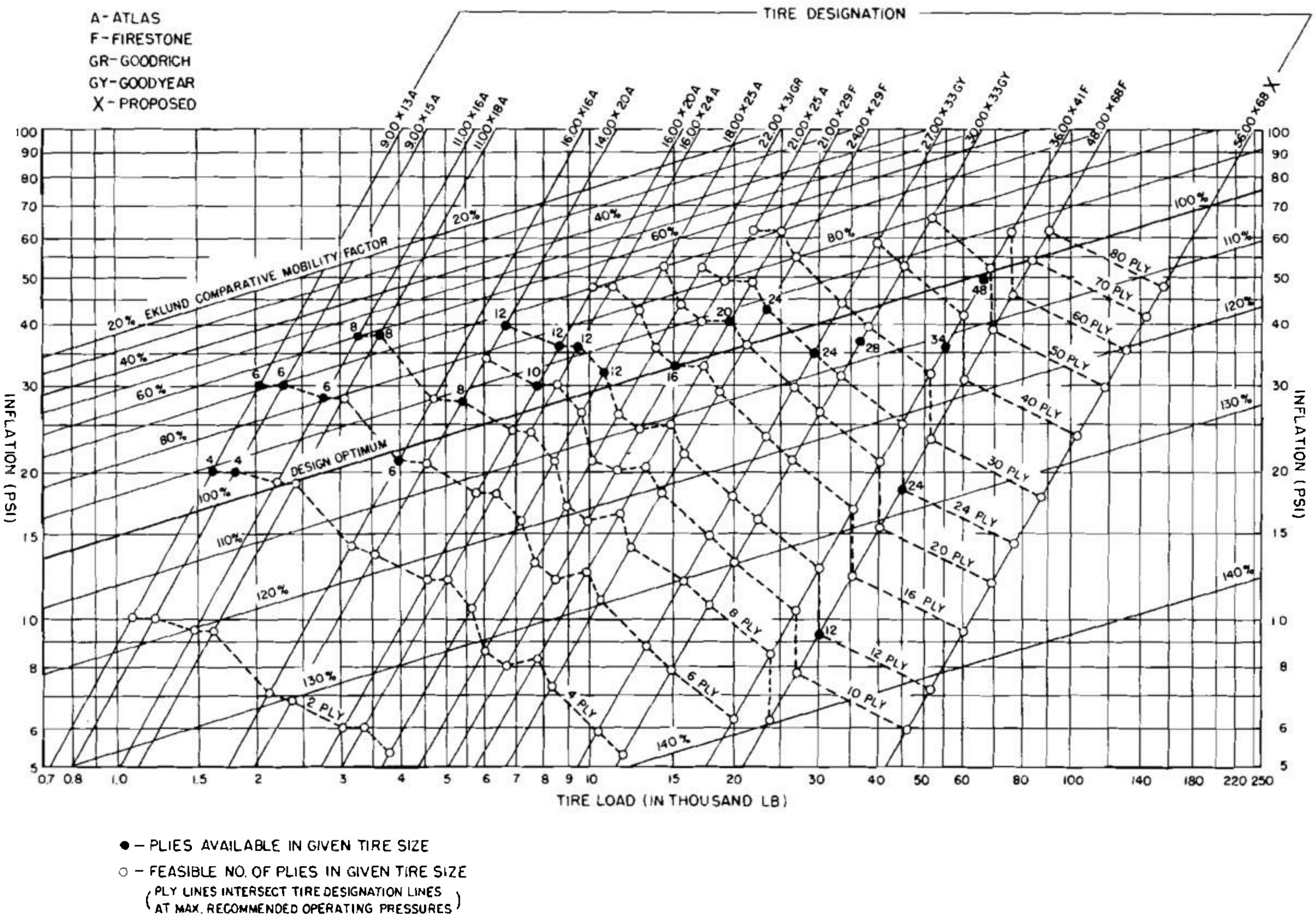


Figure 9-17. Off-Road Tire Selection Chart—Schedule A2-15, Normal Tire Life Expectancy (Speed, 10-15 mph Single Tire only. For Dual Tires at indicated load increase inflat. to 119% or at indicated inflation decrease individual tire load to 91%.)

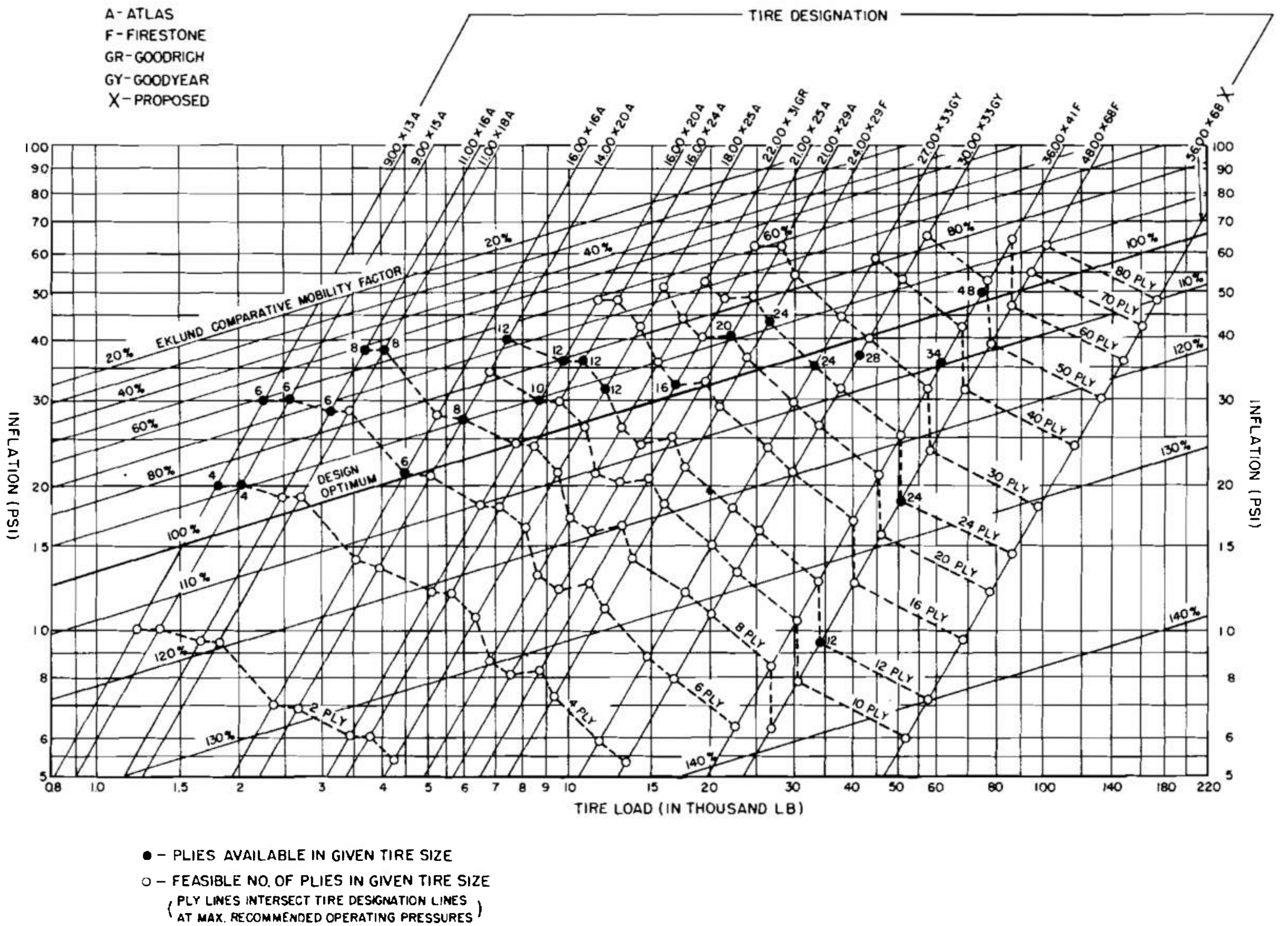


Figure 9-18. Off-Road Tire Selection Chart—Schedule A2-7, Normal Tire Life Expectancy (Speed, 5-7 mph. Single Tire only. For Dual Tires at indicated load increase inflat. to 125% or at indicated inflation decrease individual tire load to 88%.)

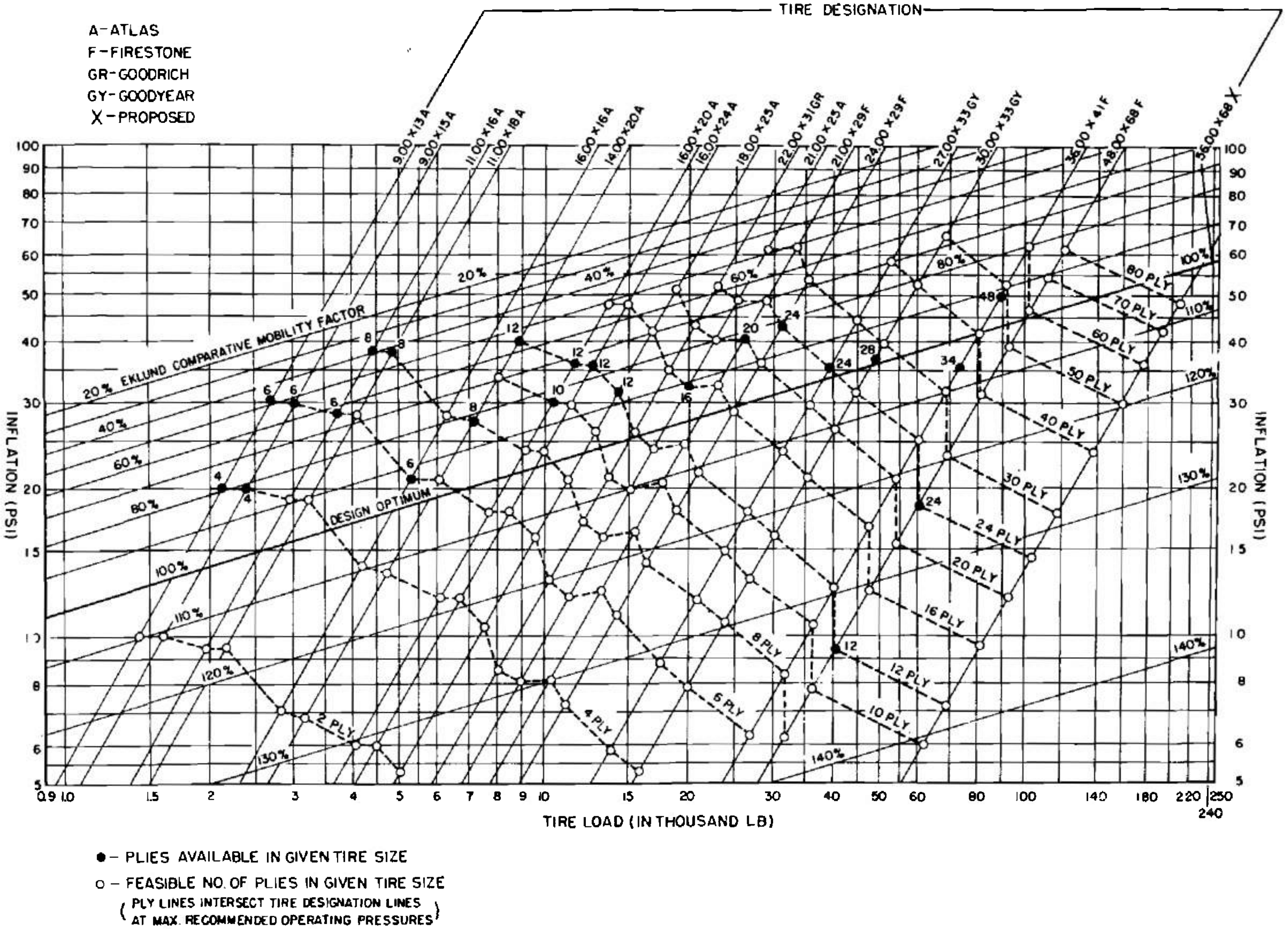


Figure 9-19. Off-Road Tire Selection Chart—Schedule A2-3, Normal Tire Life Expectancy (Speed, 1-3 mph. Single Tire only. For Dual Tires at indicated load increase inflat. to 135% or at indicated inflation decrease individual tire load to 84%.)

tire; and the 21.00 × 25 tire has twice the load capacity of the 16.00 × 20 tire. The final selection will be influenced by such factors as space and weight limitations, relative costs, effects on steering, and service requirements.

9-42 SINGLE VERSUS DUAL TIRES

9-42.1 OFF-ROAD MOBILITY

In off-road motor transport operations, a marked increase in mobility is achieved when, at any given inflation pressure or ground bearing pressure, a large single tire of any desired load capacity is used to replace a pair of dual tires, each of half that load rating at the same inflation pressure. This does not mean that singles are always superior to dual tire combinations. As illustrated in the preceding example, the 18.00 × 25-12 ply is superior to the 14.00 × 20-8 ply dual but is inferior to the 16.00 × 20-10 ply dual. The 21.00 × 25-16 ply is, however, superior to the 16.00 × 20-10 ply dual. The rule holds, however, that if we replace any dual pair with a single tire capable of carrying the full wheel load at the same inflation pressure required by the duals, we have achieved increased off-road mobility. On dirt roads, soft soil, sand, mud, ice, and snow, the tractive effort exhibited by the larger size tire is higher than the combined effort of the pair of smaller dual tires; and the rolling resistance of the larger tire is less than the pair. When applied to either a truck or a truck-tractor and trailer combination, the net result is a very substantial gain in mobility. Furthermore, the increased axle height of the large tire gives the vehicle more ground clearance than is possible with dual tires.

9-42.2 VEHICLE WIDTH AND STABILITY

The larger tire is invariably narrower than a corresponding pair of duals. For any given tire width limit, the large single tire provides greater clearance between tires for frame, machinery, and suspension components. Design and fabrication problems are, therefore, minimized and the resulting wider wheel-track provides greater vehicle stability. Conversely, the large single tires permit a narrower vehicle of a given stability to be designed when narrowness is a requirement.

9-42.3 WEIGHT OF RUNNING GEAR

The large single tire running gear assembly may be designed lighter in weight than the tires, axle, hub, wheel, and suspension assembly required for duals. In designing axles, spindles, and hubs for the dual tire combination, the designer must consider that one tire must frequently carry repeated full load shocks and, in some cases, must carry the full dual wheel load for prolonged periods. The large single tire design needs make no provision for such shifting of loads. The length of the cantilever from the spring saddle to the outer tire load-line is always greater for the dual pair than the comparable distance in the single tire design. The designer can take advantage of this fact and utilize lighter weight spindles, axle housings, and bearings. Furthermore, the single tire and wheel are not as heavy as the combined dual tires and wheels.

9-42.4 EFFECTS OF ROADS AND TRAILS

Hard surfaced roads are usually crowned in the center, causing the inner pair of dual tires to carry excess loads. Dirt roads and cross-country trails very frequently exhibit the reverse profile with elevated sides and depressed centers. In this case, the outer tires of the dual pair must carry excess loads. This condition is so prevalent in certain desert operations that tire failures and axle housing failures due to this cause are recognized as a most serious maintenance problem.

Dual tires find applications in situations that demand a low vehicle silhouette or a low level, flat platform deck, but at the cost of reduced relative mobility, vehicle width, and increased weight.

9-42.5 SAFETY AND EMERGENCY OPERATIONS

Dual tires are often regarded as a safety feature, or a limp-home device in case of one tire failure. This feature is of questionable merit, however, especially in the case of heavy vehicles. In most cases, the carrying capacity of each tire is completely utilized; and if one tire of the pair fails, the vehicle must be brought to a halt very soon or the other tire will also fail and may even catch fire. In some cases of this type, the entire vehicle was lost. The dual tire cannot, therefore, be regarded as a dependable safety device.

9-42.6 CONTROL OF INFLATION

The control of inflation of dual tires is a very difficult maintenance problem. The valve stems are hard to get at and, unless the operator conscientiously uses a tire gauge to equalize the inflation in both tires of the pair, the tires operate under a great handicap. The load line is displaced on the hub bearings and spindle when one tire carries an excessive load. Visual examination of the tire bulge is of no value, because both tires exhibit the same bulge even if one be overinflated and the other deflated. Frequent checking with a tire gauge is the only practical solution. The large single tire is easy to inflate and visual observation of the tire bulge permits a true evaluation of the inflation conditions. Furthermore, changes in inflation pressure do not shift the load line on the spindles and bearings.

On many military vehicles, driver control of tire inflation is a recognized requirement. Such control for single tires has been accomplished and will be improved. It is not impossible to devise such control for dual tires but it is a more complex design task and has not yet been successfully accomplished.

Interconnection of the tires in a pair so that both can be simultaneously inflated or deflated to identical pressures has long been recognized as a very desirable feature, but the mechanical difficulties involved and the extra work required in changing a tire has prevented the adoption of such a device. In such a device an operator controllable check valve must be provided that will permit the operator to reduce the pressure in both tires when desired but that will prevent both tires from deflating in case of puncture to one of them.

9-43 COMPARATIVE MOBILITY

9-43.1 DISCUSSION

We cannot ignore mobility considerations in vehicle design. The accompanying charts, schedules and tabulations present optimum load and inflation values based on Eklund's estimates of equivalent mobility characteristics. These estimates are based on a study of a very complete program of rolling resistance and tractive effort tests in soft sand. Tires ranging from 6.00 $\times$ 16 to 36.00 $\times$ 40 size, in numerous ply construction and tread patterns, were tested by the Corps of Engineers and the Ordnance Corps during WW II. These data have been augmented by numerous agencies since that time by additional sand tests and tests in mud, snow, and soft soils.

These tire tests have shown that, at identical inflation pressures and exterior dimensions, the heavier the carcass construction and tread thickness, the higher the rolling resistance and the lower the tractive effort. In order to achieve equal mobility, the heavier carcass tire must be operated at lower inflation than the flexible carcass tire. Furthermore, the larger tires exhibit a higher degree of mobility than do the smaller tires and the larger the tire the higher the permissible inflation for comparable mobility in all soft soil conditions.

9-43.2 COMPARATIVE MOBILITY ESTIMATES

A method of making comparative mobility estimates was established by Eklund in 1945 after a study of all available data. When applied to off-road operations, this method yields satisfactory results. The estimates do not purport to be quantitatively exact for all types of terrain and for all types of tires. They possess very useful comparison values, however, and are reliable over a wide range of terrain types for the tires marked "optimum ply" construction. They are useful in vehicle design and enable the designer to select tires that will endow his vehicle with improved mobility. In actual design it is recommended that, where a choice is possible between two tires, the one with the greater mobility factor be selected.

In many cases, it may not be possible to design for maximum mobility. Such design demands lightly constructed, flexible tires of such extreme size that it might not be possible to confine them within the dimensional limits of the vehicle. The designer must compromise and be willing to accept an "optimum mobility" or an "acceptable mobility" that will permit the vehicle to perform acceptably over the type of road or terrain. In most cases, he cannot afford to design for a great excess of mobility over the normal service demand. He should strive, however, for as high a degree of mobility as is practicable.

9-43.3 EKLUND'S MOBILITY FACTORS

Eklund established a relationship between tire load, size, and inflation pressure to produce the best mobility over all types of terrain. This relationship is expressed by Equation 9-7 in paragraph 9-25. A vehicle that fulfilled all the requirements of this equation was considered to have a par mobility with a numerical value of 100. He showed that the greater the deviation of a wheel load from this optimum condition, the poorer was the resultant mobility. In order to assure that a selected tire would not be too large or too small for the right amount of mobility, he devised an index of relative mobility, or a mobility factor MF. This mobility factor applies to each tire of the vehicle, and was originally stated as

$$MF = \frac{1}{2}\ (200\text{-}100d) \qquad (9\text{-}24)$$

where d is the average ratio of the variation between actual and optimum conditions and may be written as

$$d = \frac{L_d + P_d}{2} \qquad (9\text{-}25)$$

where

$L_d = \dfrac{L_a - L_o}{L_o}$, variation of the load from the optimum, dimensionless (9-26)

$L_a =$ actual load, lb

$L_o =$ optimum load, lb

$P_d = \dfrac{P_a - P_o}{P_o}$, variation of the inflation pressure from the optimum, dimensionless (9-27)

$P_a =$ actual inflation pressure, psi

$P_o =$ optimum inflation pressure, psi

Equation 9-24 may, therefore, be written as

$$MF = 150 - 25\left[\frac{L_a}{L_o} + \frac{P_a}{P_o}\right] \qquad (9\text{-}28)$$

Eklund recommended in his report (Ref. 18) that a vehicle not have a mobility factor less than 85 nor more than 115.

The mobility factor as established by Eklund is not to be construed as an exact expression of drawbar pull, rolling resistance, or absolute mobility. It is merely a reasonable approximation of off-road mobility that is useful in making comparative evaluations of vehicle performance or of tire selections.

Eklund's work has been criticized because it does not provide the designer with an insight into the true behavior of a wheel and because it does not take into account variations in soil properties. If soil properties had been varied through a wide range during Eklund's experiments, the resulting equations would represent very broad approximations; and if the soil conditions had not been varied, the equations would represent a very limited range of circumstances. Nevertheless, despite their shortcomings, the Eklund equations have been applied to advantage by many engineers.

The mobility factor is indicative of the performance of powered wheels as well as unpowered, or trailing wheels. The higher the MF value, the higher the tractive coefficient on a powered wheel and the lower the rolling resistance of a trailing wheel. When comparing two all-wheel drive prime movers or two unpowered wheel trailers, first determine the MF for each tire, and then add the MF values of all the tires and divide by the number of tires to establish an average value. These average values provide good estimates of comparative mobility for either unit.

The combination of high MF value for the tractor and high MF values for the trailer indicate the unit with the highest comparative mobility for any two similar combinations but this system does not permit comparison of a tractor-trailer combination with an all-wheel-drive truck.

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CHAPTER 10

SUSPENSIONS FOR TRACKED VEHICLES*

SECTION I GENERAL DISCUSSION (Ref. 1)

10-1 INTRODUCTION

The design goals for suspension systems for tracked vehicles are identical to those applying to the vehicle as a whole. Among these goals are: (a) maximizing mobility, utility, reliability and durability, (b) minimizing fuel consumption, weight, maintenance and cost, and (c) achieving a high level of standardization of components.

The purpose and function of the suspension system for tracked vehicles are discussed in Chapter 2. The characteristics, purpose, and functions of the various components and subsystems that are combined to form the total suspension system are discussed in subsequent sections of this chapter. Components within the current suspension systems for tracked vehicles include: (a) track assemblies, (b) road wheels and tires, (c) sprockets, (d) support rollers, (e) road wheel arms or links, (f) spring systems, (g) damping systems (shock absorbers), (h) bump stops, and (i) track envelope adjusters. A number of standard types of these components are illustrated in Appendix II, showing how these components are integrated into a suspension system for the track-laying vehicles.

In addition, a rather extensive tabulation of a representative number of standard suspension systems and their characteristics are also given in Appendix II. These data should provide useful guidance to those designers who are new to the military vehicle design field and should serve as a useful check of past and current design accomplishments to the more experienced military designer.

10-2 GENERAL TRACKED SUSPENSION TYPES

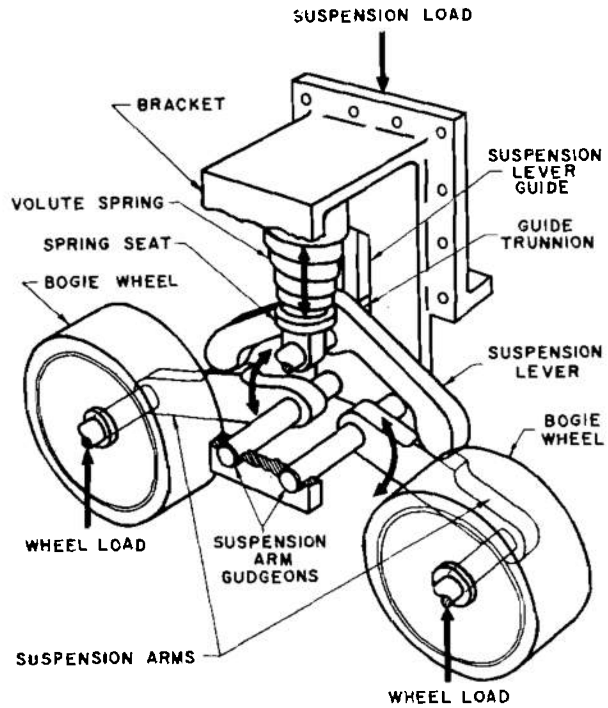

Figure 10-1. Vertical Volute Spring Bogie

10-2.1 BOGIE SUSPENSIONS

A bogie for a tracked vehicle (tank bogie) is a suspension assembly consisting of a system of links, arms, and springs interconnected in a manner to permit two or more road wheels to function together in tandem. This type of suspension was used extensively on early tracked vehicles but is now obsolete The independent type of suspension is used exclusively on modern American tracked vehicles.

The bogie suspension provides a walking-beam effect that divides the load equally between a pair of tandem wheels. When one of the wheels is displaced, or is subjected to a vertical track force, an equal force is reflected on the wheel of the bogie unit. This arrangement permits the vehicle to ne-

* Written by Nicholas R. Rome of the IIT Research Institute, Chicago, Illinois.

TABLE 10-1
CHARACTERISTICS AND QUALITIES OF BOGIE SUSPENSION VERSUS INDEPENDENT SUSPENSION FOR TRACKED VEHICLES

INDEPENDENT SUSPENSION	BOGIE SUSPENSION
1. The displacement of one or several road wheels does not substantially change the displacement of the remaining road wheels.	1. The maximum displacement of a single road wheel is greater than the maximum simultaneous displacement of all the road wheels of the entire bogie unit.
2. Changing the loading of one road wheel does not change the loading of the others while the hull is stationary. Redistribution of the loads on the road wheels is possible only by changing the position of the hull.	2. Changing the loading of one road wheel correspondingly changes the loading of the other wheels of the bogie unit.
3. In encountering a terrain irregularity, an impact is immediately received by only one road wheel.	3. In encountering a terrain irregularity, the impact is received by all of the road wheels of the particular bogie unit.
4. Decommission of one road wheel does not impair the effectiveness of the remaining road wheels except for the increased load.	4. Decommission of one road wheel impairs the functioning of the entire bogie unit.
5. Large diameter road wheels are usually employed; never less than 18 in.	5. Numerous small-diameter road wheels are used, improving weight distribution but reducing tire life.
6. Mechanically less complex than bogies, thereby facilitating maintenance. More protected from damage by enemy action and terrain debris.	6. More complex than independent; therefore, more subject to malfunction and difficult to maintain. More exposed to ballistic attack.
7. Usually softer suspension with higher reserve of energy absorption and road wheel deflection, compared with bogies. Dampers essential.	7. Usually harder suspension resulting in harsh jolting ride.
8. Recommended for high-speed (over 10 mph) operation because of superior impact absorption reserve.	8. Inferior for high-speed operation because of preloading of suspension spring by interconnection of road wheels in tandem. Superior for low-speed locomobility because interconnection equalizes load.
9. No inherent track tension adjustment to compensate for road wheel deflections.	9. Substantial inherent track tensioning compensation for wheel deflections.
10. Combined types offer certain functional advantages but not advised because of increased parts inventory requirement.	

gotiate substantial terrain irregularities without transmitting undue shocks to the sprung mass and without varying excessively the load distribution or vehicle elevation. The bogie suspension behaves in the manner intended at low speeds only. For the typical vehicle, at speeds above 10 mph, the walking beam effect is lost and the entire bogie unit "pancakes," reducing the remaining spring travel to a small value.

Various types of bogie suspension systems are described in Ref. 1. Schematics of bogie suspension units are shown in Figures 10-1 and 10-2. Additional information on existing bogie suspensions is given in Appendix II.

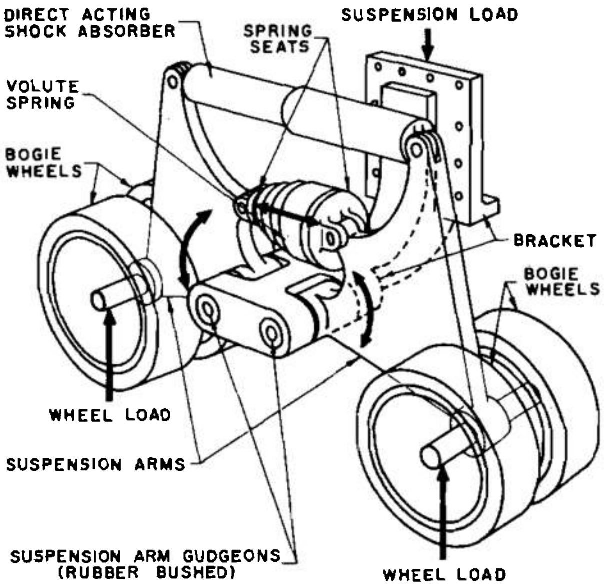

Figure 10-2. Horizontal Volute Spring Bogie

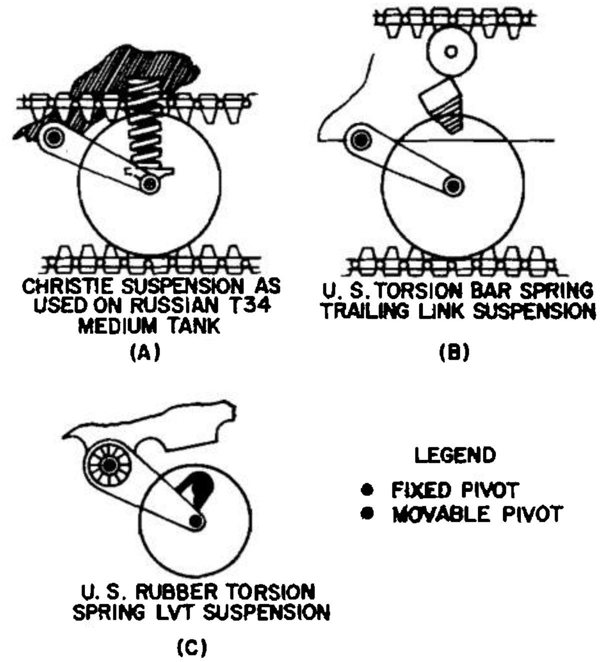

Figure 10-3. Independent Trailing-Arm Suspensions

10-2.2 INDEPENDENT SUSPENSIONS

The independent suspension system for tracked vehicles employs a separate elastic element for each road wheel. With an independent system, the verticle displacement of one road wheel does not change the spring force of the remaining suspension springs except through an influence on the sprung mass. Having greater wheel travel than the typical bogie suspension, the typical independent suspension is relatively softer. For example, the typical torsion bar independent suspension has a spring rate about one-quarter that of the volute spring bogie suspension. This is a disadvantage from the standpoint of pitching and bouncing. Damping is essential with the softer spring suspension for satisfactory performance. Table 10-1 compares the characteristics and merits of the typical independent suspension with the typical bogie suspension as applied to tracked vehicles.

Independent suspensions are classified according to the type of springs they employ (Figure 10-3). The earliest systems used coil springs while current standard vehicles use torsion bars almost exclusively. Experimental vehicles have demonstrated the great potential of the hydropneumatic suspension systems which may replace the torsion bar systems in the near future.

Various types of independent suspension systems are described in Ref. 1. A typical torsion bar suspension system is shown in Figure 10-4.

SECTION II GENERAL DESIGN CONSIDERATIONS

Fundamental design requirements applying to all suspension systems of military vehicles are presented in paragraph 9-2. Several additional design considerations are presented below.

10-3 ROAD WHEEL TRAVEL

In general, increasing the road wheel travel and at the same time lowering the spring rate results in a higher reserve of energy absorption for

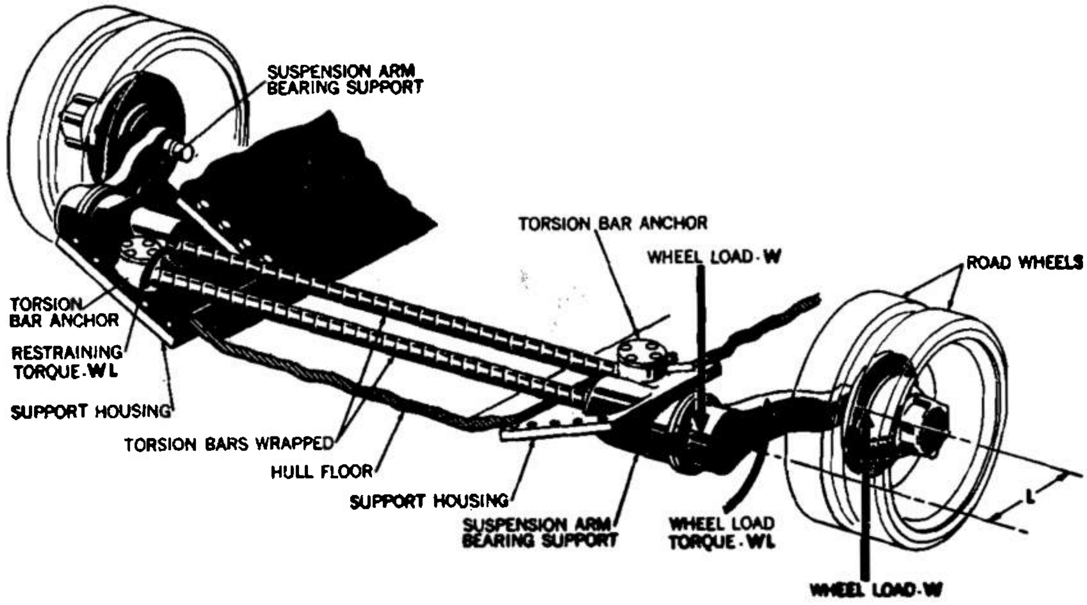

Figure 10-4. Typical Torsion Bar Suspension System

the suspension unit. This condition is favorable to increased cross country speeds. However, since there is no inherent track envelope adjustment to compensate for road wheel deflections with the conventional independent suspension systems, increasing the wheel travel in these cases makes the problem of maintaining the proper track tension more difficult.

10-4 TERRAIN-INDUCED DYNAMIC LOADS (Refs. 2, 3)

In the area of advanced analysis of dynamic loads induced by the terrain, three basic approaches have been developed. The first approach consists of writing the equations of motion of the vehicular system, and then assuming a deterministic terrain function which is representative of the real configuration of the terrain to be traversed and applying it within the system of equations. The second approach is similar, differing only in that it makes use of an actual terrain profile as the input to the equations of vehicular motion. Inasmuch as the first two methods are quite analytical, the third approach can be called a practical approach since it uses empirical methods resulting from structural analysis and experimental techniques applied to a number of actual vehicles. This latter approach is given in Appendix I.

The classical deterministic approach has been to assume a terrain profile in terms of a series of periodic functions such as sine waves. These may have any number of frequency components. After the idealized system is described, the differential equations of motion are written and solved for the desired information.

As an example of the general procedure, Figure 10-5 shows a schematic of an idealized road wheel subject to a simple sinusoidal road function. It is assumed that the tire does not leave the road.

The periodic displacement, for the present case, can be expressed as

$$x_1 = X_1 \sin \omega t. \tag{10-1}$$

where

x_1 = vertical displacement of the tire at any time, in.

X_1 = amplitude of the wave, in.

ω = frequency of the ground wave, rad/sec

t = time, sec

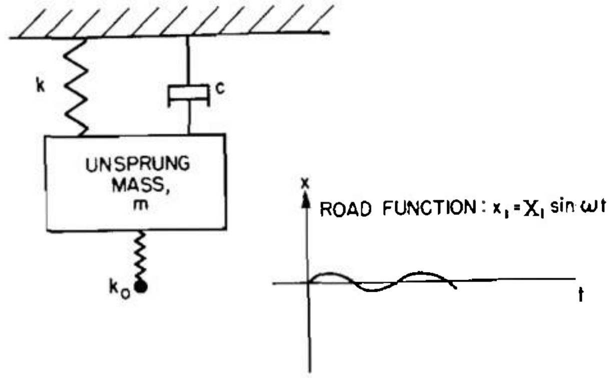

Figure 10-5. Idealized System of Wheel-Tire-Spring

The differential equation of the system (assuming viscous damping), is

$$m\ddot{x}_2 + c\dot{x}_2 + (k + k_o)x_2 = k_oX_1 \sin \omega t \quad (10\text{-}2)$$

where

m = mass (unsprung mass of the vehicle), lb-sec^2/in.
x_2 = vertical displacement of the mass m, in.
k = spring rate for the suspension spring, lb/in.
k_o = spring rate for the tire, lb/in.
c = damping constant of proportionality, lb-sec/in.

The dynamic force F_{dyn} (lb) developed at the point of tire and ground contact is expressed as

$$F_{dyn} = k_o(x_1 - x_2) \quad (10\text{-}3)$$

using the values of x_1 and x_2 from Equation 10-1 and the solution of Equation 10-2. Methods of solving the latter equation are presented in textbooks on differential equations. The solution consists of two parts: the complementary function and the particular solution. These concepts are discussed in Section VIII of this chapter.

The second approach to the problem of dynamic vehicle behavior does not utilize an assumed terrain function, but rather, uses a measured geometric terrain profile as the input to the equations of motion.

Bekker, working with Bogdanoff and Kozin, established that the terrain profile should be described by means of random wave techniques in which frequency would be considered in cycles per foot rather than on a time basis. This effort led to a power spectral density analysis of terrain (Ref. 3) and the development of equations relating the vehicle behavior to this form of input. The details of this particular approach is presented in Ref. 4; the discussion which follows is a brief outline of the method.

The device used to measure the ground profile is called a Profilometer. This apparatus was developed by the University of Michigan (Ref. 5) after a comprehensive review of the various survey methods. The Profilometer is based on a slope-integration method in which the horizontal component x and the vertical component y are to be determined. The following relationships are assumed:

$$y - y_o = \int_o^s \sin \Theta \, ds \quad (10\text{-}4)$$

$$x = \int_o^s \cos \Theta \, ds \quad (10\text{-}5)$$

where

y_o = elevation at the beginning of run
y = instantaneous elevation at any point along the run
x = horizontal distance along the run
Θ = slope angle relative to the horizontal reference at distance s from the origin

In order to solve Equations 10-4 and 10-5, it is necessary to make a continuous measurement of Θ and s. The ground roughness is averaged over the incremental distance ds which was selected as 5¼ inches in the case of the data reduction procedure selected.

A limited series of terrain profiles have been measured using the Profilometer and the power spectral density functions have been developed. The results indicate that terrain profile can be described using statistical techniques and that, although terrain may vary in roughness, the general composition of the roughness is relatively uniform. A more complete treatment of the present subject is found in Chapter 7, Section II.

The third method of determining the dynamic loads experienced by the vehicle is given in Appendix I.

10-5 TRACK DESIGN CONSIDERATIONS (Ref. 6)

Basic design and performance consideration relative to track weight ratios, power losses, flotation, traction, vibration, maintenance, and high-speed performance are presented in Section XII of this chapter. The discussion that follows concerns the supporting and tensioning components of the tracks.

The suspension system of tracked vehicles must provide components for supporting, tensioning, retaining, and laying the track. Metal track skids, steel and rubber-tread support rollers, and the tops of large diameter road wheels (flat track suspension) have been used to support the tracks. At present, three to six track support rollers are mounted ou each side of a tank to support the track from the drive sprocket to the compensating idler wheel. Mounting brackets are bolted to the side of the hull and, where desired, are provided with yokes for attaching the fixed end of shock absorbers.

Track tension in modern production tanks is maintained by two components: the compensating idler wheel and the tensioning idler roller. Compensating idler wheels of the same size and construction as the road wheels are mounted either on individual suspension arms at the front of the tank or on an extension of the front road wheel arm. Compensating wheel suspension arms are linked to the front road wheel suspension arms in such a way that the vertical motion of the road wheel produces a horizontal motion of the compensating wheel, tending to maintain tension or a stable track envelope at all times. Adjustment of the compensating wheel to provide correct track tension in the static condition is accomplished either through an adjustable connecting link or through an eccentrically bushed wheel spindle.

An additional tensioning idler roller, incorporated in some suspensions, is carried on a separate torsion bar-sprung suspension arm mounted between the road wheel and the drive sprocket. Track tension is maintained through the action of the tensioning idler torsion bar spring. The idler arm spindle and torsion bar are supported in the suspension arm support housing of the rear road wheel.

Track throwing tendencies of a specific suspension system are a function of track tension, lateral stiffness of the suspension system, and the design of the sprocket and track. The effect of track design and tension on track throwing, and the effects of track tension on the operating characteristics of tracked vehicles are covered in Section XIII of this chapter.

10-6 LUBRICATION (Ref. 7)

In general, all mating parts whose working surfaces are subject to wear or high contact forces should be provided with appropriate means for lubrication. Exceptions to this are certain surfaces, such as drive sprockets, on which lubricants are objectionable. Permanently lubricated assemblies and assemblies that require no lubrication should be used at all points where they can meet the rigorous requirements of the military environment and where their application is functionally and economically feasible. Factory sealed bearings and rubber-bushed journals are examples of such devises.

Service points for checking, replenishing, and draining of lubricants—as well as fuel, hydraulic fluid, gas, coolants, electrolyte, etc.—should be readily accessible and should incorporate features that facilitate these operations without being vulnerable to damage or contamination.

10-7 MATERIAL REQUIREMENTS (Ref. 8)

Detailed requirements for certain materials used in suspensions are given in military specifications. For materials not covered by the specifications it is suggested that material properties be taken from MIL-HDBK-5.

In addition to the design mechanical properties of the materials, some of the basic considerations are: resistance to corrosion, wear properties, and availability. Regarding critical materials, the Army Supply Manual TB-754-1 *Conservation of Material* should be consulted.

SECTION III DETERMINATION OF BASIC SUSPENSION DIMENSIONS TO MEET MOBILITY REQUIREMENTS (Refs. 9, 10)

10-8 APPARENT GROUND-CONTACT AREA

An apparent ground-contact area A_a, for tracked vehicles, can be calculated by the following equation

$$A_a = nW_tL_t \qquad (10\text{-}6)$$

where

W_t = overall track width
L_t = length of track ground contact between centerlines of extreme wheels
n = number of tracks

While this ground-contact area is used in calculating the ground pressure at zero submergence for tanks with respect to the limitation specified in AR 705-8*, it is not a satisfactory method of determining ground pressure for the purpose of mobility studies. The actual ground-contact area of tracked vehicles is a function track tension, suspension design, and soil properties. A brief discussion of these factors is given in the paragraphs which follow.

10-9 GROUND PRESSURE

10-9.1 APPARENT AVERAGE VERSUS ACTUAL GROUND PRESSURE

The apparent average ground pressure P_a, in psi, tracked vehicles is determined by the equation

$$P_a = \frac{GVW}{A_a} \qquad (10\text{-}7)$$

where

GVW = gross vehicle weight, lb
A_a = apparent ground-contact area (see Equation 10-6), sq in.

Ground pressure, used in this sense, approximates the pressure corresponding to penetration to the pin center lines and assumes uniform load distribution over the length of the track in contact with the ground.

As stated in paragraph 10-8, the actual ground pressure distribution of vehicles possessing non-rigid loading surfaces differs from the assumed uniform load distribution. One of the major factors contributing to the actual pressure distribution is the track deflection between road wheels. A detailed analysis of the problem is found in Ref. 10. It is shown that for a hypothetical case in which: (a) only vertical forces are acting on the track (vehicle at rest), (b) the track is perfectly flexible and continuous, and (c) the diameter and spacing of the wheels are such that they do not interfere with the natural deflection of the track, the approximate pressure distribution for a constant load being a function of the soil "hardness." In general, the highest ground pressure is developed at the centerlines of the road wheels but the portion of the load carried by the middle portion of the track decreases as the ground "hardness" increases. In the example, when the ground reaches a certain "hardness," the middle portion ceases to carry any of the load.

Other factors such as track tension and noncontinuity of the track contribute to the problem of determining the actual pressure distribution. Methods of analyses are presented in Refs. 10 and 11.

10-9.2 UNIFORM GROUND PRESSURE CONCEPT

The subject of the most desirable ground pressure distribution for tracked vehicles is currently under investigation. Previous studies have shown where, theoretically, a uniform ground pressure results in the lowest resistance to linear motion in soft terrain (Ref. 10). However, analyses of the steering behavior of tracked vehicles indicate that a significant reduction in the moment of turning resistance may be achieved by deviating from a uniform pressure distribution. For example, a longitudinal triangular-type pressure distribution, with the maximum pressure at the center of the contact area, results in a lower moment of turning resistance than that of a uniform distribution with otherwise similar vehicles (Ref. 10).

* See Chapter 2, pars. 2-3.1 and 2-3.3.

10-10 RATIO OF TRACK LENGTH TO TREAD (*L/T*)

The ratio of track length L to tread T for a given tracked vehicle has a significant effect on its steering behavior and power requirements. The upper and lower limits of the L/T ratio for satisfactory steering and stability are determined by the interaction of the tracks and the ground (Ref. 9). If the ratio exceeds a value of approximately 1.7, the average vehicle operating on the most favorable ground surface will not steer without excessive slippage of the outer track. In addition, ratios exceeding 1.7 impose excessive power demands on the vehicle. If, however, the L/T ratio is less than unity, steering becomes relatively unstable and excessive use of the steering system becomes necessary for average vehicle and ground conditions. In actual practice, values between 1.125 and 1.69 are used.

The correlation of physical limits of tracked vehicles with respect to suspension systems is presented in Chapter 2.

SECTION IV SUSPENSION SYSTEM NOISE AND VIBRATION

10-11 GENERAL DISCUSSION (Ref. 11)

The problem of noise and vibration, and their effects on personnel are discussed in Chapter 2.

From a tactical point of view, silence with respect to vehicular operations is clearly desirable. The importance of silencing the power plants of military vehicles has been emphasized many times in the past, however, other major sources of noise in tracked vehicles require equal consideration if they contribute significantly to the total noise. Potential sources of noise in tracked vehicles (excluding the engine-transmission complex) include the tracks, sprockets, road wheels, idlers, return rollers and structural members of the body or hull. Presently available information does not permit quantitative treatment of all of the various factors of the total noise and vibration but it is possible to analyze some of the major sources of noise and vibration related to the suspension system of tracked vehicles.

These are discussed in the following subparagraphs, while means of reducing or eliminating the undesirable or detrimental phenomena are discussed in paragraph 10-12.

10-11.1 VIBRATION RESULTING FROM CHORDAL ACTION OF TRACKS (Ref. 2)

Since the conventional track consists of a number of rigid links and pinned joints, it is subjected to a polygonal or chordal action as it passes around a sprocket. As shown in Figure 10-6, the roller A is accelerated from V_1 to V_2 when the sprocket rotates through angle Θ. Since r_2 is greater than r_1, V_2 is greater than V_1. A similar, but reverse, change in the speed of the link occurs when it leaves the sprockets. This action tends to cause vibration of the tracks. A more detailed treatment of the forces involved is given in Section XII of this chapter.

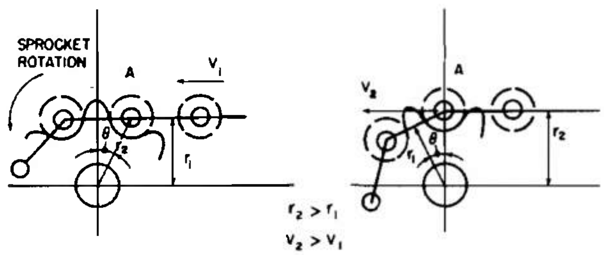

Figure 10-6. Chordal Action

10-11.2 IMPACT OF THE TRACK ON ROTATING SUSPENSION MEMBERS (Ref. 13)

Since the rigid linked track does not form a smooth curve when it passes around a sprocket, road wheel, or idler; impacts are unavoidable as the links engage the wheels. A schematic of a track wrapped around a rotating suspension member is shown in Figure 10-7. If we assume links B and C are in contact with the rotating member

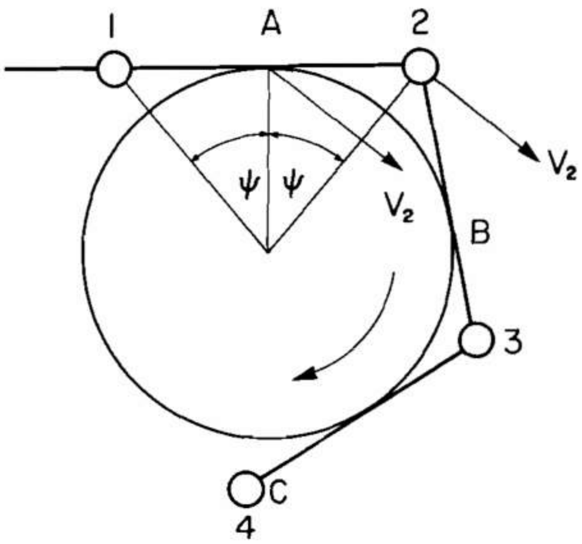

Figure 10-7. Impact of Track on Rotating Member

and link A is at an impending contact position, joint 2 will have tangential velocity V_2. Furthermore, link A can be assumed to be moving so as to remain parallel to its lead-in horizontal position. In this case, the center of the link will have a vertical (toward the center of the wheel) component of velocity of $V_2 \sin \psi$, where ψ is the angle subtended at the center by half a pitch. During the operation of the system, the vertical component at the center of the link is reduced to zero by impact between the members. The point on the link at which the impact occurs depends to some extent on the configuration of the rotating member. In the case of a road wheel or idler and a wrapped track, the middle of the link will usually impact; whereas in the case of a sprocket, the end often is the impact surface.

In the case of a leading road wheel of small diameter and a long pitched track, the situation may be somewhat different since no links are wrapped around the wheel. As shown schematically in Figure 10-8, link A is held to the ground by the road wheel, while the wheel has a component of velocity of $V \sin \Theta$ perpendicular to link B at the point of contact with the wheel, where Θ is the angle of attack of link B with the ground.

These various impacts and the inherent fluctuation in the longitudinal velocity of the track re-

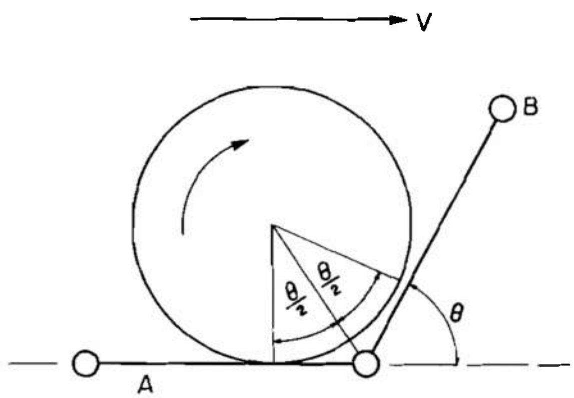

Figure 10-8. Impact of Road Wheel on Link

sult in vibration and a fluctuating track tension superimposed on any steady tension. The kinetic energy losses resulting from these phenomena are discussed in Section XII of this chapter.

10-11.3 IMPACT OF THE TRACK LINKS ON THE TERRAIN (Ref. 14)

Another type of impacting occurs when a rigid linked tracked vehicle is in motion. This occurs when the track link initially contacts the ground, normally at the leading road wheel as the vehicle moves forward. Figure 10-9, diagrammatically shows a road wheel rolling on link A, while the adjacent link B slopes up at an angle of 2Θ. At the instant of contact of link B, the angles formed by the points of contact, the center of the wheel, and the common hinge pin will be equal to Θ.

If the distance from the hinge to the point of contact of either link is denoted as x, the velocity of the center of the wheel (and of the vehicle) will be equal to dx/dt. Since the inclination of link B is 2Θ, its angular velocity will be $2d\Theta/dt$. To determine $d\Theta/dt$ in terms of the velocity of the vehicle V, the radius of the road wheel R, and Θ the following analysis applies

$$\tan \Theta = \frac{x}{R} \tag{10-8}$$

$$\frac{d}{dt}(\tan \Theta) = \frac{d}{dt}\left(\frac{x}{R}\right) \tag{10-9}$$

$$\frac{d}{d\Theta}(\tan \Theta)\frac{d\Theta}{dt} = \frac{1}{R}\frac{dx}{dt} \tag{10-10}$$

$$\frac{d\Theta}{dt} = \frac{V \cos^2 \Theta}{R}, \text{ rad/sec} \tag{10-11}$$

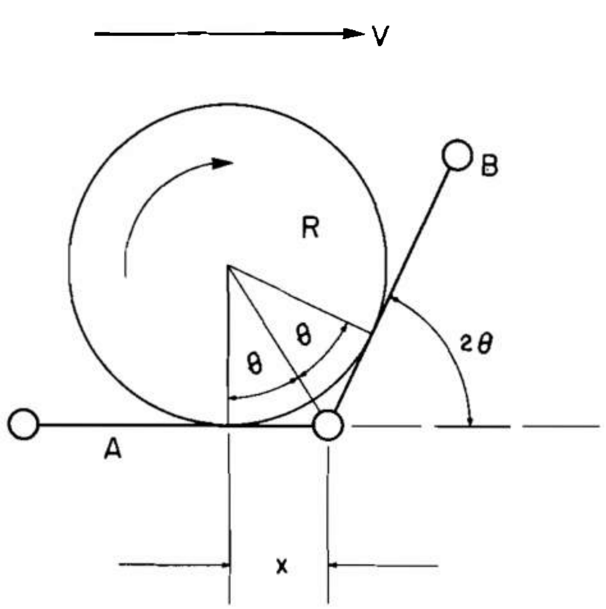

Figure 10-9. Impact of Link on Terrain

At the instant link B contacts the ground, 2Θ and Θ becomes zero; then $\cos^2\Theta = 1$, and from Equation 10-11

$$\frac{d\Theta}{dt} = \frac{V}{R}, \text{ rad/sec} \qquad (10\text{-}12)$$

If the pitch of the link is l (ft), the velocity of its center V_c will be $(l/2)(2d\Theta/dt)$, and hence

$$V_c = \frac{lV}{R}, \text{ fps} \qquad (10\text{-}13)$$

As shown by Equation 10-13, the velocity at which the center of link B impacts the terrain is not dependent on the angle of attack of the track but varies directly as the pitch of the track and the speed of the vehicle, and inversely as the radius of the leading road wheel.

The power lost as a result of the above impact phenomena can be calculated as follows. Assume the mass of the link is concentrated at its center. The kinetic energy per link T can be expressed as:

$$T = \frac{w}{2g}\left(\frac{lV}{R}\right)^2, \text{ ft-lb} \qquad (10\text{-}14)$$

where

w = weight per link, lb
g = acceleration due to gravity, fps^2
$\frac{lV}{R}$ = velocity at center of link (see Eq. 10-13), fps

Since the number of links per sec for 2 tracks is $2V/l$, the power lost, P_L, can be written as

$$P_L = \frac{WlV^3}{550gR^2}, \text{ HP} \qquad (10\text{-}15)$$

10-11.4 BUMP STOPS (Ref. 15)

Another source of noise and vibration associated with the suspension system are the bump stops normally included in the system. With a torsion bar suspension, a volute bumper spring or a solid bumper stop is attached to the hull over each suspension arm to limit and cushion extreme arm displacement. Impacts on the bump stops are transmitted directly to the hull causing noise and vibration.

10-12 BASIC METHODS OF REDUCING OR ELIMINATING NOISE AND VIBRATION

10-12.1 ISOLATION (Refs. 16, 17)

Isolation, as it is applied in the context of vibration theory, is a reduction in the capacity of a system to respond to an excitation, attained by the use of resilient supporting members. In steady-state forced vibration, isolation is expressed quantitatively as the complement of transmissibility. *Transmissibility* is defined as the nondimensional ratio of the response amplitude of a system in steady-state forced vibration to the excitation amplitude. The ratio may be an expression of forces, displacements, velocities, or accelerations.

A comprehensive theoretical treatment of vibration and shock isolation is presented in Refs. 16 and 17. The present discussion is limited to some of the basis practical aspects of isolation and isolators with respect to military vehicles.

In conventional vehicles, the suspension and terrain induced excitation may effect each of three different elastic systems separately. The first system is the sprung mass of the vehicle; the second system is composed of the unsprung mass; and the third system is formed by the local body structure which can experience vibration at the natural frequencies of the various structural members.

The natural frequencies of these systems are primary factors in determining the design requirements and the applicability of isolators.

The analytical expression of transmissibility based on the natural frequency of the supported (isolated) body and the frequency of the disturbing force (Ref. 16) shows that for an isolator to perform its function, the natural frequency of the supported body must be small in comparison to the frequency of the disturbing force. This requirement frequently offers difficulties when the actual design of an isolator is attempted and very low natural frequencies are required. Since the natural frequency ω_n is equal to $\sqrt{g/\delta_s}$, the statical deflection δ_s necessary for small values of ω_n is frequently beyond the practicable range. With regard to the first system, the natural frequency of the sprung mass of military vehicles usually is between 1 and 4 cps. It is not practicable to use isolators with a lower natural frequency than the natural frequency of the main elastic support system (suspension). Therefore, isolators are not effective with respect to the overall motion of the vehicle body.

The natural frequency of the unsprung mass, second system, usually is between 10 and 20 cps. In general, it is not feasible to design isolators to alleviate vibration from this source owing to the relative low frequencies. A major problem in the application of isolators is to avoid resonance with this source of vibration.

The major function of isolators, with respect to military vehicles, concerns the third system—the local body structure. The natural frequencies of the structural members of the vehicle body usually are higher than the frequencies associated with the elastic suspension system. A principal function of isolators is to alleviate the effects of the resonance conditions that would develop when the natural frequencies of equipment coincide with the natural frequencies of supporting structural members.

The isolation requirements and the design or selection of isolators for vehicular applications can be determined more effectively if the principal natural frequencies of the vehicle are known. The natural frequency of the unsprung mass is of particular importance because: (a) the vibration induced by this source may be of significant severity and, (b) it borders on the frequency range typical of optimum isolator design.

In the application of isolators to equipment within a vehicle, the resulting natural frequencies of the equipment in all modes of vibration should be different than the natural frequency of the unsprung mass. It is necessary to consider all natural modes of vibration, both coupled and uncoupled, and then to design or select isolators accordingly. If the natural frequency of the unsprung mass is as high as 20 cps, then the natural frequencies of the isolated equipment should have a minimum value of 25 cps. If their natural modes of vibration are substantially uncoupled, this goal is reasonably attainable; otherwise the maximum natural frequency becomes unreasonably high when the minimum of 25 cps is attained.

It is desirable to incorporate some damping in the isolation system where the excitation covers a wide range of frequencies and may have random and/or transient properties. These conditions are common in the military environment.

If the inherent damping of the load carrying elements is insufficient to meet the specifications, separate damping elements are included in parallel with the elastic members.

A comprehensive treatment of the application and design of isolators is presented in Ref. 6.

10-12.2 DAMPING (Ref. 18)

The present discussion of damping is limited to vibration and noise control by applied surface treatment of structural members. Damping associated with the main suspension system of a vehicle is presented in Section VIII of this chapter.

As previously stated, the structural members of the body of a vehicle can be a major source of vibration and noise. These phenomena frequently can be controlled by the application of a surface damping treatment. The applied damping treatment can increase the net damping capacity of the structural material by an order of magnitude.

Materials most frequently used for surface damping treatment are (a) mastic deadeners, (b) high-polymer mastic deadeners, (c) asphalted felt, and (d) fibrous blankets (glass, cotton, flax, jute, kapok, wood, mineral wool, etc.).

Because the movement of the structural member is communicated to the vibration-damping material, the inherent effects of hysteresis in the former material are supplemented by the much higher hysteresis of the deadener material or by its dissipative effects of viscosity, fiber flexure, and internal friction, as well as the friction effects at the interface.

A comprehensive presentation of the subject of vibration and noise control by applied damping treatments is found in Ref. 16.

10-12.3 STRUCTURAL AND DESIGN MODIFICATION (Ref. 19)

Another approach to the problem of reducing or eliminating noise and vibration of military vehicles consists of structural and/or design modification.

If the forcing function is of the discrete spectrum type, it may be possible to shift the natural frequencies of the structure so that coincidence with the forcing frequencies is avoided. This may involve stiffening or weakening the structure. Another method of changing the natural frequencies is to add a spring and mass to the structure and, consequently, add another degree-of-freedom.

If the forcing function is of the discrete and/or continuous spectrum type, the addition of damping will reduce the response at the resonant frequencies.

Methods of introducing damping through structural modification include: (a) replacement of the original material by another having higher internal dampening, (b) the addition of surface damping materials, and (c) the addition of slippage and, hence friction, between members.

Since one of the principal sources of secondary disturbances and vibrations is the unsprung mass, and since wheel dance is one of the major sources of these disturbances, the amount of noise and vibration can be altered by varying factors such as the major springing system rate, the road wheel tire spring rate, the mass of the road wheel, and the damper characteristic. These factors are discussed in Sections VII and VIII of this chapter.

SECTION V ROAD WHEELS AND HUBS

10-13 DESIGN CONSIDERATIONS

10-13.1 NUMBER OF WHEELS REQUIRED

The number of road wheels as a parameter in the soil-vehicle relationship is discussed in Part Two—Land Locomotion. The present discussion concerns some of the mechanical aspects of the problem.

The number of wheels required varies with the weight of the vehicle. For example, the Main Battle Tank, M60 (102,000 lb) utilizes six dual 26 × 6 road wheels on each side, while the Heavy Tank, T43 (120,000 lb) utilizes seven dual 26 × 6 road wheels on each side. Thus, each dual wheel assembly supports approximately 8,500 pounds if a uniform load distribution is assumed. However, even under static conditions, the loading of the road wheels of a conventional vehicle may not be uniform owing to the location of the center of gravity. Further changes in loading occur as a result of the dynamic conditions experienced by a vehicle.

Increasing the number of road wheels to compensate for increase in vehicle weight sometimes results in a decrease in suspension efficiency. However, a decrease in efficiency may be outweighed by advantages obtained in using standardized interchangeable components. Currently, two standard sizes of road wheels are in use: 26 × 6, usually employed on medium and heavy tanks, and 25½ × 4½, used on light tanks.

Analytical studies and empirical data show that rolling resistance for a tracked vehicle decreases as the number of road wheels increases (Ref. 20). However, since the number of wheels is limited by track length, the specification of a large number of wheels dictates that either small wheels or overlapping wheels must be used. This problem is analyzed in Ref. 10.

10-13.2 DIAMETER AND WIDTH

Two fundamental design considerations related to the diameter of the road wheel are: (a) rolling

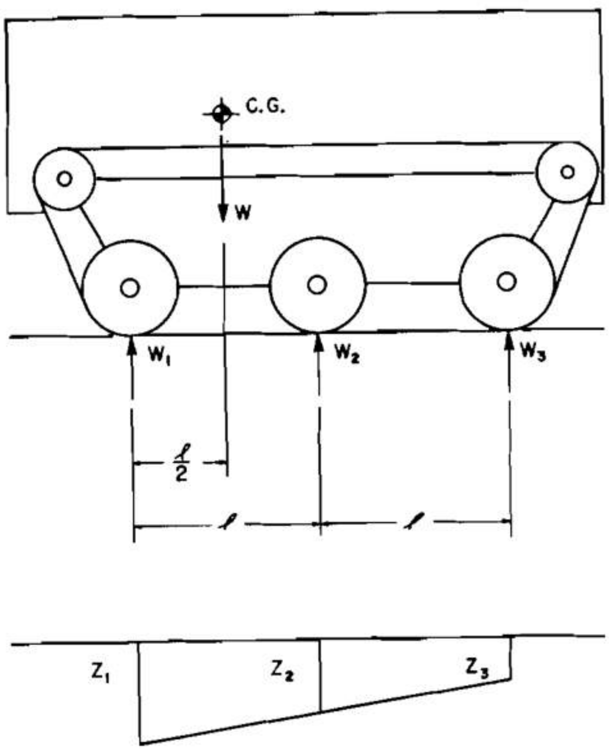

Figure 10-10. Road Wheel Load Diagram for Rigid Suspension Vehicle

resistance, and (b) power lost as a result of the track link striking the ground.

A number of investigators have shown analytically that sinkage and rolling resistance in soils are related to the diameter of the road wheels (Ref. 21). In general, an increase in road wheel diameter (other conditions unchanged) results in a decreased sinkage and rolling resistance.

The impacting of track links on the terrain is discussed in paragraph 10-11.3. Equation 10-15 of paragraph 10-11.3 shows that the power lost through the impacting varies inversely as the second power of the radius of the leading road wheel.

The width of road wheels enters into the general design considerations from the standpoint of the track and the wheels themselves. Since the track is wider than the wheels acting on it, deflection and transverse load distribution of the links must be considered. In addition, the bearing loads on the track and the wheels must be considered. These subjects are discussed in Section XII of this chapter.

10-13.3 SPACING

Normally, the spacing of the road wheels of tracked vehicles is nonuniform within limits imposed by the number of wheels, the diameter of the wheels, and the length of the track. As a road wheel travels over a conventional track, it encounters a discontinuity in the running surface at each joint of the track. Noise, vibration, and wear are minimized if the road wheels do not encounter the track joints simultaneously. To accomplish this, the spacing of the wheels is correlated with the pitch of the track shoes.

10-13.4 LOAD CALCULATIONS

10-13.4.1 Static Loads (Ref. 22)

The load acting on each road wheel of a tracked vehicle can be determined by the methods that follow.

For the initial calculation, consider a vehicle having three equidistant road wheels on each side, a rigid suspension, and a loose track (as shown in Figure 10-10). The equations of equilibrium can be written as

$$W_1 + W_2 + W_3 - W = 0 \qquad (10\text{-}16)$$

$$W\frac{l}{2} - lW_2 - 2lW_3 = 0 \qquad (10\text{-}17)$$

Since the problem is statically indeterminate an additional assumption is required.

This assumption or condition concerns the sinkage of the wheels. Since the suspension is rigid, the sinkage will be such that the centers of the wheels will remain on a straight line. Furthermore, since the center of gravity is displaced to the left of the center wheel, the left wheel will tend to sink more than the right wheel as shown in Figure 10-10. This condition can be expressed as

$$Z_1 - Z_2 = Z_2 - Z_3 \qquad (10\text{-}18)$$

As was discussed in Chapter 4, the sinkage Z_i, in., for each wheel (assuming the area of the ground-contact of each wheel is the same) is given by

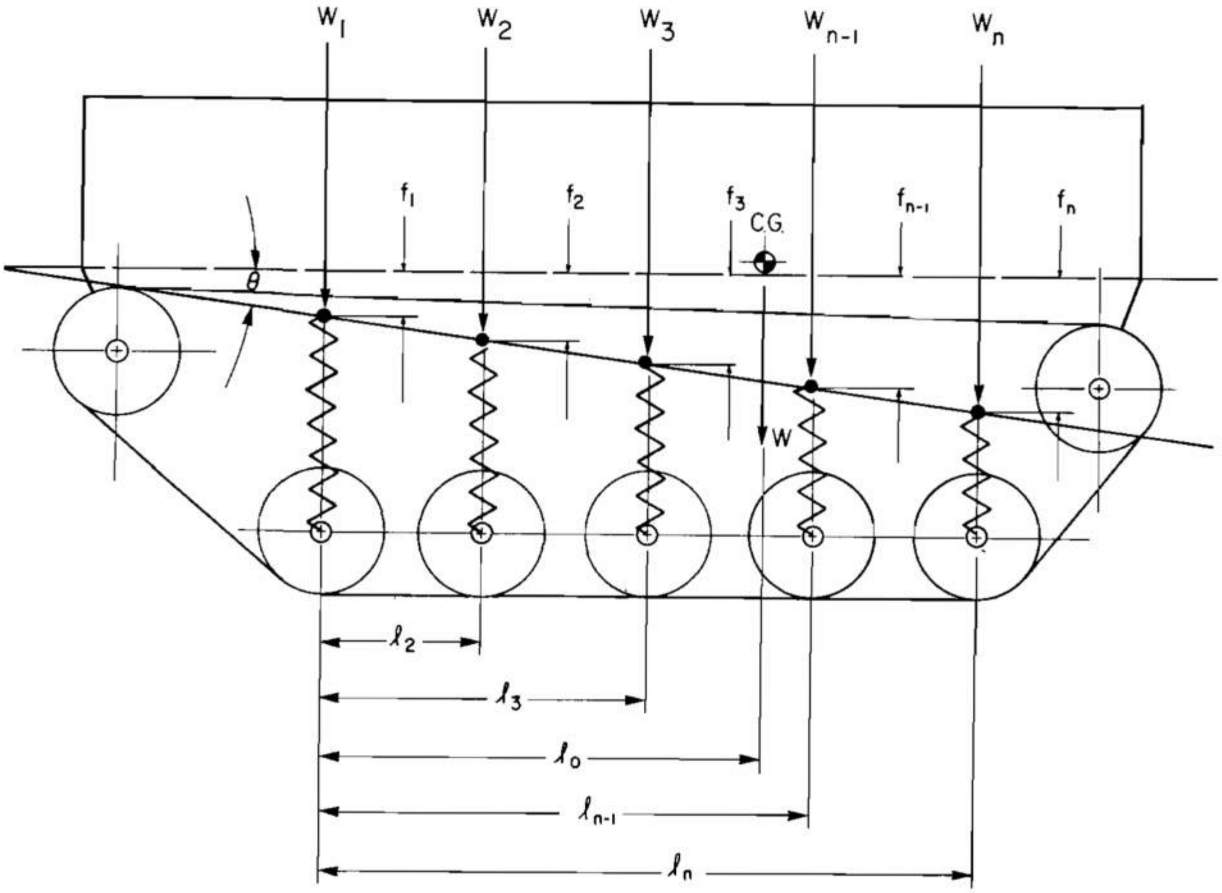

Figure 10-11. Road Wheel Load Diagram for Independently Sprung Vehicle

$$Z_1 = \left(\frac{W_1}{Ak}\right)^{1/n} \tag{10-19}$$

$$Z_2 = \left(\frac{W_2}{Ak}\right)^{1/n} \tag{10-20}$$

$$Z_3 = \left(\frac{W_3}{Ak}\right)^{1/n} \tag{10-21}$$

where

W_i = load acting on each wheel, lb
A = area of the ground-contact of each wheel, sq in.
k = soil constant (see Chapter 4), lb/in.$^{n+2}$

By using the case of Bernstein's $n = \frac{1}{2}$ and Equations 10-18 to 10-21, the following equation can be written

$$W_3^2 = 2W_2^2 - W_1^2 \tag{10-22}$$

The solution of Equations 10-16, 10-17, and 10-22, yields the following values for the three unknown loads

$$W_1 = 0.556\ W \tag{10-23}$$

$$W_2 = 0.390\ W \tag{10-24}$$

$$W_3 = 0.054\ W \tag{10-25}$$

These equations show the influence of the location of the center of gravity on the distribution of the wheel loading.

The method described above may be extended to evaluate the loads on the road wheels of vehicles possessing spring suspension systems. The discussion which follows is limited to individually suspended wheels; a thorough discussion of vehicles having nonindependent suspensions is found in Ref. 10.

Figure 10-11 represents a vehicle possessing five independently sprung wheels on each side. If the deflection, in inches, of the individual springs is denoted as f_1, f_2, f_3 . . . . f_n, the following n equations are obtained.

$$W_1 = cf_1 \tag{10-26}$$

$$W_2 = cf_2 \tag{10-27}$$

$$W_3 = cf_3 \tag{10-28}$$

$$\vdots$$

$$W_n = cf_n \tag{10-29}$$

where

W_1 . . . . W_n = load acting on each wheel, lb

c = rate of individual springs (assumed equal), lb/in.

The deflection of the springs can be expressed by the following (n-1) equations

$$f_2 = f_1 + l_2 \tan \Theta \tag{10-30}$$

$$f_3 = f_1 + l_3 \tan \Theta \tag{10-31}$$

$$\vdots$$

$$f_n = f_1 + l_n \tan \Theta \tag{10-32}$$

Finally, two more equations can be written

$$W_1 + W_2 + W_3 + \ldots . W_n = W \tag{10-33}$$

$$W_2 l_2 + W_3 l_3 + \ldots . W_n l_n = W l_o \tag{10-34}$$

The total number of equations will be $(2n + 1)$, and the total number of unknowns will also be $(2n + 1)$, i.e., n forces, W; n deflections, f; and the angle Θ. This permits the determination of the unknowns by algebraic manipulation.

10-13.4.2 Dynamic Loads

Dynamic loading acting on the road wheels is extremely complex owing to the nature of the input forces. Dynamic loading of the road wheels and related components results from vehicle acceleration, braking, turning maneuvers, gun recoil, ballistic impact, blast, or the terrain. Two aspects of dynamic loading are discussed in Section II of Chapter 2, and in Section II of this chapter.

10-13.4.3 External Forces

Gun recoil loads are transmitted through the gun mount and vehicle structure into the earth. As part of the vehicle structure the road wheels are subjected to the dynamic recoil loads. The gun recoil loads are obtained by established methods of calculation and by field testing.

The investigation should be conducted for different angles of elevation and azimuth. For the calculated loading, short peaks or spikes in the recoil data of less than 3 milliseconds duration have no appreciable effect and can be neglected (Ref. 8).

10-14 TIRE DESIGN (Ref. 23)

Bogie and road wheel tires are constructed of solid rubber vulcanized to a steel wheel. Considerable testing and development has gone into the design and construction of these tires. Many rubber compounds, both natural and synthetic, have been tried in an effort to increase durability. A variety of tire width-and-thickness combinations have been used, together with tread designs such as annular grooves. As a result of this extensive development program, three important design requirements have been established:

(a) Tire width should not be more than seven inches. If more width is required, two tire sections should be used instead of one. Lateral shearing forces set up at the steel base by the extreme distortion of the rubber load are thus reduced.

(b) Tire diameter should be as large as practicable because the load-carrying capacity of tires of the same width varies approximately as D to the 2.25 power, where D is outside diameter. This factor was established primarily as a function of the heat build-up in a tire under rated load and speed.

(c) Tread thickness should be limited to an amount necessary to overcome the danger of blowout where heat developed by flexing may actually break down the rubber in the center of the tread section. At present, the

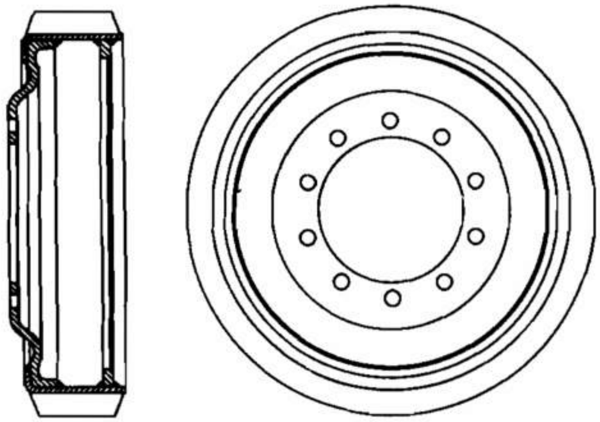

Figure 10-12. 26-inch Dual Road Wheel Assembly (Medium and Heavy Tank)

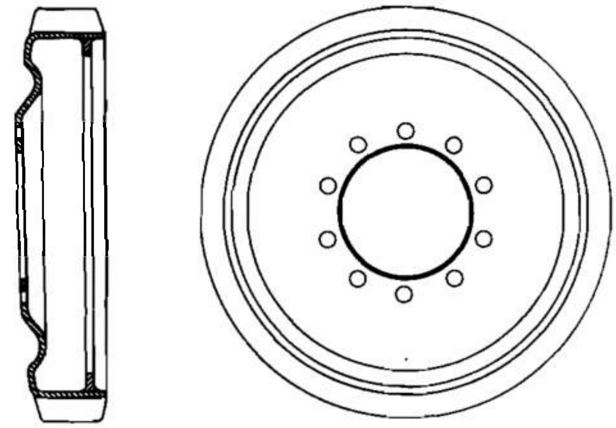

Figure 10-13. 25-1/2-inch Road Wheel Assembly (Light Tank)

tire thickness of medium and heavy tanks is 1-5/8 inches and of light tanks, 1-11/32 inches.

Other factors affecting tire life are type of suspension, and tank mass and speed. Tanks with soft springs and independent suspensions affect tire life less than tanks with bogie-type suspension and stiff springs with high friction linkages. Soft, frictionless springs reduce peak impact loads on tires, particularly on a solid mass of rubber vulcanized to a steel base.

Mass of the tank affects tire life in that the heavier the tank the higher the inertia forces acting when the mass oscillates over rough terrain. Since kinetic energy varies directly as the mass and as the square of the speed, forces are set up in component parts of the suspension which cannot be estimated unless measured dynamically, especially on track-laying vehicles operating on grades. During such slope operations, test data show that an appreciable change in load distribution takes place on front and rear road wheels while the load on the center wheels remains approximately constant.

10-15 WHEEL DISC DESIGN (Refs. 24, 25, 26)

10-15.1 CHARACTERISTICS

Basic design objectives in the development of a road wheel are: (a) high strength-to-weight ratio, (b) rigidity, (c) minimum weight, (d) reliability, and (e) adaptability to manufacture. These objectives have been the bases for the evolutionary process that has produced the various types of road wheels.

The approach to road wheel design has been largely empirical. The typical road wheel of tracked vehicles at the beginning of World War II was of welded steel construction. It had an integral hub and rim, and a single flat disc with lateral ribs or gussets. The disc was cut out between the ribs to reduce weight. The tire was molded to a separate steel band which was pressed onto the wheel. Shortly thereafter, the first designs were modified to a type that was better adapted to mass production techniques and was less expensive to make. The modified wheel used a lighter gage stock and was formed by using two symmetrical stampings for the disc; these were riveted to each other and to the rim. The tire was attached as before.

The next type of road wheel, developed during World War II and used on medium and heavy tanks, is shown in Figure 10-12. In this design, the pressed on rim was eliminated and the tire was molded directly to the wheel. A wheel used extensively on light tanks during World War II and in Korea is shown in Figure 10-13. While this wheel is somewhat similar to the wheel shown in Figure 10-12, it differs in one important respect in that it has a one-piece disc and rim rather than

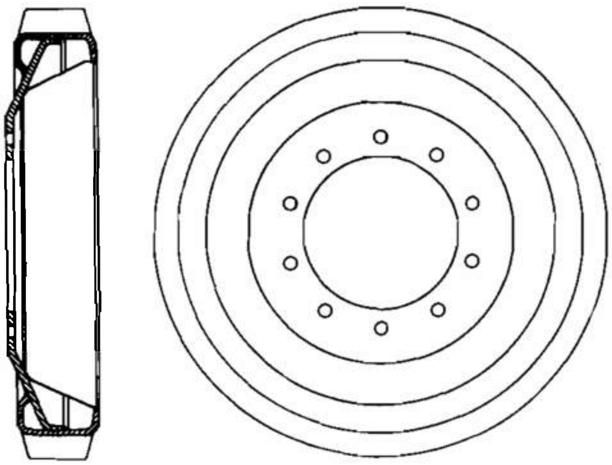

Figure 10-14. 28-inch Track Wheel

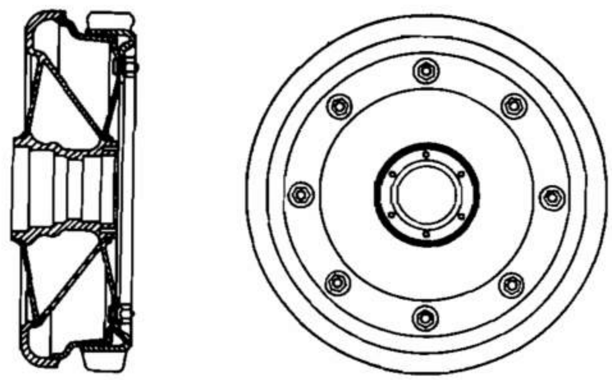

Figure 10-15. 19-inch Bogie Wheel for Amphibious Vehicle

the two-piece welded construction. The principal advantage in this type of construction is its economy of manufacture made possible by the lighter service requirement permitting the use of lighter gage low alloy steel.

A later design is presented in Figure 10-14. This wheel represents some of the basic features of present-day one-piece track wheels. It has a tapered disc which supports the rim at its center. This not only provides the rigidity inherent in the cone shape of the disc, but it also supports the rim at its optimum point—its center. After assembly to the disc, the rim flange is turned inward forming the reinforcing rim.

Figure 10-15 shows a wheel designed for amphibious tracked vehicles. Wheels for amphibious vehicles are designed so they provide a significant flotation factor. This is accomplished by incorporating water tight chambers into the design, or chambers filled with styrofoam.

Figure 10-16 shows a steel wheel design for a flat track suspension system. Figure 10-17 shows a recent aluminum wheel design for the same type of suspension.

In keeping with the objective of high strength/weight ratios for road wheels, forged aluminum road wheels have been developed. Forged aluminum (2014-T6) road wheels have given excellent service on the M60 Main Battle Tank and are the standard units on these vehicles. Currently, forged aluminum road wheels are under development for the New Main Battle Tank. The requirements for this wheel included 44,000-pound radial load, 32,-000-pound side load and a minimum life of 4,000 miles in a 30-inch diameter wheel weighing 95 pounds or less, including rubber and metallized wear surface.

The use of aluminum alloy for road wheels has introduced the need for additional wear protection in the track guide area. Several approaches to this problem have been considered. One method of achieving adequate wear surfaces on the wheels is to provide replaceable steel wear rings as part of the wheel assembly. These rings are positioned to contact the track center guides during operation of the vehicle. Another method consists of extending the tires so that they contact the center guides. An advanced technique consists of coating the wear surfaces with a steel alloy spray deposit. Spray-applied Metcoloy No. 2 has proven suitable for this application.

10-15.2 TESTING PROCEDURES AND STRESS ANALYSIS

In road wheel design, as in many other fields, laboratory tests are invaluable in the development of particular designs. Four basic techniques are widely used to study and evaluate road wheels in the laboratory, namely: (a) dynamic destructive fatigue tests, (b) impact tests, (c) static load-deflection tests, and (d) strain distribution tests.

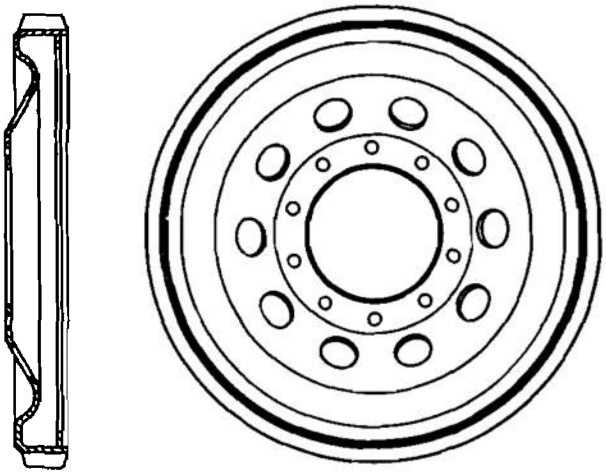

Figure 10-16. 31-inch Track Wheel

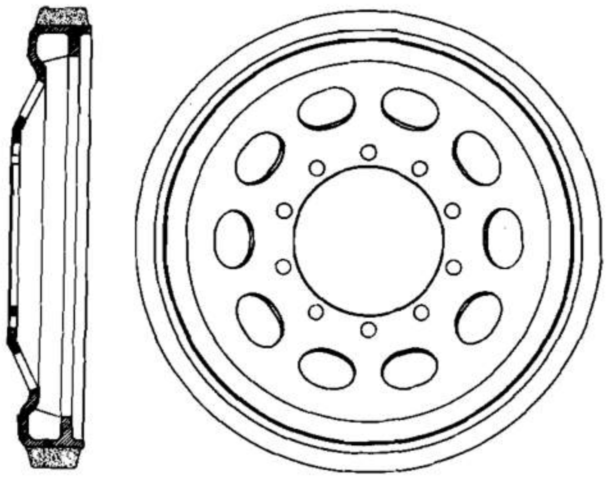

Figure 10-17. Recent 33-inch Aluminum Wheel Design

The dynamic destructive fatigue tests are conducted by loading a wheel in a prescribed manner while the wheel is rotating. The fluctuating load produces fatigue failure in known critical areas of the wheel. The test is excellent for obtaining comparative endurance data on several designs including field proven units.

Impact tests are used to evaluate shear strength of welds or rivets or rim stiffness. Static load tests using radial, lateral, or combined radial and lateral loading are frequently applied for the purpose of comparing deflection and set on various wheels under given loads. SR-4 strain gages are frequently utilized for this type of testing.

Strain distribution tests employing brittle lacquer (stress-coat) are being used to an increasing extent in appraising wheel designs.

10-16 ROAD WHEEL HUB DESIGN (Refs. 27, 28)

10-16.1 BEARING CONSIDERATIONS

The current wheel hub design incorporates tapered-roller bearings which are best suited for the heavy radial and thrust loads developed during vehicular operations. The design and selection procedures for these bearings are well covered in standard reference books as well as technical catalogs published by bearing manufacturers. The present discussion outlines some of the fundamental design considerations regarding rolling-contact bearings.

The bearings of road wheel hubs are subjected to (a) a normal static load, (b) a basic dynamic load, and (c) variable terrain-induced loads (including side loading). Static loads are those developed in a bearing at rest. The limit to the static load which a given bearing can carry is determined by permanent deformation. Bearings are given a basic static load rating defined as that load which will not deform the elements at any one point of contact more than 0.0001 times the diameter of the rolling elements. If a load higher than this is imposed on the bearing while it is rotating, the deformation is distributed evenly and no practical impairment occurs until the deformations become quite large. However, under these conditions fatigue life is relatively short.

The basic dynamic load determines the life expectancy of a rolling-contact bearing. The spherical elements of ball bearings have point contact and the cylindrical elements of roller bearings have line contact—until they are deformed under load. The fluctuating distortion eventually leads to fatigue of the materials. For standard steel rolling-contact bearings, the expected life is inversely proportional to the cube of the load. The number of stress cycles to fatigue of the steel is the life expectancy of a bearing. Defects in the bearings, mounting, lubrication, and operating conditions may accelerate failure and shorten the life.

Speed of any bearing is measured in surface feet per minute, with the radius figured from the center of the shaft to the center of the ball or roller. Bearing fatigue life depends on the number of stress cycles, as well as the magnitude of the stresses. Doubling the speed reduces the life by one-half. Although, standard formulas are available for calculating the load to be used for bearing life when the load varies rather uniformly and shock multiplication factors have been developed for these conditions, these formulas are not adequate for the widely varying loading conditions experienced by road wheel bearings. For the purposes of design, the specifications of bearings in existing relevant military components as well as technical information from bearing manufacturers are valuable references.

10-16.2 BEARING SEALS

Bearing seals are applied to road wheels, track support rollers, and torsion bar support arm spindles in an attempt to exclude dirt and water from bearing surfaces and retain the lubricant. These seals are of extreme importance to tanks in mud and water, or in high dust concentrations. Simple automotive-type seals have proved inadequate for the rugged service encountered in tank use. Hence, labyrinth arrangements have been provided outside the seal zone to inhibit the entrance of mud, water, and dust.

Bearing seals are of two types (a) circumferential (lip type) and (b) face type. Circumferential seals are generally of the double-lip type, incorporating a leather lip held in position by a "garter spring" and either a leather or neoprene second lip. In this type seal, the lips are installed toward the outside, i.e., away from the bearing, so that lubricant can pass through the bearing and flush the sealed zone.

Face-type seals used in tank suspensions consist of two lapped and matched rings rotating in spring contact in a plane perpendicular to the axle. The rings may both be hardened steel, each a dissimilar metal, or one of metal and the other of some other substance such as leather, carbon, or cork. These seals have resulted in longer life and permit the use of lighter lubricants. Face-type seals appear to adjust themselves more readily to load deflections than do lip-type seals. Face-type seals are almost developed to the point that suspension components can be sealed for the life of the vehicle and periodic maintenance eliminated.

SECTION VI ROAD WHEEL ARMS (Refs. 29, 30)

10-17 GENERAL DISCUSSION

A typical road wheel arm is shown in Figure 10-4. Current vehicles utilize solid forged road wheel arms. The arm is attached rigidly to a spindle which is carried in roller bearings in a cast support housing bolted to the bottom of the hull. The road wheel suspension arm is usually press-fitted on the hollow, externally serrated spindle (Fig. 10-4). Forged one-piece arm and spindle units are also in production. The spindle is internally splined to mate with the torsion bar spring.

As is apparent from Figure 10-4, the wheel load imposed on the road wheel arm induces both bending and torsional stresses in the member. In addition, these stresses vary in magnitude with time. Hence, the design of road wheel arms should be based on combined fatigue stress analyses. A comprehensive treatment of stress due to combined effects is found in Reference 30. In addition to the states of stress developed in a member, many other factors influence the value of the fatigue strength. Some of these are (a) surface treatment, (b) chemical composition, (c) localized stresses, (d) methods of fabrication, (e) stress concentration, (f) rate of application of stress, (g) stress history, and (h) environmental conditions and effects (Ref. 30).

SECTION VII VEHICLE SPRINGING

10-18 ELASTIC SUPPORT SYSTEM CHARACTERISTICS (Ref. 31)

10-18.1 LINEAR SPRING CHARACTERISTICS

The force-deflection characteristic of a *linear spring* is expressed as

$$F_S(\delta) = k\delta \tag{10-35}$$

where

$F_S(\delta)$ = force, lb (developed by the spring as a function of the deflection δ, in.)

k = spring rate, lb/in.

The maximum acceleration of the spring-mass system is

$$\ddot{x}_{max} = \omega_n^2 \delta_{max},\ \text{in./sec}^2 \tag{10-36}$$

where

$\omega_n = \sqrt{k/m}$ natural frequency of the spring-mass system, rad/sec

δ_{max} = maximum initial deflection, in.

10-18.2 NONLINEAR SPRING CHARACTERISTICS

10-18.2.1 Hardening Spring

One type of nonlinear spring may be characterized by tangent elasticity, in which case, the slope of the curve representing spring force vs deflection increases with increasing deflection. A representative curve having this characteristic is shown in Figure 10-18(A). The equation of a representative curve for a spring having a hardening characteristic is

$$F_S(\delta) = \frac{2kd}{\pi} \tan\left(\frac{\pi\delta}{2d}\right) \tag{10-37}$$

where

k = initial slope of the curve, lb/in.

d = value of the limiting deflection, at which point a vertical asymptote exists (see Fig. 10-18(A)), in.

Other terms are as defined for Eq. 10-35.

The relationship between maximum acceleration and maximum deflection for the hardening spring can be expressed as

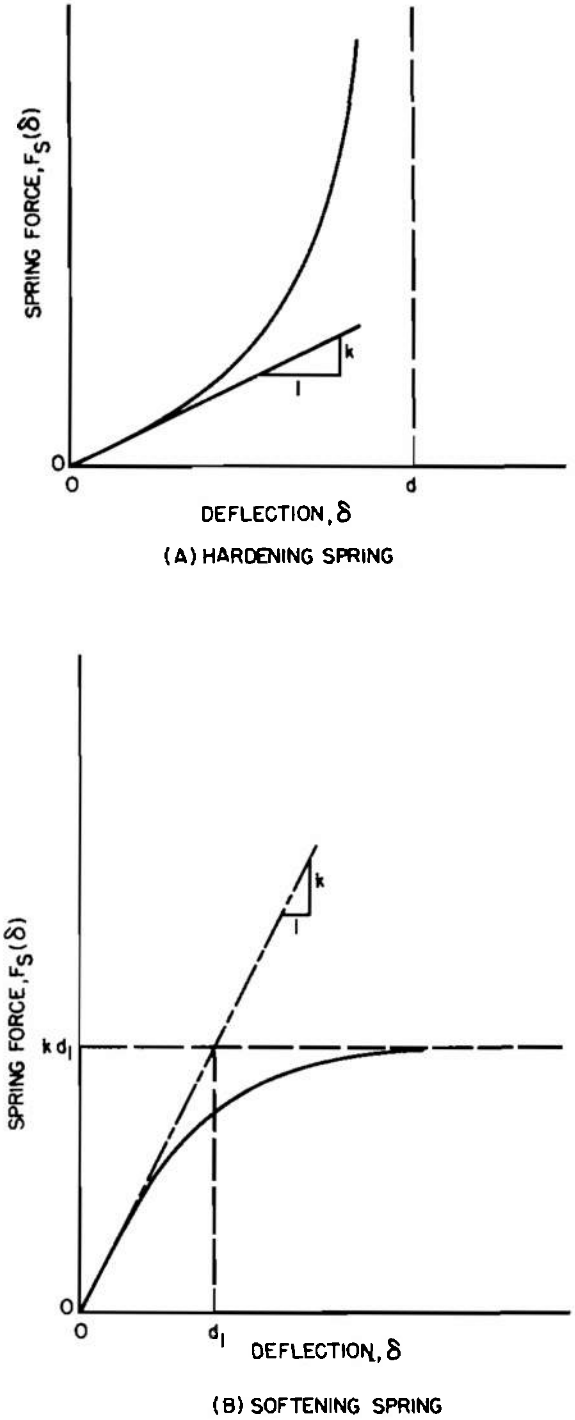

Figure 10-18. Typical Force-Deflection Curves for Nonlinear Springs

$$\ddot{x}_{max} = \frac{2\omega_n{}^2 d}{\pi} \tan\left(\frac{\pi\delta_{max}}{2d}\right) \qquad (10\text{-}38)$$

The interpretation of the natural frequency ω_n in Equation 10-38 indicates that the natural frequency of the hardening spring depends upon the amplitude for large deflections. For small amplitudes, however, the natural frequency has the same meaning as in a linear system.

10-18.2.2 Softening Spring

A nonlinear spring characterized by hyperbolic tangent elasticity is the softening spring. In this case, the slope of the curve representing the force vs deflection decreases with increasing deflection (Figure 10-18(B)). The force-deflection characteristic for a typical softening spring is

$$F_S(\delta) = kd_1 \tan h\left(\frac{\delta}{d_1}\right) \qquad (10\text{-}39)$$

where

k = initial slope of the curve, lb/in.
d_1 is defined by Figure 10-18(B), in.
Other terms are as defined for Eq. 10-35.

The relationship between maximum acceleration and maximum deflection for the softening spring can be expressed as

$$\ddot{x}_{max} = \omega_n{}^2 d_1 \tan h\left(\frac{\delta_{max}}{d_1}\right) \qquad (10\text{-}40)$$

The relationship between ω_n and the deflection is similar to that of the hardening spring.

10-19 SPRING TYPES

10-19.1 TORSION BARS (Refs. 32, 33)

The discussion of torsion bar springs given here is a simplified condensation of an extensive subject. Furthermore, new knowledge and techniques are constantly being developed in this field. Therefore, for a comprehensive treatment of this subject refer to the most current editions of References 32 and 33.

10-19.1.1 Description

A *torsion bar spring* is an elongated elastic member which, in normal usage, is fixed at one end and loaded (stressed) by applying a couple, acting in a plane normal to the axis of the bar, to the opposite end. The configuration of torsion bar springs varies with respect to cross section and number of elements composing the spring. The torsion bar springs used in current military suspension systems are straight bars of circular cross section, splined at each end.

10-19.1.2 Design Considerations

For effective operation, torsion bars springs must be made of high grade materials free from inclusions or internal defects. In subsequent processing, such as machining and heat-treating operations, great care must be exercised to maintain high quality.

In the case of steel bars, the torsion bar must be preset during the manufacturing process to take advantage of elastic properties most effectively. Loading must be carefully considered since a preset bar fails quickly with repeated applications of service torque loads applied in the wrong direction. Since prestressing is performed in the direction of the loading, right- and left-hand bars must be produced if the bars are not symmetrical. While presetting does increase resistance to fatigue failure, additional improvement is obtained by the shot-peening process.

After a steel torsion bar has been precisely machined to a smooth finish, it is heat-treated, shot-peened, preset, indexed, marked, and processed to prevent nicking and corrosion in handling and in service. Such techniques as Parkerizing and spiral wrapping have been used but more satisfactory methods are being developed.

For materials other than steel the basic design considerations include factors such as maximum stresses, endurance limits, stress raisers, and environmental effects.

10-19.1.3 Basic Design Calculations

The following discussion is limited to the basic torsion bar types and one of the linkage systems frequently used in suspensions.

For a *round bar* in torsion—i.e., a straight bar of circular cross section—loaded in torsion only, the following relationships can be expressed

Windup Angle $\varphi = \dfrac{32TL}{\pi d^4 G} = \dfrac{2SL}{dG}$, rad (10-41)

Torsional Rate $T' = \dfrac{T}{\varphi} = \dfrac{\pi d^4 G}{32L}$, in.-lb/rad (10-42)

Stress (shear) $S = \dfrac{16T}{\pi d^3} = \dfrac{\varphi dG}{2L}$, psi (10-43)

Stress Rate $\dfrac{S}{\varphi} = \dfrac{dG}{2L}$, psi/rad (10-44)

Design Load $P = \dfrac{\pi d^3 S_v}{16r}$, lb (10-45)

Resilience* $U = \dfrac{Pf}{2} = \dfrac{S^2_v V}{4G}$, in.-lb (10-46)

Notation:

T = torque applied to the bar, in.-lb
L = active length of bar, in.
d = diameter of round bar, in.
G = shear modulus of elasticity, psi
S = shear stress, psi
S_v = design shearing stress, psi
r = length of the moment arm, in.
f = deflection at the end of the moment arm, in.
V = volume of spring material, cu in.

For a *rectangular bar* in torsion, i.e.—a straight bar of rectangular cross section—loaded in torsion only, the following relationship can be expressed

Windup Angle $\varphi = \dfrac{TL}{\eta_3 a^3 bG} = \dfrac{\eta_2 SL}{\eta_3 aG}$, rad (10-47)

Torsional Rate $T' = \dfrac{T}{\varphi} = \dfrac{\eta_3 a^3 bG}{L}$, in.-lb/rad (10-48)

Stress (shear) $S = \dfrac{T}{\eta_2 a^2 b} = \dfrac{\eta_3 \varphi aG}{\eta_2 L}$, psi (10-49)

Stress Rate $\dfrac{S}{\varphi} = \dfrac{\eta_3 aG}{\eta_2 L}$, psi/rad (10-50)

Design Load $P = \dfrac{2a^2 bS_v}{9r}$, lb (10-51)

Resilience $U = \dfrac{Pf}{2} = \dfrac{4S_v{}^2 V(K^2+1)}{45G}$, in.-lb (10-52)

Notation:

a = short side of the bar, in.
b = long side of the bar, in.
$K = \dfrac{a}{b}$
η_2 = Saint Venant's stress coefficient (see Figure 10-19) } dimensionless
η_3 = Saint Venant's stiffness coefficient (see Figure 10-19) } dimensionless

Other terms are as defined above, for the round bar.

The torsion bar spring and lever, Figure 10-20, is frequently used in suspension systems. In this figure, the deflection f and the angles α and β are measured from a reference line which is perpendicular to the applied load P and passes through the center of the torsion bar (Ref. 32). The deflections f and α are considered positive when above the reference line, and negative when below; β is considered positive when as shown. Symbols used are

f = deflection of the end of the lever from the reference line, in.
α = angle between the lever centerline under load P and the reference line, deg
β = angle between the lever centerline and the reference line under zero load, deg.

The load deflection characteristics of the torsion bar spring and lever are not linear; the following equation defines their relationship:

*Resilience is defined as the work done in deflecting a spring through a distance f.

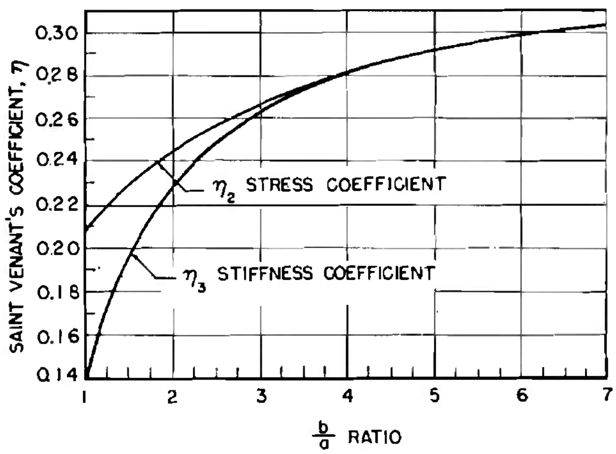

Figure 10-19. Saint Venant's Coefficients for Rectangular Bar in Torsion

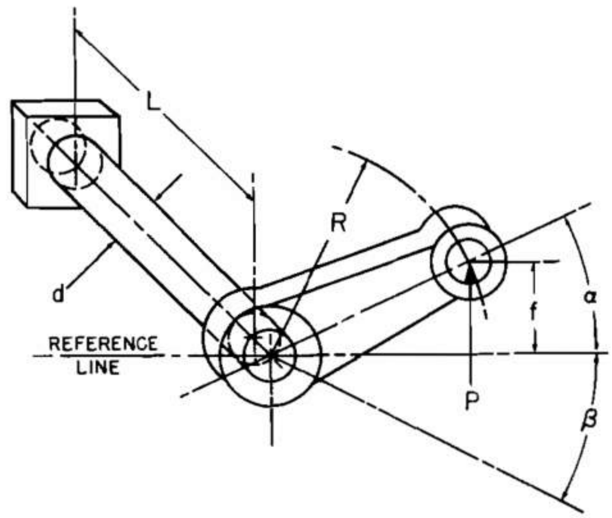

Figure 10-20. Torsion Bar Spring and Lever

$$P = \frac{T}{R \cos \alpha}, \text{ lb} \qquad (10\text{-}53)$$

Since torque equals torsional rate multiplied by windup angle

$$P = \frac{T'}{R} \frac{(\alpha + \beta)}{\cos \alpha}, \text{ lb} \qquad (10\text{-}54)$$

The vertical rate at the end of the lever is given by $K = dP/df$. If we use the value of P above and the relation $f = R \sin \alpha$ (see Figure 10-20), the differentiation leads to

$$K = \frac{T'}{R^2} \frac{1 + (\alpha + \beta) \tan \alpha}{\cos^2 \alpha}, \frac{\text{lb}}{\text{in.}} \qquad (10\text{-}55)$$

The static deflection at any point is defined as $\delta = P/K$. With the above values, this becomes

$$\delta = R \frac{\cos \alpha}{\frac{1}{\alpha + \beta} + \tan \alpha}, \text{ in.} \qquad (10\text{-}56)$$

All angles in the previous equation are expressed in radians.

The design of torsion bar spring *end fastenings* is of primary importance if satisfactory performance is to be obtained. Since the round steel torsion bar is the type most frequently used, a great deal of development has been accomplished on end fastenings for this configuration.

Of all known arrangements, *splined ends* permit the smallest end diameters with round bars. Satisfactory static strength may be obtained even if the diameter of the end is small; but if the endurance life of the end is to be equal to that of the bar, experience indicates that the outer diameter of the end should be at least 1.20 times the diameter of the bar. Where possible, it is advisable to keep this ratio above 1.30. The splines themselves must be designed with the following considerations in mind:

(a) Stress Concentration at the Spline Root. At the point where the anchor first contacts the spline (next to the taper section) the bar still is under the full windup torque. Even with the enlarged diameter, the stress at the bottom of the spline root may be nearly as high as that in the body of the bar. To it is added a stress pattern resulting from the localized pressures on the spline flanks so that the total stress may be very high. This stress concentration can be minimized by a large root radius and by a high pressure angle of the spline. If premature breakage still occurs, a large fillet radius will permit shot peening of this area, thereby obtaining satisfactory life.

(b) Compressive Stress on the Flanks of the Spline. A design that is favorable for condition (a) above reduces the flank area and, therefore, increases the specific pressure on the flanks. If this pressure is too high, an excessive local permanent set will occur somewhere near the contact area and may initiate fatigue fractures.

Experience has shown that a specific pressure of 150,000 psi is permissible with the alloy steels recommended for torsion bar springs (see paragraph 10-36.1.9). This is the pressure calculated for full length contact on all splines.

Male and female splines, when new, will not show full contact but, during the first few load applications, the high spots will experience permanent deformation and the required contact area will be established. On parts which have been fabricated with reasonable accuracy, the permanent set required to produce this contact is not large enough to cause damage.

The *spline length* for satisfactory performance is a function of the spline diameter. Both male and female splines deform elastically during load application; therefore, there is a definite limit to the useful length. Experience has shown that i is useless to make the splines longer than about 0.4 times the spline diameter. A well designed spline of this length will show contacts on all faces varying in length between 50 percent and 100 percent of spline length.

The *transition section*—from bar diameter to end diameter—should be gradual in order to keep stress concentrations to a minimum. A taper of 30 deg included angle and fillet radius of 1.3 to 1.5 times body diameter have been found satisfactory.

The windup angle of a short element of length dx and diameter y under an applied torque T is

$$d\phi = \frac{32\,T\,dx}{\pi\,y^4\,G}, \text{ rad} \qquad (10\text{-}57)$$

From this the windup angle of a tapered section in radians can be calculated by integration as

$$\phi_{taper} = \left(\frac{32\,T\,l}{\pi\,d^4 G}\right)\left(\frac{1}{3}\right)\left[\frac{d}{d_e} + \left(\frac{d}{d_e}\right)^2 + \left(\frac{d}{d_e}\right)^3\right] \qquad (10\text{-}58)$$

where

d = diameter of the small end (equal to the diameter of the bar), in.

d_e = diameter of the large end of the bar, in.

l = length of the taper, in.

This is correct for any taper angle. By comparing this equation with Equation 10-41 for

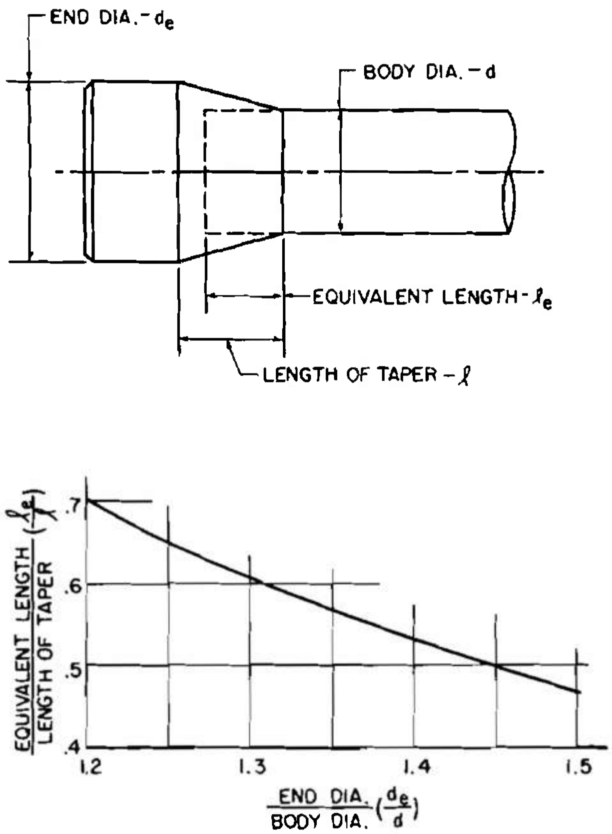

Figure 10-21. Equivalent Length of Taper

windup angle, it is apparent that the tapered portion has the same flexibility as a bar of diameter d and of a length

$$l_e = \frac{l}{3}\left[\frac{d}{d_e} + \left(\frac{d}{d_e}\right)^2 + \left(\frac{d}{d_e}\right)^3\right] \qquad (10\text{-}59)$$

The term l_e is the equivalent length of the taper section (in.). Figure 10-21 gives the value of this equivalent in a convenient form. The total active length of the bar becomes

L = length between tapers + equivalent lengths of both tapered ends

The fillet radius between taper and cylindrical section will cause a change in the equivalent length. However, on all but extremely short bars it is accurate enough to assume that the equivalent length of taper, including stiffening caused by the fillet radius, is equal to one-half the length of the taper.

Stresses in the torsion bar *anchor member* or *hub* will not be as high as they are in the bar, although high local stresses exist. The female splines must also be designed with good fillets in order to avoid high stress concentrations, but shot peening is not necessary.

The high pressure angle required for durable spline design will produce large radial components of the spline pressures and, therefore, a tendency to burst the anchor as if it were loaded by internal hydraulic pressure.

The female splines should be longer than the male splines and so positioned that they will overlap the male splines at both ends, regardless of tolerances and assembly variations.

The hardness of the anchor can be considerably lower than that of the bar. In many installations it has been held below Rockwell C-30. In order to make broaching possible after heat treatment, the hardness should not exceed Rockwell C-36.

10-19.1.4 Materials and Processing of Steel Torsion Bars

Detailed material and processing schedules for steel bars are given in Reference 32. In general, alloy steels such as SAE 9260, 9262, and 8660 and carbon steel SAE 1046 are used for torsion bar springs. For satisfactory results, it is mandatory that the processing schedule developed for military torsion bar springs be followed. This schedule includes factors such as: (a) composition, (b) bar size, (c) quality requirements, (d) deep etch tests, (e) fracture tests, (f) Jominy hardenability tests, (g) end upsetting, (h) normalizing, (i) straightening, (j) rough turning, (k) finish grinding, (l) Magnaflux inspection, (m) spline cutting, (n) heat treatment, (o) quenching, (p) tempering, (q) final hardness testing, (r) final straightening, (s) shot peening, (t) presetting, and (u) protective coatings.

The presetting and shot peening processes are advisable for the severe, unidirectional loading experienced in military service. For torsion bars made of through-hardened steel treated to Rockwell C-50, Reference 32 recommends a maximum applied shear strain γ_a of 0.022 radian during the presetting operation. This will result in a permanent set of about 0.005 to 0.008 radian. The shear strain γ_a may be calculated from

$$\gamma_a = \frac{1}{57.3}\left(\frac{\varphi_a r}{l}\right) = \frac{\varphi_a d}{114.6\, l}\text{, in./in.} \qquad (10\text{-}60)$$

where

φ_a = maximum angle of twist (during presetting), deg

l = active length of the bar, in.

d,r = bar diameter and radius, respectively, in. The constant 57.3 converts units of shear strain to in./in.

For maximum endurance under repeated loading, shot peening is also required. This should be done prior to the presetting process. Presetting is of no value when reversal of torque is present.

10-19.1.5 Conventional Lateral Installations

At the present time, laterally installed torsion bar suspension systems are used on all American production tanks. The basic arrangement of these systems is essentially the same for each type of tank, the major difference being the number and size of road wheels required. The interchangeability of many of the components, the simplicity of construction, the ease of maintenance, and the increased spring protection against ballistic attack are factors that have contributed to the standardization of this suspension system. A disadvantage is the increase in tank height caused by locating the torsion bars between the turret floor and the bottom of the hull.

The road wheels of the torsion bar suspension are fully independent, each mounted on a solid forged suspension arm (Figure 10-4). The arm is attached rigidly to a spindle carried in roller bearings in a cast support housing bolted to the bottom of the hull. Turning of the spindle in its bearings is spring-controlled by a torsion bar engaged internally by splines with the spindle. On some tanks a track-adjusting compensating wheel is attached to the front road wheel suspension arm through an adjustable link. On other tanks an individually sprung tensioning idler roller bears against the track behind the rear road wheel. Track support rollers are used in most suspensions to support the returning tracks above the road wheels.

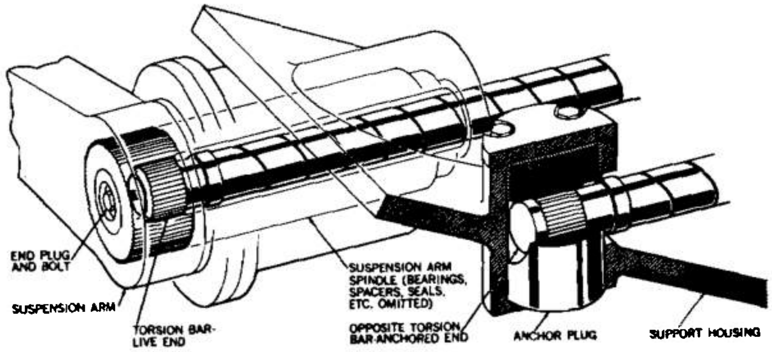

Figure 10-22. Torsion Bar Suspension Arm End Support Details

Direct-acting shock absorbers are attached between the hull and the suspension arms of the extreme front and rear road wheels and a number of intermediate wheels, depending on the need for damping. Road wheels at or near the center of the tank are rarely equipped with shock absorbers because their location reduces the need for shock control. A volute bumper spring is attached to the hull over each suspension arm to limit and cushion extreme arm travel. In some suspension designs, solid bumper stops are used for suspension arms near the center of the vehicle.

Figure 10-22 is a detailed view of a typical suspension arm and torsion bar end supports. Cast armor steel support housings, one for each road wheel, are bolted over openings in the hull bottom. Each support houses a bearing-mounted suspension arm spindle and includes a seat for the opposite torsion bar plug. The road wheel suspension arm is usually press-fitted on the hollow, externally serrated spindle. Forged one-piece arm and spindle units are also in production. The spindle is internally serrated to receive the torsion bar spring. One serration on the spindle and on the torsion bar is omitted to facilitate removal and installation.

10-19.1.6 Longitudinal and Interconnected Torsion Bar Systems

Longitudinal torsion bar springs have been developed for nonmilitary wheeled vehicles. These are used in conjunction with the link systems discussed in Chapter 9.

Interconnected torsion bar systems have been developed for several nonmilitary wheeled vehicles in order to have equal pitch and bounce frequencies of the sprung mass.

10-19.1.7 Torsion Bar With Tube

Combination torsion bar springs consisting of a solid member and a tubular member, with the former positioned within the latter, permit the use of relatively short bars. The combination also permits either series or parallel arrangements of the individual springs.

10-19.1.7.1 Series Arrangement

Figure 10-23(A) shows the schematic of a series combination of torsion bar springs. In a series arrangement, the total angular deflection for a given force or torque equals the sum of the angular deflections of each spring as if each carried the entire load. If a torque T is applied to the disk J of the system in Figure 10-23(A), the total deflection of the disk is $\Theta_T = \Theta_1 + \Theta_2 = T/k_{t1} + T/k_{t2}$; since $\Theta_T = T/k_{te}$, the equivalent torsional spring rate of the bar k_{te} can be expressed as

$$k_{te} = \frac{k_{t1}\,k_{t2}}{k_{t1} + k_{t2}}\text{, in.-lb/rad} \qquad (10\text{-}61)$$

10-19.1.7.2 Parallel Arrangement

Figure 10-23(B) shows the schematic of a parallel combination of torsion bar springs. When a torque T is applied to the disk J of this system, the total angular displacement of the disk has the following relation:
$1/\Theta_T = 1/\Theta_1 + 1/\Theta_2 = k_{t1}/T + k_{t2}/T$; again $\Theta_T = T/k_{te}$ and the equivalent torsional spring rate becomes

$$k_{te} = k_{t1} + k_{t2} \qquad (10\text{-}62)$$

10-19.1.8 Laminated Torsion Bars

Laminated torsion bars, composed of elements having rectangular cross sections, have been devel-

Figure 10-23. Combination Torsion Bar Springs

oped for suspension systems. In one design, the rectangular elements are stacked to form a square bar with retaining sleeves at each end. In the laminated bar—composed of flat rectangular elements—only the center element of a group will be in pure torsion. If a group has an even number of elements, i.e., no center element, none of the elements will be in pure torsion. This causes sliding between the elements if their ends are not constrained with respect to longitudinal motion. A torsion bar spring of this type will possess inherent friction damping. This small amount of damping is desirable to reduce wheel dance and the resulting secondary vibration induced into the sprung mass of the vehicle.

10-19.1.9 Nonmetallic Torsion Bars (Ref. 34)

Currently, a research program is being conducted to determine the feasibility of filament wound Fiberglas torsion bars for military tracked vehicle suspension systems. A conceptional spring under investigation is shown in Figure 10-24. When developed, the glass reinforced plastic torsion bars are expected to produce the following overall vehicle improvements over the presently used equivalent steel torsion bar systems.

(a) Increased mobility through:
  (1) increased wheel travel
  (2) lower spring rate
  (3) inherent damping characteristics
  (4) reduced suspension/vehicle weight ratio

(b) Simpler maintenance as a result of reduced weight.

These improvements are consequences of the following advantages of the filament wound bars over steel bars

(a) Higher energy storage capacity (approximately 5 to 1 on a volume basis)
(b) Lighter weight (approximately ¼ that of steel)
(c) Optimum utilization of the material (proper orientation of the fibers permits utilization of the material in pure tension).

These factors are illustrated by Figure 10-24.

10-19.2 PNEUMATIC SPRINGS

10-19.2.1 General Description

The usual pneumatic or air spring for a vehicle suspension system consists, essentially, of a fabric reinforced elastomeric bellows of one or two convolutions, sealed at each end by suitable mounting plates, pedestal or clearance chambers. Although the elastomer deflects elastically when the spring is loaded, the principal spring is the air within the unit. These springs may be installed on vehicles so they are direct-acting or may be loaded by means of a linkage.

One of the characteristics of this type of spring is a variable spring rate (the analytic aspects of this characteristic are discussed in a following paragraph). With a small clearance volume, the rate of change of the spring constant is quite pronounced. For systems using a bellows arrangement, a factor affecting the spring rate at low loading is the stiffness of the bellows wall. In some systems the bellows is only slightly bowed at low loading and contributes substantially to the spring stiffness. The more refined air springs employ a system of values to maintain a midpoint vehicle level under variations in the static loading by controlling the air pressure within the bellows. Under these conditions, the bellows is usually bowed and resistance to deflection is substantially reduced. The action of the air spring is essentially friction-free and provides inherent isolation from high frequency vibrations.

The bellows type of air springs generally operate at pressures of 65 to 75 psi under static loading, but have been used successfully under static load pressures of 2 to 100 psi. The usual installation is such that the bellows can be inflated to increase ground clearance, thereby improving passability in difficult terrain.

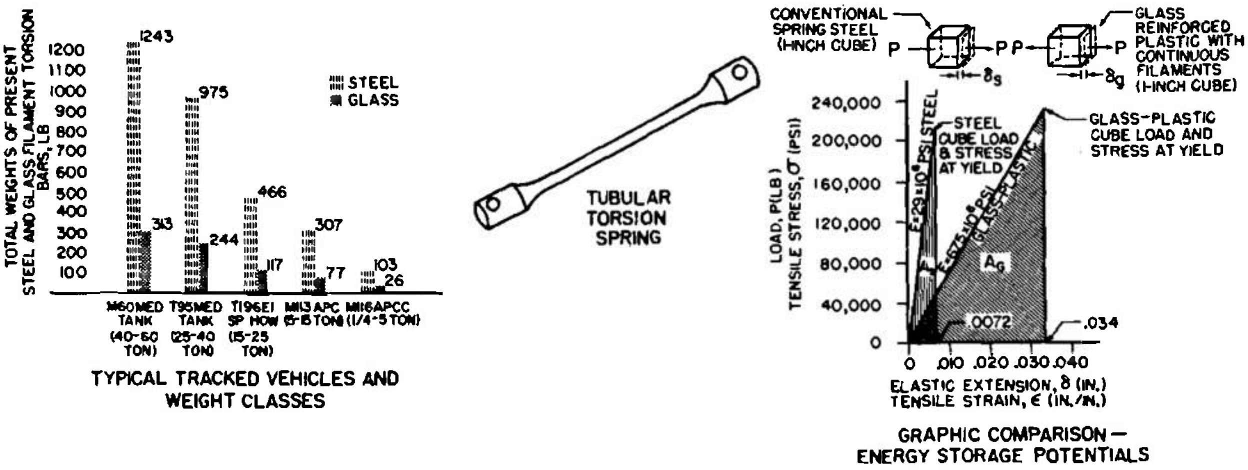

Figure 10-24. Filament Wound Fiberglas Torsion Bars

The developmental effort on air springs for military vehicles is relatively slight compared to the effort being applied to the hydropneumatic spring systems. The advantages of the latter system are discussed in paragraph 10-19.4.

10-19.2.2 Analytical Considerations (Ref. 35)

The behavior of the pneumatic spring can be analyzed by assuming the bellows of Figure 10-25 to be a piston and cylinder of area A. Denote the pressure and volume of the gas in the equilibrium and displaced positions as P_o, V_o, and P, V, respectively, and assume an adiabatic change, then the following gas law holds

$$PV^{\gamma} = P_oV_o^{\gamma} \qquad (10\text{-}63)$$

where

γ = ratio of specific heats for the gas

Differentiating with respect to the displacement x gives

$$\frac{dP}{dx} = -\gamma P_oV_o^{\gamma}V^{-(\gamma+1)}\frac{dV}{dx} \qquad (10\text{-}64)$$

Since the volume corresponding to any displacement x is $(V_o - Ax)$

$$\frac{dV}{dx} = -A \qquad (10\text{-}65)$$

Furthermore, since the spring rate k is equal to the force per unit displacement

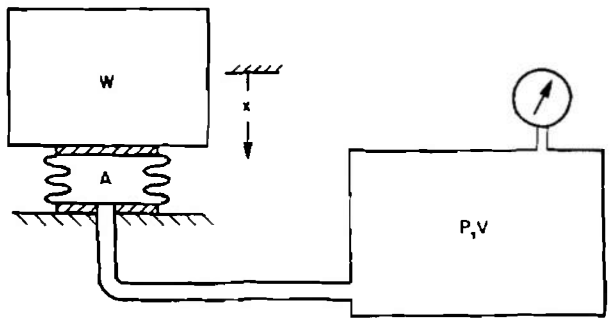

Figure 10-25. Schematic of a Pneumatic Spring

$$k = A\frac{dP}{dx} = \frac{\gamma P_oA^2}{V_o}\left(1 - \frac{Ax}{V_o}\right)^{-(\gamma+1)}, \text{ lb/in.} \qquad (10\text{-}66)$$

This equation indicates that the spring rate of the pneumatic spring is dependent on the displacement and hence is nonlinear. But, if the magnitude of the displacement x is assumed to be small (as is the case for present systems), Equation 10-66 can be simplified to give an approximate spring rate

$$k \cong \frac{\gamma P_oA^2}{V_o} = \frac{\gamma WA}{V_o}, \text{ lb/in.} \qquad (10\text{-}67)$$

where

W = weight of the sprung mass, lb

A = area of the piston, in.^{2}

V_o = initial volume of the pneumatic spring system, cu in.

Equation 10-67 indicates that a very soft spring is possible by providing a very large initial volume V_o for the system. The initial pressure P_o is provided by the weight W acting on the piston area A. If damping is desired, throttling valves may be placed in the pneumatic lines. The natural frequency ω_n of the system (rad/sec) is expressed as

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{\gamma A g}{V_o}} \qquad (10\text{-}68)$$

where g is the acceleration of gravity.

10-19.3 HYDRAULIC SPRINGS (Ref. 36)

10-19.3.1 General Description

The hydraulic or liquid spring consists of a sealed plunger or piston working in a highly finished cylinder against an enclosed volume of liquid. Although the deflections of the metal parts of the assembly contribute to the spring rate, the principal source of elasticity is derived from the compressibility of the fluid. The maximum compressibility (percent reduction of volume) of liquids currently available is about 12 percent for the limiting pressure of 20,000 psi.

The hydraulic spring is characterized by a very high spring rate and compactness. Although it has about the same energy-storing capacity for its weight as the helical coil spring, it occupies less functional space.

The disadvantages of the hydraulic spring are associated with sealing the fluid under the high pressures developed in the device. The plunger and cylinder must be finely finished and held to close tolerances to function properly and provide satisfactory sealing. The seals are also subject to wear and temperature failure.

The physical characteristics of the liquid spring are such that they are suited to bump stop applications. This application is discussed in Section IX of this chapter.

10-19.3.2 Analytical Considerations

All fluids may be compressed, to some degree, by the application of pressure; elastic energy being stored in the process. Thus fluids are elastic media and, as such, possess a modulus of elasticity. Since fluids are not rigid, the modulus of elasticity must be defined on the basis of volume—a "bulk modulus." The bulk modulus E is expressed by the equation

$$E = -\frac{V\,dP}{dV}, \text{ psi} \qquad (10\text{-}69)$$

where

V = initial volume, cu in.
dP = change in pressure, psi
dV = corresponding change in volume, cu in.

The negative sign denotes a decrease in volume with an increase in pressure.

The spring rate for a liquid spring is

$$K = -\frac{A^2 E}{V}, \text{ lb/in.} \qquad (10\text{-}70)$$

where

A = area of the piston, sq in.

The bulk modulus of a given liquid varies slightly with pressure; hence the spring rate is nonlinear. In addition, the bulk moduli of liquids differ considerably, for example, the average bulk modulus of water is 300,000 psi, while the value for kerosene (JP-1) is 200,000 psi. Bulk modulus also changes with temperature, for example, water exhibits a bulk modulus of 293,000 psi at 32°F, 334,000 psi at 130°F, and 300,000 psi at 212°F. As a result, the load-deflection performance is affected by relatively moderate changes in temperature.

Normally, the hydraulic spring is under pressure even when the spring is extended to its maximum full length and, as such, is preloaded. This condition can be altered to suit service conditions but some preloading is desirable from the standpoint of sealing.

The energy losses within the hydraulic spring are predominantly friction losses which are represented by the hysteresis loop of a load-unload cycle. Good spring design dictates the use of low friction seal elements and bore finishes of 8 μ in. RMS or finer.

Actual tests on several commercial hydraulic springs show that the static loading efficiencies depend on the spring stroke (as a percentage of the maximum stroke). For example, a spring rated at 3800 pounds maximum load at a stroke of 1½ inches developed an efficiency of 80 percent at an 80 percent spring stroke, while at 20 percent spring stroke the efficiency dropped to 50 percent.

Static tests do not necessarily reflect the dynamic efficiency of the hydraulic spring. For the spring described above, the average efficiency under dynamic loading was found to be 89 percent (at approximately $1\frac{1}{4}$ inch stroke).

Tests on commercial liquid springs indicate that definite limits exist for steady-state loading frequencies. The limit on steady-state frequency is mainly a result of the hysteresis of the unit. This limit is a function of the individual spring type. Spring failures at high frequencies are normally caused by overheating. When the heat so generated becomes significantly greater than the steady-state heat output, the rise in the fluid temperature causes excessive pressure, excessive wear, and eventual leakage and failure.

10-19.4 HYDROPNEUMATIC SPRINGS

10-19.4.1 Basic Discussion (Ref. 37)

The hydropneumatic spring, as used on vehicular suspension systems, consists of a closed volume of gas separated from a chamber containing hydraulic fluid by means of a flexible diaphragm or floating piston. The resiliency of the system is derived from the compressibility of the gas. The hydraulic portion of the system is employed to maintain or adjust ground clearance, effect body or hull leveling, effect suspension lockout, and provide the system damping. In the usual hydropneumatic spring, the gas system is charged to a predetermined pressure and is not used as a source of adjustment.

Since the rate of the hydropneumatic spring reflects the adiabatic compression of the gas, it is nonlinear. It is a power curve expressed by Equation 10-66.

10-19.4.2 The Hydropneumatic Suspension System (Refs. 38, 39)

10-19.4.2.1 General Discussion

The current interest in the development of a suitable hydropneumatic suspension system for tanks has resulted in the design and development of several experimental suspensions of this type. These systems have been installed on a modified T95 Medium Tank, an experimental T85 Prime Mover, and an Air Spring Test Vehicle.

Some of the advantages or improvements that the hydropneumatic suspension system under consideration offers over the conventional mechanical spring system can be listed

(a) Increased wheel travel is available. For the system applied to the T95 Tank, almost 19 inches of total displacement is available—with up to 14 inches from neutral to bumpout. This value can be compared to 8 inches of travel for conventional torsion bar suspensions and about 3 inches for the volute spring suspensions.

(b) The nonlinear spring rate is better suited to the varying suspension loads. The spring rate for the hydropneumatic system is quite low near the normal or neutral position but increases rapidly as bumpout is approached.

(c) Height and attitude control are available. For normal operation, automatic leveling valves maintain the normal operating height of the vehicle. However, to better suit particular conditions, the driver may select another height. For example, to achieve a minimum vehicle silhouette, the ground clearance of the tank can be reduced to 6 inches from the normal height of 17 inches. If maximum ground clearance is desired, the vehicle can be raised until the clearance is 25 inches. The suspension system also permits the vehicle to be leveled on slopes of up to 20 percent.

(d) With the hydropneumatic suspension, a hydraulic lockout of the elastic support system is readily accomplished (see Section X).

(e) The relatively compact hydropneumatic suspension units do not extend under the basket area of the tank, hence a lower tank silhouette is possible. Externally, the shock absorbers and their mounting brackets are eliminated.

In addition to the improvements listed above, the hydropneumatic suspension system permits flexibility in vehicle application. Some of the characteristics of the system are

(a) The normal or preselected height of the vehicle is not affected by changes of the sprung weight since the leveling system senses position and not load or pressure.
(b) The spring rate can be changed by varying the volume and/or pressure of the pneumatic spring.
(c) The damping can be varied by changing the orifice size.
(d) The normal or static vehicle height can be changed by changing cams.

10-19.4.2.2 Design Details

The following discussion is a brief outline of the design, details, and objectives of the hydropneumatic suspension units used on the modified T95 Medium Tank.

Among the major physical design objectives for the suspension unit were (a) to obtain a cartridge type assembly which could be readily installed or removed; (b) to incorporate leveling valves which would sense arm position; and (c) to combine the five functional components (actuator, manifold, valve, damper, and accumulator) into one unit so that high dynamic pressures are not transmitted through lines and fittings.

The basic components of the hydropneumatic suspension assembly are: (a) the rotary hydraulic actuator, (b) the valve section, and (c) the accumulator. The rotary actuator is mounted in the hull of the vehicle and is operated directly by the road wheel arm. The rotary actuators developed for the program have the following design characteristics

(a) a torque capacity of 56,000 in.-lb per 1,000 psi,
(b) a total rotation of 77 deg (71 deg used), and
(c) a double-acting capability (single-acting used).

In addition, the units contained three vanes, needle bearings, a metal-to-metal face-type external seal, and external stops to prevent internal contact between vanes and reaction shoes. The aluminum manifold is bolted directly to the rotary actuator. Its function is to direct the output of the actuator and to house the two leveling valves. The accumulators are similar to commercial piston types except for the high-strength requirements (20,000 psi minimum burst strength). The accumulators are pre-charged with dry nitrogen gas.

10-19.5 TORSION-ELASTIC SPRINGS (Refs. 40, 41)

10-19.5.1 Description and Characteristics

The use of rubber and other elastomers loaded in tension or compression as the principal elastic medium has not proven successful in applications for vehicular suspension systems. However, rubber loaded in shear exhibits favorable characteristics and several military as well as commercial vehicles have successfully employed suspensions that utilize this approach. The rubber torsion spring, in general, consists of a metal shaft and a concentric metal shell with the elastomer between them and bonded to each. Either the shaft or the shell is rigidly attached to the sprung mass and the other metallic component is connected to the unsprung mass by an arm or linkage such that relative angular motion between the shaft and shell results when the unsprung mass is deflected. The spring action results from the twisting motion which loads the rubber in annular shear.

Rubber torsion springs operate relatively friction-free and suspensions employing them require shock absorbers to damp undesirable oscillations. Rubber compounds with high hysteresis properties are available but are not suited for use as springs because of their poor elastic properties.

By reference to Table 10-2, it is apparent that rubber torsion springs possess a favorable energy-storing capacity for their weight. However, since rubber is so much less dense than steel, a rubber torsion unit occupies more volume than a comparable steel torsion spring. An undesirable characteristic of rubber is that it is temperature-sensitive; at low temperatures it tends to become stiff and hard especially when idle. Future improvements in elastomers and changes in vehicle requirements may permit greater use of this type of spring.

In addition to the favorable energy-storage

TABLE 10-2
ENERGY-STORING CAPACITY OF SPRINGS

Type of Spring	Energy-Storing Capacity	
	in.-lb per lb of Spring	in.-lb per cu in. of Spring Material
Leaf, equal length steel leaves	100–150	30–40
Leaf, optimum stepped leaves	300–450	85–125
Volute, steel	500–1000	140–280
Hydraulic	600–900	—
Helical, round steel wire	700–1000	200–280
Torsion Bar, steel	1000–1500	280–420
Torsion Bar, rubber	2000–4000	80–160

capacity, rubber torsion springs possess characteristics of basic design simplicity, freedom from noise, resistance to corrosion, and freedom from maintenance. Suspensions employing rubber torsion springs and incorporating an automatic leveling system have been constructed.

10-36.5.2 Design Consideration

A cylindrical spring loaded in torsional shear is shown in Figure 10-26. The stress S_s at any radius r, in., for a spring of width h, in., subjected to a moment M, in.-lb, can be expressed as

$$S_s = \frac{M}{2\pi r^2 h}, \text{ psi} \tag{10-71}$$

The shear stress has a maximum value for the minimum value of r.
Hence

$$(S_s)_{max} = \frac{M}{2\pi r_i^2 h}, \text{ psi} \tag{10-72}$$

The total angular rotation of one metallic element with respect to the other, Θ, (in radians) can be expressed as

$$\Theta = \frac{M}{4\pi h G}\left(\frac{1}{r^2_i} - \frac{1}{r^2_o}\right) \tag{10-73}$$

where

M = applied moment, in.-lb
h = length of the material in shear, in.
G = modulus of elasticity in shear of the elastic material, psi
r_i, r_o = inner and outer radii of the elastomeric cylinder, respectively, in.

Working stresses for the rubber torsion spring are generally limited to 25 to 50 psi in shear.

10-19.6 HELICALLY WOUND AND LEAF SPRINGS (Refs. 43, 44)

A comprehensive treatment of helically wound and leaf spring design is presented in the references denoted above. These references should be consulted for detailed information on all phases of spring design. Additional information is available in Refs. 42, 45-48. The information presented in the following paragraphs will serve as an outline of the basic characteristics of helically wound and leaf springs.

NOTE:

In applying the spring equations (Eqs. 10-74 to 10-85), one should note that the formulas are sufficiently accurate for most design purposes when the spring index c (see par. 10-19.6.1) is large and the helix angle is small (less than about 15 deg). Some guidance to the magnitude of the spring index is available. Some spring designers recommend c = 9 as the ideal value; other general rules give values between 6 and 9, with 3 or 4 as a realistic lower limit. Only in very unusual cases, where other design considerations might prevent a higher value of c, should the lower limit be used.

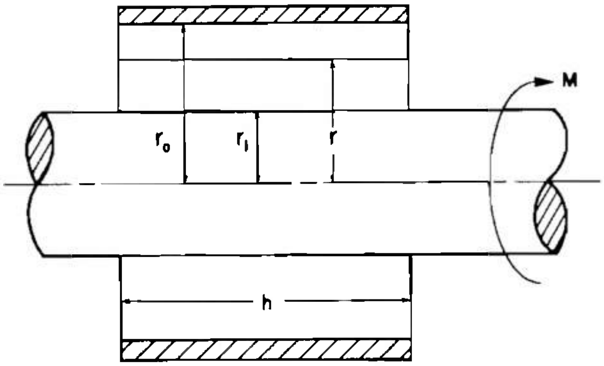

Figure 10-26. Cylindrical Rubber Torsion Spring of Constant Thickness

10-19.6.1 Helical Springs (Compression and Tension) (Ref. 42)

Helical springs consist of round, square, or rectangular wire or bars wound in the form of a helix. They are loaded along the axis of the helix and may be designed to function as compression springs or extension springs. Compression springs are incorporated in vehicular suspension designs much more frequently than extension types. Currently, helical coil springs are used extensively on the lighter weight military wheeled vehicles in conjunction with independent wheel suspensions. Their use on tracked vehicles has been limited, owing to the space required by the spring compared to the space required by torsion bars.

For a cylindrical helical spring of circular cross section wire, the following equations apply

$$P = \frac{1}{k}\left(\frac{\pi d^3 S_v}{16r}\right) \tag{10-74}$$

$$f = \frac{64nr^3P}{d^4G} = \frac{1}{k}\left(\frac{4\pi nr^2S_v}{dG}\right) \tag{10-75}$$

$$U = \frac{Pf}{2} = \frac{1}{k^2}\left(\frac{S_v^2V_o}{4G}\right) \tag{10-76}$$

where

$P =$ design load, lb
$f =$ deflection for a given load P, in.
$U =$ work performed in deflecting a spring from 0 to f, (resilience) in.-lb
$d =$ diameter of the wire, in.
$S_v =$ design shear stress, psi
$r =$ mean radius of the coil, in.
$n =$ number of active coils
$G =$ shear modulus of elasticity, psi
$k = \dfrac{4c-1}{4c-4} + \dfrac{0.615}{c}$, stress correction factor
$c = \dfrac{2r}{d}$, spring index
$V_o = \dfrac{nr(\pi d)^2}{2}$, volume of the spring material, cu in.

For a cylindrical helical spring of rectangular cross section wire the following equations apply

$$P = \frac{1}{k}\left(\frac{2b^2hS_v}{9r}\right), \text{ lb} \tag{10-77}$$

$$f = \frac{7.2\pi nr^3P(K^2+1)}{b^3hG} = \frac{1}{k}\left[\frac{1.6\pi nr^2S_v(K^2+1)}{bG}\right], \text{ in.} \tag{10-78}$$

$$U = \frac{Pf}{2} = \frac{1}{k^2}\left[\frac{4S_v^2V_{\square}(K^2+1)}{45G}\right], \text{ in.-lb} \tag{10-79}$$

where

$b =$ width of the wire (measurement radial to spring axis), in.
$h =$ height of the wire (measurement parallel to spring axis), in.
$K =$ ratio b/h
$V_{\square} = 2\pi nrbh$, volume of the spring material, cu in.

Other symbols are as defined for helical springs of circular cross section, except* $c = 2r/b$, spring index

10-19.6.2 Helical Torsion Springs (Ref. 42)

Helical torsion springs are loaded by applying the load in a plane normal to the axis of the helix. The load tends to wind or unwind the spring around an axial support which passes through

*Equations for the stress correction factor exist for rectangular cross sections, but results are in close agreement with the factor k for circular cross section which may be used for most calculations. For more detailed analyses see Ref. 43 for stress (and deflection) correction factors.

the spring. Usually, the spring is installed so that the load tends to wind the spring around the support.

For a cylindrical helical torsion spring of circular cross section wire the following equations apply:

$$P = \frac{1}{k}\left(\frac{\pi d^3 S}{32 r_t}\right), \text{ lb} \qquad (10\text{-}80)$$

$$f' = \frac{Plr_t^2}{EI} = \frac{64 Plr_t^2}{\pi E d^4} = \frac{1}{k}\left(\frac{2 r_t l S}{dE}\right), \text{ in.} \qquad (10\text{-}81)$$

$$U = \frac{Pf'}{2} = \frac{1}{k^2}\left(\frac{S^2 V_o}{8E}\right), \text{ in.-lb} \qquad (10\text{-}82)$$

where

S = design bending stress, psi
r_t = moment arm from the spring axis to the point of application of P, in.
l = length of the spring wire when uncoiled, in.
E = modulus of elasticity of the material, psi
I = area moment of inertia of the circular cross section, in.^{4}
f' = circumferential deflection at a distance r_t from the spring axis, in.

Other symbols are as defined for Equations 10-74 to 10-76, except $k = \frac{4c-1}{4c-4}$, stress correction factor.

For a cylindrical helical torsion spring of rectangular cross section wire, the following equations apply

$$P = \frac{1}{k}\left(\frac{b^2 h S}{6 r_t}\right), \text{ lb} \qquad (10\text{-}83)$$

$$f' = \frac{Plr_t^2}{EI} = \frac{12\, Plr_t^2}{hb^3 E} = \frac{1}{k}\left(\frac{2 r_t l S}{bE}\right), \text{ in.} \qquad (10\text{-}84)$$

$$U = \frac{Pf'}{2} = \frac{1}{k^2}\left(\frac{S^2 V_\square}{6E}\right), \text{ in.-lb} \qquad (10\text{-}85)$$

where

$$k = \frac{3c-1}{3c-3}, \text{ stress correction factor}$$

Symbols S, r_t, l, E, I, and f' are as defined for Equations 10-80 to 10-82.

Symbols b, h, $V_\square$, and $\dot{c}$ are as given for Equations 10-77 to 10-79; P and U as for Equations 10-74 through 10-76.

10-19.6.3 Volute Springs

10-19.6.3.1 Description and Characteristics (Ref. 45)

The usual volute compression spring consists of a relatively wide, thin strip of metal wound on the flat so that each turn or coil fits inside the preceding one with a partial overlap. Loading the spring causes the coils to telescope into each other. The spring, when fully compressed, has a solid height equal to the stock width.

A volute spring differs from the cylindrical helical compression spring in several respects. The coil radius of the volute spring gradually decreases from the larger outer coil to the small inner coil. Since deflection rate or stiffness varies inversely as the cube of the coil diameter, the coil at the large end of a volute spring is much softer than the coil at the small end. Therefore, when a load is applied, the greatest deflection occurs in the large coils, accordingly, these coils are stressed more than the coils at the smaller end. The stress at any point in a coil is limited by "bottoming" of that point. Up to the load where bottoming begins, the volute spring has a constant deflection rate which is a composite of the individual coil rates. The total deflection for loads within this range is thus distributed over the actual length in inverse relation to the rate at any point. When the first coil begins to bottom, the spring rate begins to increase rapidly as shown in Figure 10-27. As the coils bottom progressively with an increase in load, the rate also increases until, finally, the spring is compressed solid.

When loaded, the rectangular section of volute springs and the round section of coil springs are both subjected to torsional stresses and direct shear stresses. The torsional stresses are much more critical than the shear stresses. A rectangular section is not well adapted to torsional stresses compared to a round section. With the latter, the stress is uniform around the circumference, whereas the stress is a maximum at or near the middle of the long sides of a rectangular section and a minimum at the corners (edges of the bar). As a result, a

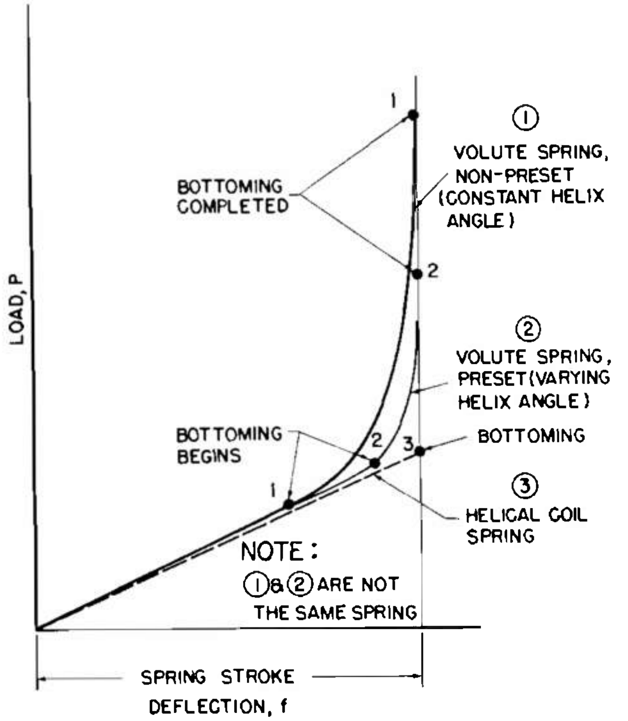

Figure 10-27. Typical Load Deflection Curves of Volute and Helical Coil Springs

rectangular bar is less efficient than a round bar when compared on a weight basis. However, the rectangular bar can be crowded into a more compact space; therefore, volute springs can usually store more energy in limited spaces.

A volute spring is usually designed to produce a desired load-deflection curve when preset. The relationship between the constant rate and variable rate portions of the curve is determined by the ratio of the smallest to largest free coil radius, and by variations of the helix angle along the blade. Through proper correlation of these factors, the distribution of stresses along the blade and the energy-storing capacity can be controlled.

In volute springs used in tank suspensions, the helix angle is varied by presetting so that bottoming deflection occurs at nearly the same load for all coils. Such a spring has a constant rate over most of its deflection range and a variable rate for cushioning purposes at extreme deflections. Load-deflection curves of preset and nonpreset volute springs designed for the same application are shown in Figure 10-27. These curves should be compared on a specific energy-storing basis since the nonpreset spring has a much higher bottoming load and, therefore, would be much heavier if the elastic limit were not exceeded in service.

10-19.6.3.2 Design

Detailed analyses and design data are presented in Refs. 43 and 47. The following discussion introduces the basic characteristics of volute springs.

For a volute spring of rectangular cross section wire and of constant helix angle loaded in compression the equations below apply (Ref. 43). A general limitation on the equations is that the wire thickness is about 1/3, or less, of the width and the spring is wound with the wire width parallel to the spring axis—this is realistic for practical volute springs.

Certain loads as related to deflections of the spring need to be defined.

P = any compressive load on spring, lb
P_1 = load required to bottom first coil, lb
P_2 = load required to compress spring solid, lb

Subscripts 1 and 2 have the same significance when applied to shear stress s and deflection f.

$$P_1 = \frac{Ghb^3\,\alpha\left(1 - 0.63\,\dfrac{h}{b}\right)}{3\,r_1^2} \qquad (10\text{-}86)$$

$$P_2 = P_1\left(\frac{r_1}{r_2}\right)^2 \qquad (10\text{-}87)$$

where

r_1 = greatest mean radius of the active coils, in.
r_2 = least mean radius of the active coils, in.

$$\alpha = \frac{f_2}{2\pi n r_1\left(1 - \dfrac{\beta}{2}\right)} = \text{helix angle, rad}$$

f_2 = free height-solid height = solid deflection, in.

$$\beta = \frac{r_1 - r_2}{r_1}$$

G, h, b, and n are as defined for Equations 10-74 to 10-79.

For loads less than or equal to P_1, the corresponding shear stresses and deflections are

$$S_{(0\text{-}1)} = \frac{3P(c_1+1)}{2hb\left(1-0.63\frac{b}{h}\right)},\ \text{psi} \qquad (10\text{-}88)$$

$$f_{(0-1)} = \frac{6\pi n r_1^3 P K_1}{Ghb^3\left(1-0.63\frac{b}{h}\right)},\ \text{in.} \qquad (10\text{-}89)$$

where

$$c_1 = \frac{2r_1}{b},\ \text{spring index based on } r_1$$

$$K_1 = 1 - \frac{3}{2}\beta + \beta^2 - \frac{\beta^3}{4}$$

When $P = P_1$, Equations 10-88 and 10-89 give the shear stress S_1 and deflection f_1 at the bottoming of the first coil.

The stress and deflection at any load P greater than P_1 can be written as

$$S_{(1\text{-}2)} = \frac{3P\left(c_1\sqrt{\frac{P_1}{P}}+1\right)}{2bh\left(1-0.63\frac{b}{h}\right)},\ \text{psi} \qquad (10\text{-}90)$$

$$f_{(0\text{-}2)} = f_1 + f_{1\text{-}2} = 2\pi n r_1 \alpha\left(\frac{P}{P_1}K_1 - \frac{K_2}{\beta}\right),\ \text{in.} \qquad (10\text{-}91)$$

where

$$K_2 = \frac{1}{2}\left(\frac{P_1}{2P} + \frac{P}{2P_1} - 1\right)$$

For the condition when the spring is compressed solid by a load P_2, the stress S_2 and solid deflection f_2 will be

$$S_2 = \frac{2G\alpha(c_2+1)}{c_2^2},\ \text{psi} \qquad (10\text{-}92)$$

where

$$c_2 = \frac{2r_2}{b},\ \text{spring index based on } r_2$$

$$f_2 = 2\pi n r_1 \alpha\left(1 - \frac{\beta}{2}\right),\ \text{free height-solid height, in.} \qquad (10\text{-}93)$$

10-19.6.4 Leaf Springs

10-19.6.4.1 Description and Characteristics

Leaf springs are flat bar springs that are relatively thin relative to their length and width, and are loaded as beams. Leaf springs for vehicles generally consist of laminations of several leaves of unequal length. Various techniques are used in fastening the laminations and in forming the ends; these techniques have definite effects on the spring characteristics of the assembled spring. Most frequently the leaves are curved.

Although leaf springs are somewhat less efficient than helical springs in terms of energy storage per pound of material (Table 10-2), they are widely used in automotive applications since they may function as structural members as well as the elastic media of the suspension.

A large number of design variations are possible with leaf springs, therefore, a comprehensive discussion would be impractical here. For a comprehensive treatment of the subject see Reference 48.

10-19.6.4.2 Design Considerations

As a first approximation, the design of a leaf spring can be based on the assumption that the spring behaves as a beam of uniform strength (Ref. 43). On this assumption, the equations for the rate k and the maximum stress σ for a symmetric semielliptic leaf spring (Figure 10-28(A)) can be expressed as

$$k = \frac{P}{\delta} = \frac{8Enbh^3}{3l^3},\ \text{lb/in.} \qquad (10\text{-}94)$$

$$\sigma = \frac{3Pl}{2nbh^2},\ \text{psi} \qquad (10\text{-}95)$$

where

P = load, lb
δ = maximum vertical deflection, in.
E = modulus of elasticity of the material, psi
n = number of leaves
b = width of the leaves, in.
h = thickness of each leaf, in.
l = length of the longest leaf of the spring, in

For an unsymmetric semielliptic spring (Figure 10-28(B)), the equations for k and σ are

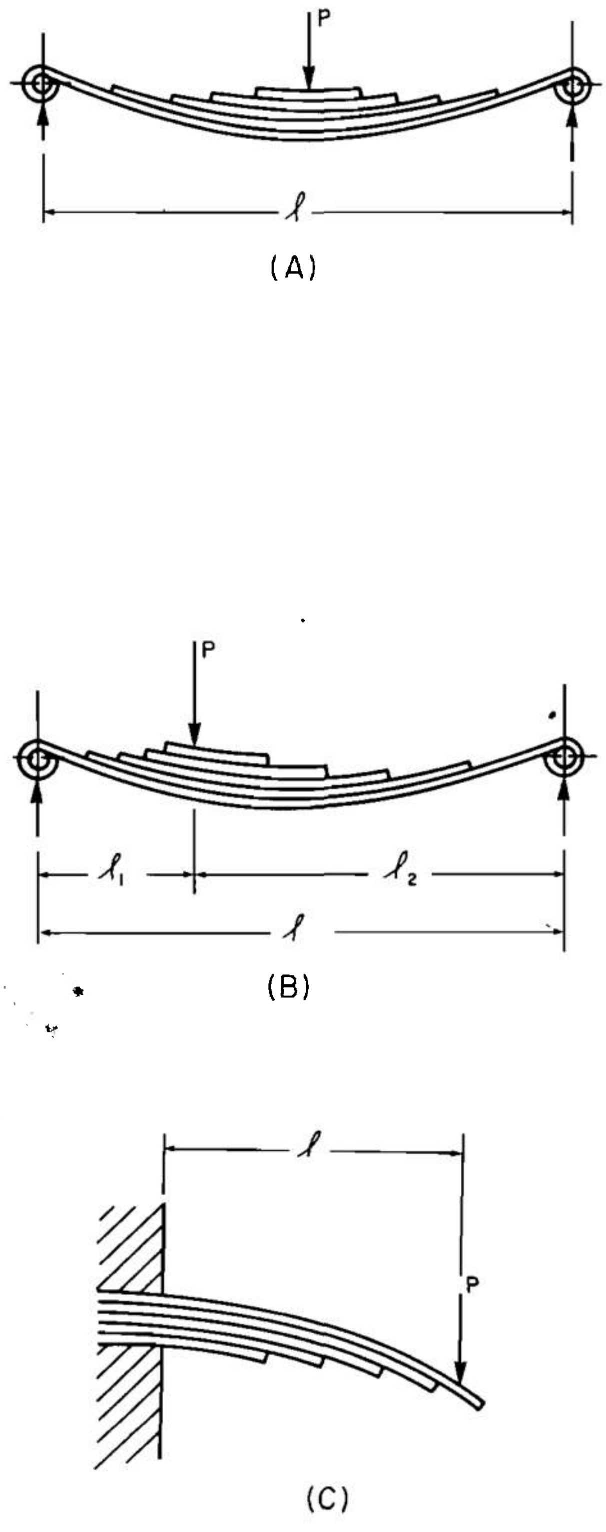

Figure 10-28. Leaf Spring Types

$$k = \frac{P}{\delta} = \frac{Ebnh^3 l}{6 l_1^2 l_2^2}\ , \text{ lb/in.} \qquad (10\text{-}96)$$

$$\delta = \frac{6Pl_1 l_2}{nbh^2 l}\ , \text{ psi} \qquad (10\text{-}97)$$

where

l_1, l_2 = lengths of spring on each side of the applied load, in.

For a cantilever leaf spring (Figure 10-28(C)), the equations for k and σ are

$$k = \frac{P}{\delta} = \frac{Ebnh^3}{6l^3}, \text{ lb/in.} \qquad (10\text{-}98)$$

$$\sigma = \frac{6Pl}{nbh^2}\ , \text{ psi} \qquad (10\text{-}99)$$

In practice, among the effects which may cause deviations from the ideal conditions assumed are (a) the use of leaves of different thicknesses, (b) the use of more than one main leaf, (c) interleaf friction, and (d) the use of spring shackles which cause angular loading at the ends. These effects may result in significant deviations from the values calculated from the equations. For more exact methods of calculation, Reference 48 should be used.

10-19.7 SPRING MATERIALS AND STRESSES

A wide variety of materials are used for springs; however, the great majority of suspension springs are made of alloy steels ranging in analysis from about 0.50% to 0.70% carbon. Materials for springs are fabricated in any one of a number of conditions—hot-rolled, cold rolled or drawn, annealed, and hard-drawn. The choice of material for the various types of springs used in systems is governed by the application in each individual case, as well as cost and manufacturing technique. In some cases, e.g., torsion bar springs for suspension systems, a recommended processing procedure and material specifications have been developed.

Carbon steels, stainless steels, and a number of nonferrous materials are utilized for spring making when the use of these materials is sufficiently advantageous or is necessary. The best selection of material for a given spring, generally, is based upon close cooperation between the manufacturer and the user. The material specifications and the processing techniques for torsion bar suspension springs are given in paragraph 10-19.1.4.

The permissible operating stresses for springs used in vehicular suspensions vary with the type of spring. Extensive data are available (Refs. 44 and 46-48) on the permissible (maximum) operating stresses for various types of spring. The discussion which follows, applies to torsion bar springs used in vehicular suspension systems (Ref. 32).

The permissible operating stress is limited by two factors (a) settling in service, and (b) fatigue failure. When springs operate on a cycle of reversed loading, their life is determined by the stress range, and maximum stresses must be chosen on that basis. The information which follows does not apply, therefore, to springs subjected to reversing loads.

Suspension springs are loaded in one direction. For satisfactory service life on military vehicles which operate both on the road and cross country, springs must possess a fatigue endurance of 45,000 cycles when tested from a small preload to maximum operating stress. Shot peening the bars permits the attainment of this life even if the operating stress is almost equal to the yield stress of the material. Shot peening will not reduce the settling; operating stress is then limited by the permissible set in service.

Materials currently available for torsion bar springs can be used at a hardness of approximately Rockwell C-50. In that condition their yield point in torsion lies between 115,000 and 120,000 psi. A nominal stress higher than this may be used if favorable trapped stresses exist; the permissible nominal operating stress then depends on the method of processing. For the processing schedule described previously, the maximum operating stress for suspension torsion bars should be 140,000 psi with a shear modulus of elasticity of 10.5×10^6 psi. The settling in percent of maximum deflection should range from 2 to 4 percent. The operating stresses given above may be used only if the bars are loaded in pure torsion. For more detailed information on the limits of nominal operating stresses obtained with different manufacturing processes see Ref. 32.

SECTION VIII VIBRATION DAMPING

10-20 THEORY OF DAMPING (Refs. 49, 50)

The purpose of damping with respect to vehicular suspension systems and a description of the basic types of damping associated with these systems are discussed in Section II of Chapter 8. In addition, basic terminology with respect to the current subject is presented in Section III of Chapter 1. The present discussion considers some of the basic analytical aspects of the damping concept.

Damping, as it applies to vibratory systems, is the dissipation of energy with time or distance. There are several basic types of damping forces, three of which are discussed in this section. Additional information on types of damping force is found in sections of this chapter.

Viscous damping, is the dissipation of energy that occurs when a particle in a vibrating system is resisted by a force that has a magnitude proportional to the magnitude of the velocity of the particle and direction opposite to the direction of the particle. Viscous damping is encountered by bodies moving at moderate speeds through a fluid. This type of damping produces a resisting force F_v, proportional to velocity V, expressible as

$$F_v = -CV \qquad (10\text{-}100)$$

where

C is a constant of proportionality.

Hydraulic damping is characterized by a resisting force that is proportional to the second power of the velocity. This relationship is attained by forcing fluid of low viscosity through a sharp-edged orifice.

Coulomb or *dry friction damping* is the dissipation of energy that occurs when a particle in a vibrating system is resisted by a force whose magnitude is a constant independent of displacement and velocity, and whose direction is opposite to the direction of the velocity of the particle. Friction damping arises from the sliding contact of dry surfaces. The force F_c developed is nearly constant and depends on the nature of the sliding surfaces, the area of contact S, and the unit pressure P_n between them as expressed by the equation

$$F_c = \mu S P_n \qquad (10\text{-}101)$$

where

μ = coefficient of kinetic friction dependent on the nature of the surfaces

Since this type of friction is constant, it frequently is the predominating damping force during the final stages of motion when other types of damping become negligible.

Solid damping or *structural damping* results from internal friction within the vibrating material itself. Solid damping differs from viscous damping in that it is independent of frequency and proportional to the maximum stress of the vibration cycle Since stress and strain are proportional in the elastic range of a material, the solid damping force is proportional to the deflection of the member.

The influence of damping in vibratory systems can be investigated by considering a simple, single degree of freedom system with viscous damping. Other forms of damping will be considered in subsequent sections of this chapter.

10-20.1 DAMPED FREE VIBRATIONS (Ref. 51)

A simple damped vibrating system having a single degree of freedom (described by a single coordinate, x) is shown in Figure 10-29. The viscous damping force F_v is proportional to the velocity of the oscillating mass M and can be expressed by the equation

$$F_v = -C\dot{x}, \text{ lb} \qquad (10\text{-}102)$$

where, C = coefficient of viscous damping, lb-sec/in.

The equation of equilibrium for the system shown in Figure 10-29 is

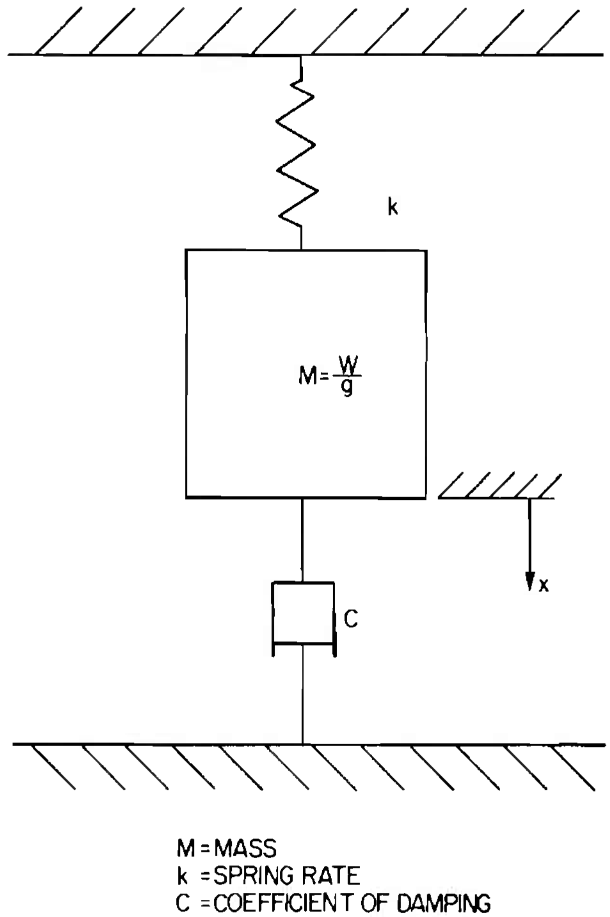

Figure 10-29. Simple Damped Vibrating System

$$\frac{W}{g}\ddot{x} + C\dot{x} + kx = 0 \qquad (10\text{-}103)$$

where

W = weight of the oscillating system, lb
g = acceleration due to gravity, in./sec^2
k = spring rate of the oscillating system, lb/in.
x = distance (along the coordinate of the system), in.
$\dot{x}$ = velocity of the oscillating system (first derivative of x with time) in./sec
$\ddot{x}$ = acceleration of the oscillating system (second derivative of x with time) in./sec^2

It is convenient to substitute in Eq. 10-103

$$K = \frac{Cg}{W} \qquad (10\text{-}104)$$

$$\omega^2 = \frac{kg}{W} \qquad (10\text{-}105)$$

where

ω = natural frequency (circular) of the oscillating system, without damping, rad/sec

then

$$\ddot{x} + k\dot{x} + \omega^2 x = 0 \qquad (10\text{-}106)$$

Assume a solution to Equation 10-106 of the form

$$x = Ae^{\alpha t} \qquad (10\text{-}107)$$

where

A and α = constants to be determined
t = time, sec

then

$$\dot{x} = \alpha\, Ae^{\alpha t} = \alpha\, x \qquad (10\text{-}108)$$

and

$$\ddot{x} = \alpha^2\, Ae^{\alpha t} = \alpha^2\, x \qquad (10\text{-}109)$$

Substituting these values in Equation 10-106, gives

$$\alpha^2 x + K\alpha x + \omega^2 x = 0 \qquad (10\text{-}110)$$

and since x is not zero for any motion, Equation 10-110 becomes

$$\alpha^2 + K\alpha + \omega^2 = 0 \qquad (10\text{-}111)$$

Solving for α

$$\alpha_{1,2} = \frac{-K}{2} \pm \frac{\sqrt{K^2 - 4\,\omega^2}}{2} \qquad (10\text{-}112)$$

Three cases can be considered

(a) When $K^2 > 4\,\omega^2$ so that $\sqrt{K^2 - 4\,\omega^2}$ is real
(b) When $K^2 = 4\,\omega^2$ in which case $\alpha = -K/2$
(c) When $K^2 < 4\,\omega^2$ so that $\sqrt{K^2 - 4\,\omega^2}$ is imaginary.

These cases will be considered in order.

Case a. $K^2 > 4\,\omega^2$. Since the radical $\sqrt{K^2 - 4\,\omega^2}$ will be less than K, the two values of α given by Equation 10-112, will be negative. If these values are denoted as $-\gamma$ and $-\delta$, the complete solution of Equation 10-106 may be written as

$$x = D_1 e^{-\gamma t} + D_2 e^{-\delta t} \qquad (10\text{-}113)$$

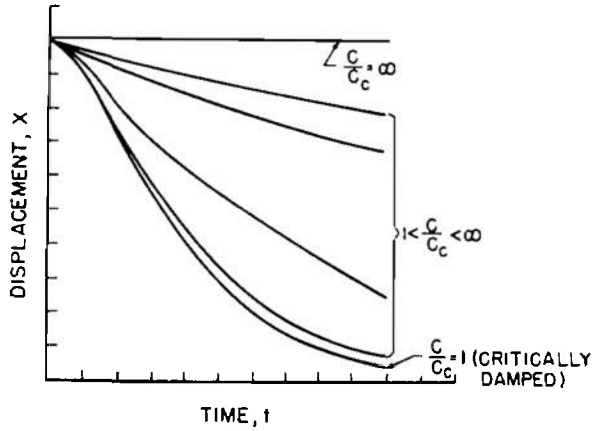

Figure 10-30. Typical Decay Curves of Overdamped Systems

The values of constants D_1 and D_2 are determined by the initial conditions of the system. For example, if $x = X$ and $\dot{x} = 0$ when $t = 0$, Equation 10-113 becomes

$$X = [D_1 e^{-\gamma t} + D_2 e^{-\delta t}]_{t=0} = D_1 + D_2 \qquad (10\text{-}114)$$

and, after differentiation, with respect to t, to obtain $\dot{x}$

$$0 = [-\gamma D_1 e^{-\gamma t} - \delta D_2 e^{-\delta t}]_{t=0} = \gamma D_1 + \delta D_2 \qquad (10\text{-}115)$$

therefore

$$D_1 = \left(\frac{\delta}{\delta - \gamma}\right) X \qquad (10\text{-}116)$$

$$D_2 = \left(\frac{\gamma}{\gamma - \delta}\right) X \qquad (10\text{-}117)$$

and

$$x = \left(\frac{\delta}{\delta - \gamma}\right) Xe^{-\gamma t} + \left(\frac{\gamma}{\gamma - \delta}\right) Xe^{-\delta t} \qquad (10\text{-}118)$$

The condition described in this case is called overdamping, and the motion is *aperiodic*. The mass moves slowly back to the equilibrium position after being displaced. Figure 10-30 shows the decay curves for several overdamped systems. The relative amount of damping in a system can be specified by the *damping factor* which is the ratio between the actual damping coefficient C and the

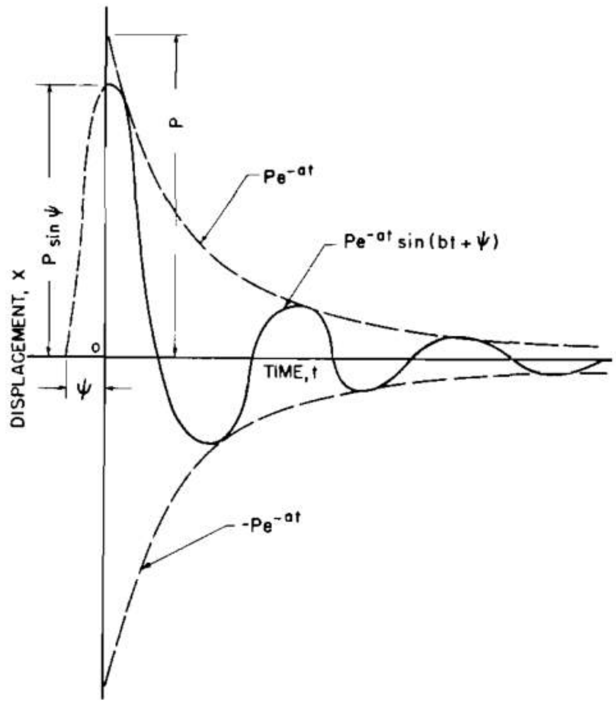

Figure 10-31. Typical Decay Curve of Underdamped System

critical damping coefficient C_c. The concept of critical damping is discussed under Case b.

Case b. $K^2 = 4\,\omega^2$. In this case, the two roots of Equation 10-111 are equal to $-K/2$. This would give a solution to Equation 10-106 having only one constant of integration and as such cannot be the complete solution. It can be shown that a complete solution is

$$x = (E + Ft)e^{-\frac{Kt}{2}} \qquad (10\text{-}119)$$

where E and F are arbitrary constants.

The values of the constants are determined by the initial conditions. If these are the same as Case a, $x = X$ and $\dot{x} = 0$ when $t = 0$; Equation 10-119 becomes

$$x = X\left(1 + \frac{Kt}{2}\right)e^{-\frac{Kt}{2}} \qquad (10\text{-}120)$$

The motion is a return towards the equilibrium position as shown by the critically damped curve on Figure 10-30. The return is more rapid than that of any overdamped system. It is the most rapid aperiodic motion obtainable. The damping in this case is known as *critical damping*.

The amount of damping in any system can be specified in terms of a nondimensional ratio known as the damping factor ζ.

$$\zeta = \frac{C}{C_c} \qquad (10\text{-}121)$$

where

C_c is the critical damping coefficient
C is the actual damping coefficient.

As shown by Figure 10-30, the decay curves vary from no return motion (C or $C/C_c = \infty$) to the limiting case of critical damping ($C/C_c = 1$).

Case c. $K^2 < 4\,\omega^2$. In this case the radical $\sqrt{K^2 - 4\,\omega^2}$ is imaginary. A solution to Equation 10-111 can be written if the roots are assumed to be: $\alpha_1 = -a + bi$ and $\alpha_2 = -a - bi$ $(i = \sqrt{-1})$. Then $a = K/2$ and $b = \sqrt{4\,\omega^2 - K^2}/2$, and the solution is

$$x = Ge^{(-a+bi)t} + He^{(-a-bi)t} \qquad (10\text{-}122)$$

$$x = e^{-at}\left[Ge^{bit} + He^{-bit}\right] \qquad (10\text{-}123)$$

Equation 10-123 can be written as

$$x = Pe^{-at}\sin(bt + \psi) \qquad (10\text{-}124)$$

Equation 10-124 shows that the motion consists of a simple harmonic motion, represented by $P\sin(bt + \psi)$ in which all the displacements are modified by the factor e^{-at}. Since this quantity continuously decreases as t increases, the motion is an oscillatory one in which the amplitude gradually decreases (Figure 10-31). The constants G, H, and P, and phase angle ψ are determined from the initial conditions of the system.

The *periodic time* of the motion (not strictly a period since the motion does not repeat exactly) represented by Equation 10-124, is

$$T_K = \frac{2\pi}{b} = \frac{4\pi}{\sqrt{4\omega^2 - K^2}} \qquad (10\text{-}125)$$

If there is no damping in the system, i.e., $K = 0$, the periodic time and, in this case the true period, would be

$$T = \frac{2\pi}{\omega} \qquad (10\text{-}126)$$

Equations 10-125 and 10-126 show that damping increases the periodic time.

The determination of the damping coefficient of a system can be done experimentally by observing and measuring the values of the amplitude of the vibration at the beginning and the end of an interval during which a whole number of vibrations occurs. If we assume that n vibrations occur, and that the amplitudes are x_1 and x_{1+n} (at the beginning and end of motion, respectively) the ratio between them becomes

$$\frac{x_1}{x_{1+n}} = \frac{e^{-at}}{e^{-a(t+2n\pi/b)}} = e^{2a\,n\pi/b} \qquad (10\text{-}127)$$

By taking the logarithms of both sides there results

$$ln\left(\frac{x_1}{x_{1+n}}\right) = ln\,e\left(\frac{2\pi a\,n}{b}\right) = \frac{2\pi\,a\,n}{b} \qquad (10\text{-}128)$$

then

$$\frac{a}{b} = \frac{ln\left(\frac{x_1}{x_{1+n}}\right)}{2\pi\,n} = \frac{K}{\sqrt{4\,\omega^2 - K^2}} \qquad (10\text{-}129)$$

Since from Eq. 10-104

$$C = \frac{KW}{g} \qquad (10\text{-}130)$$

Equation 10-129 solved for K and combined with Equation 10-130 gives the damping coefficient

$$C = \left(\frac{2W\omega}{g}\right)\frac{\left(\frac{a}{b}\right)}{\sqrt{1 + \left(\frac{a}{b}\right)^2}},\ \text{lb-sec/in.} \qquad (10\text{-}131)$$

10-20.2 DAMPED FORCED VIBRATIONS (Ref. 52)

The previous discussion considered free vibrations. An important vibratory phenomenum with respect to vehicle design is the forced vibration. *Forced vibration* occurs in a spring-mass oscillatory system if the response is imposed by the excitation. If the excitation is periodic and continuing, the oscillation is steady-state.

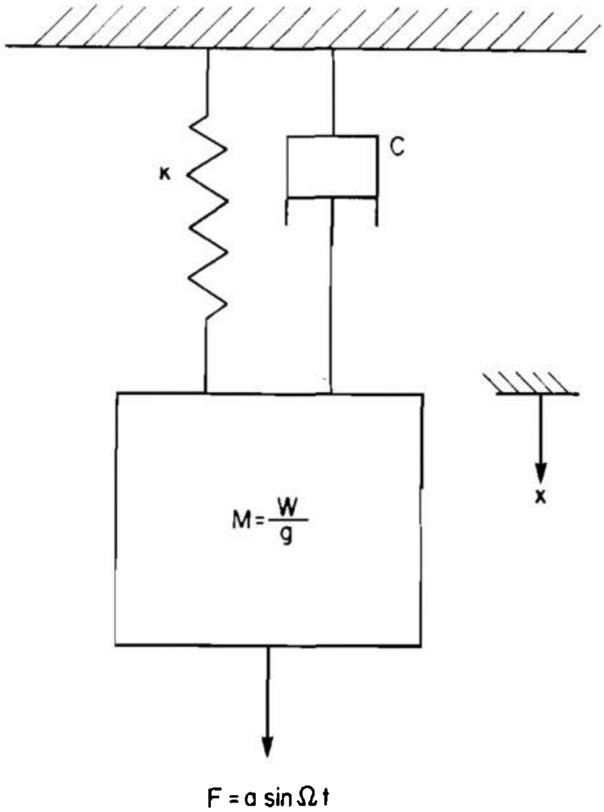

Figure 10-32. Forced Vibration With Viscous Damping

Although the forced vibrations experienced by an automotive vehicle usually are not periodic, the assumption of a periodic stimulus applied to the simple single degree of freedom system of paragraph 10-20.1 will facilitate the examination of damped forced vibrations. Viscous damping will be assumed again.

Figure 10-32 shows a system with a disturbing force represented by $F = a \sin \Omega t$. The equation of motion is (see Eq. 10-103 for comparison)

$$\frac{W}{g}\ddot{x} + C\dot{x} + kx = a \sin \Omega t \qquad (10\text{-}132)$$

or

$$\ddot{x} + K\dot{x} + \omega^2 x = A \sin \Omega t \qquad (10\text{-}133)$$

where

$$K = \frac{Cg}{W} \qquad (10\text{-}134)$$

$$\omega^2 = \frac{kg}{W} \qquad (10\text{-}135)$$

$$A = \frac{ag}{W} \qquad (10\text{-}136)$$

Units are the same as for Equations 10-103 to 10-107, except

a = magnitude of forced vibration, lb
Ω = circular frequency of the forced vibration, rad/sec

It can be shown that the solution to Equation 10-132 consists of two parts (a) the complementary function, and (b) the particular solution. The complementary function is the solution of the equation $\ddot{x} + K\dot{x} + \omega^2 x = 0$. This has been treated in the preceding paragraphs covering free damped vibration. The particular solution is any function of x and t which will satisfy Equation 10-133. The complete (general) solution is the sum of these two parts.

As shown in paragraph 10-20.1, the complementary function may take one of three forms depending on the relative magnitudes of the values of K^2 and $4\omega^2$. In all of these cases, the motion tends to diminish as the period of time increases. For a system under damped forced vibration, when the motion is reduced by the damping to one of relatively small magnitude, the motion represented by the particular solution will remain. This remaining motion is the forced vibration.

A function which will satisfy Equation 10-132 as the particular solution is known to be

$$x = B \sin(\Omega t + \varphi) \qquad (10\text{-}137)$$

where

B = amplitude of the forced vibration
φ = phase angle by which the motion lags the impressed force

If Equation 10-137 is differentiated twice with respect to time and the results substituted in the equation of motion for damped forced vibrations (Eq. 10-133), the following complete solution can be written

$$\left.\begin{aligned} x_1 &= D_1 e^{-at} + D_2 e^{-bt} \\ x_2 &= (E + Ft)e^{-\frac{Kt}{2}} \\ x_3 &= Pe^{-at} \sin(bt + \psi) \end{aligned}\right\} + \frac{A \sin(\Omega t + \varphi)}{\sqrt{(\omega^2 - \Omega^2)^2 + K^2\Omega^2}} \qquad (10\text{-}138)$$

A relation between amplitude of forced vibration B and term A (defined by Eq. 10-136) was obtained in the evaluation of the complete solution. This relation is included in the given form of Equation 10-138.

$$B = \frac{A}{\sqrt{(\omega^2 - \Omega^2)^2 + K^2\Omega^2}}, \text{ in.} \qquad (10\text{-}139)$$

The appropriate form x_1, x_2, or x_3 of Equation 10-138 is used, respectively, according to whether K^2 is greater than, equal to, or less than $4\omega^2$. The values of D_1, D_2, E, F, P, and ψ are determined from initial conditions.

As shown by Equation 10-139, the amplitude B of the forced vibration depends on the relationship between the natural frequency ω of the system and the frequency of the forcing function Ω. When Ω is very small, $B \cong A/\omega^2 = a/k$, which is the static deflection of the spring under a force equal to the maximum value of the exciting force. When Ω is relatively large with respect to ω, $B \cong 0$.

The maximum amplitude B_{max} of the forced vibration is expressed by the equation

$$B_{max} = \frac{2A}{K\sqrt{4\omega^2 - K^2}}, \text{ in.} \qquad (10\text{-}140)$$

When this condition exists, the frequency of the forced motion has the value Ω_m which is expressed by the equation

$$\Omega_m = \sqrt{\frac{2\omega^2 - K^2}{2}}, \text{ rad/sec} \qquad (10\text{-}141)$$

As shown by Equation 10-140, the maximum value of the amplitude of the forced motion depends on the value K and hence on the damping coefficient C. If no damping were present in the system, then the amplitude would be infinite. For any finite amount of damping, the maximum amplitude is finite but may be large. When the forced motion has the value of frequency Ω_m, the system is said to resonate. *Resonance* of a system in forced vibration exists when any change, however small, in the frequency of excitation causes a decrease in the response of the system. The resonant frequency f_r is expressed as

$$f_r = \frac{\Omega_m}{2\pi}, \text{ cps} \qquad (10\text{-}142)$$

When the damping is not great, the resonant frequency of the damped system will approach that of the free undamped vibration of the system.

The value of the phase angle φ also depends on the relationship between ω and Ω as well as the amount of damping in the system. The value of the phase angle by which the motion lags the impressed force corresponding to resonance φ_m is given by the equation

$$\tan \varphi_m = \frac{\sqrt{4\omega^2 - K^2}}{K} \qquad (10\text{-}143)$$

The ratio of the amplitude of the forced vibration to the static deflection, i.e., the deflection under a steady force equal in magnitude to the maximum value of the exciting force, is termed the magnification factor. The *magnification factor* M can be expressed by the equation

$$M = \frac{B}{\frac{a}{k}} \qquad (10\text{-}144)$$

If the frequency ratio $\Omega/\omega = r$, the equation for the magnification factor becomes

$$M = \frac{1}{\sqrt{(1 - r^2)^2 + \left(\frac{Kr}{\omega}\right)^2}} \qquad (10\text{-}145)$$

The maximum value of the magnification factor M_{max} is obtained at resonance when $\Omega = \Omega_m$. This can be written as

$$M_{max} = \frac{2\omega^2}{K\sqrt{4\omega^2 - K^2}} \qquad (10\text{-}146)$$

The response of the system at the natural frequency, as opposed to the resonant frequency, is obtained by setting $\Omega = \omega$. For this condition the magnification factor becomes

$$M = \frac{\omega}{K} \qquad (10\text{-}147)$$

The preceding discussion reveals some of the basic influences that viscous damping has on the behavior of vibratory systems. Other types of damping will influence the system in characteristic ways. For example, friction or constant damping in a system under free vibration causes an amplitude decrease at a constant rate (the envelopes of the decay curve are straight lines). In all cases, however, there is a dissipation of energy resulting in a decrease in the motion of the system.

10-20.3 REQUIREMENTS AND CONTROL (Ref. 53)

The approach to the problem of suspension system damping requirements has been largely experimental. Experimental evaluation of damping requirements is conducted on actual vehicles or by means of analog computer simulation. In either case, the experimental approach should be guided by several proven principles. These principles are summarized below.

(a) Design techniques should be employed which reduce the inherent damping of the suspension to a minimum. This permits better control of the total damping applied to the system.

(b) Damping is essential when the frequency of the road disturbances are at or near the natural frequency of two suspension systems.

(c) Vehicles with relatively soft (low rate) springs require less damping in terms of energy absorbed than those with hard springs but damping is more essential to maintain stability of the vehicle.

(d) The aim is to damp out as quickly as possible each vertical impulse induced by the road profile without increasing unduly the maximum accelerations applied to the sprung mass.

(e) A small amount of friction damping is desirable to reduce wheel dance.

10-21 HYDRAULIC DAMPING

10-21.1 CHARACTERISTICS OF HYDRAULIC FLUID DAMPING

10-21.1.1 Basic Types

Three basic types of liquid damping can be recognized, depending on their resistance-velocity

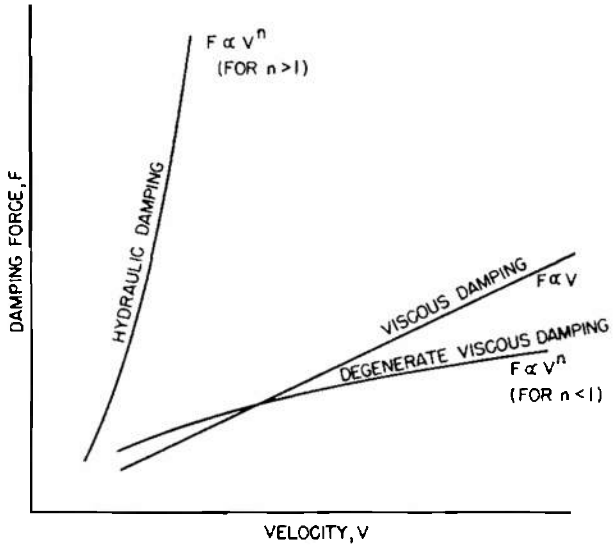

Figure 10-33. Characteristic Curves of Hydraulic Dampers

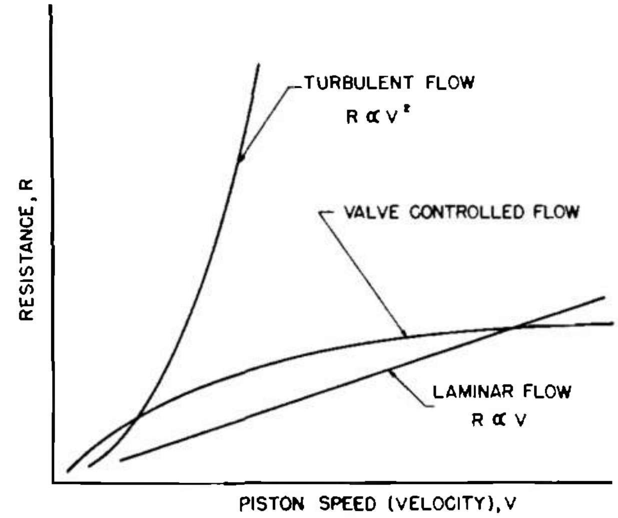

Figure 10-34. Flow-Resistance Curves for Basic Hydraulic Dampers

characteristics (a) *viscous damping* is characterized by a resisting force that has a magnitude proportional to the magnitude of the velocity of the input force; (b) *degenerate viscous* damping is characterized by a resisting force that has a magnitude proportional to the magnitude of a power of the velocity less than unity; and (c) *hydraulic damping* is characterized by a resisting force that is proportional to the second power of the velocity.

Figure 10-33 shows the velocity-force relationship of these damping characteristics. In spite of the generally accepted definitions of liquid damping listed above, the term "hydraulic damping" is used in much of the technical literature to cover all three types of damping. In addition the terms "hydraulic shock absorber" or "hydraulic damper" are used to describe devices which do not operate with true hydraulic damping in the defined sense. In the discussion which follows these terms will be used in their broad sense and the specific type of damping under consideration will be clearly indicated.

Early experiments with hydraulic dampers using simple orifices to control the resistance to flow were found to give resistance-flow curves as shown in Figure 10-34 (Ref. 54). If a single large fixed area orifice is used, the flow is turbulent over the operating range and the resistance varies as the square of the piston velocity. If a sufficient number of smaller fixed orifices is used, the flow through the jets becomes laminar and the resistance to flow varies directly as the piston velocity. This type velocity-resistance behavior is desirable for damping the initial vertical displacement occurring when a wheel strikes and passes over an obstacle since the resistance increases linearly with vehicle velocity for a given obstacle.

For rebound and subsequent related oscillations, the damping required is independent of vehicle speed since the time of recovery after the initial displacement and all subsequent half-oscillations are controlled by the spring periodicity which is constant. The time of initial recovery is independent of the size of the bump; therefore, the fluid velocity through the orifices is directly proportional to the size of the bump and, to a large extent, to the amount of energy to be absorbed. If the resistance to oil flow varies directly as the vertical velocity, the rebound damping approaches the ideal condition.

It is not practicable to utilize the large number of small holes required to achieve laminar flow and the problem of clogging with dirt would be serious. Therefore, modern hydraulic shock absorbers use flow control valves which produce a desired re-

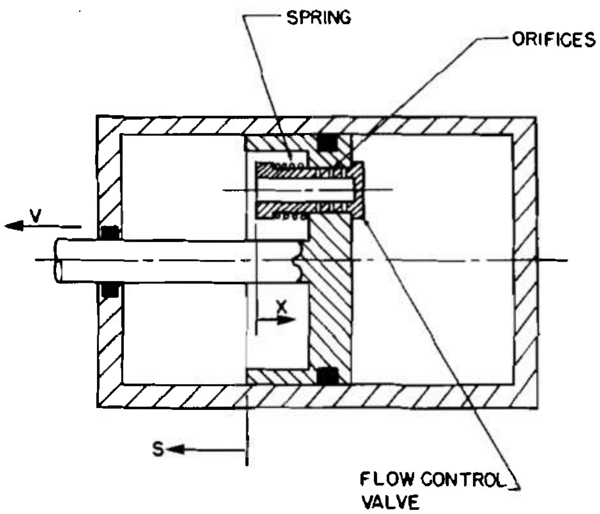

Figure 10-35. Linear Motion Damping Cylinder

sistance-velocity curve. A typical curve of this type is shown in Figure 10-34.

The flow control valve is a spring-loaded valve which opens (initial opening is termed blow-off) to a greater or lesser extent with increase or decrease of fluid pressure (velocity). The effect is that a variable size orifice and, by varying the spring rate and port shape, a wide range of resistance/flow curves can be obtained. General slow speed damping is normally accomplished by bleed orifices built into the valve mechanism which is usually self-cleaning on blow-off.

Another function of the flow-control valve is to provide a degree of compensation for variations in viscosity resulting from temperature changes.

10-21.1.2 Analytical Considerations (Ref. 55)

The discussion that follows applies to linear motion dampers having spring-loaded variable orifice area valves. A schematic of a linear motion damping cylinder is shown in Figure 10-35. As the piston is displaced in direction S, the flow control valve is actuated by the resulting hydraulic pressure, causing a motion in the direction X. This displacement of the valve varies the orifice area, making it smaller or larger depending on the design of the valve.

Under dynamic conditions, the instantaneous rate of flow Q is the product of the area A_p, sq in. and velocity V_p, (in./sec), of the piston

$$Q = A_p V_p, \text{ cu in./sec} \qquad (10\text{-}148)$$

The instantaneous rate of flow through the damping valve orifices can also be written as

$$Q = C_d A_o \sqrt{\frac{2P}{\rho}}, \text{ cu in./sec} \qquad (10\text{-}149)$$

where

C_d = coefficient of discharge, dimensionless
P = instantaneous pressure of fluid in the cylinder, psi
ρ = mass density of the fluid, lb-sec^2/in.^{4}
A_o = instantaneous area of damping valve orifices, sq in.

From Equations 10-148 and 10-149, the pressure can be expressed as

$$P = \frac{\rho A_p^2 V_p^2}{2 C_d^2 A_o^2}, \text{ psi} \qquad (10\text{-}150)$$

The force exerted on the damping valve by the spring must be balanced by the force on the valve caused by the hydraulic pressure. Therefore, the effective area A_v of the damping or flow control valve spool must be

$$A_v = \frac{kX}{P}, \text{ sq in.} \qquad (10\text{-}151)$$

where

X = displacement of the damping valve, in.
k = damping valve spring rate, lb/in.

The relationship between the type of damping and the orifice area A_o, for two types of damping, can be written based on the previous analysis.

Case a. Viscous Damping: By definition, in viscous damping the resisting force F_d is directly proportional to the velocity of the piston. To achieve this state, it is necessary to provide the adequate orifice area so that

$$F_d = c_1 V_p, \text{ lb} \qquad (10\text{-}152)$$

where

c_1 = viscous damping coefficient, lb-sec/in.

The damping force on the piston rod can also be expressed as

$$F_d = A_p P, \text{ lb} \tag{10-153}$$

Equations 10-152 and 10-153 may be combined giving

$$V_p = \frac{A_p P}{c_1}, \text{ in./sec} \tag{10-154}$$

Equation 10-154 combined with Equation 10-150 gives

$$P = \frac{2\, c_1^2 C_d^2 A_o^2}{\rho\, A_p^4}, \text{ psi} \tag{10-155}$$

Substitution of Equation 10-155 into Equation 10-151 and rearranging terms gives the flow control valve displacement

$$X = \frac{2 A_v c_1^2 C_d^2 A_o^2}{\rho\, A_p^4 k}, \text{ in.} \tag{10-156}$$

Equation 10-156 gives the instantaneous value of X in terms of orifice area A_o since all other factors are design constants. Turbulent flow through the orifices is assumed.

Case b. Hydraulic Damping. In this case, the damping force is proportional to the square of piston velocity. To achieve this condition, the orifice area A_o must be such that

$$F_d = c_2 V_p^2, \text{ lb} \tag{10-157}$$

where

c_2 = hydraulic damping coefficient, lb-sec²/in.²

The velocity V_p of the piston can be expressed from Equations 10-157 and 10-153 as

$$V_p = \sqrt{\frac{A_p P}{c_2}}, \text{ in./sec} \tag{10-158}$$

Substitution of Equation 10-158 into Equation 10-150 and simplifying gives

$$A_o = \sqrt{\frac{\rho\, A_p^3}{2\, C_d^2 c_2}}, \text{ sq in.} \tag{10-159}$$

Equation 10-159 indicates that a fixed area orifice is required since all of the factors on the right side are design constants.

10-21.2 GENERAL DESCRIPTION (Ref. 56)

10-21.2.1 Single-Acting, Cam-Operated

A typical single-acting, cam-operated hydraulic shock absorber is shown in Figure 10-36. This type of shock absorber provides significant damping in one direction only. Normally, a single-acting damper is installed so that damping occurs during rebound. When the sprung and unsprung masses of the suspension move toward each other, the shock absorber arm rotates counterclockwise, moving the cam to the right, thereby permitting the piston spring to move the piston to the right. This causes the intake valve in the piston to open and allows oil to flow from the reservoir into the increasing cylinder volume. Because the piston motion and oil flow are caused by the piston spring, the shock absorber has little effect on the suspension during this phase of the motion. During rebound the cam moves to the left, forcing the piston to the left against the oil in the cylinder. The intake valve is closed by the pressure and the oil in the cylinder is forced out through the relief valve. The controlled flow through the relief valve generated the damping force.

10-21.2.2 Opposed-Cylinder, Double-Acting

One type of double-acting hydraulic shock absorber is shown in Figure 10-37. The two opposing pistons are rigidly interconnected at their reservoir ends where they straddle the cam. Each piston is equipped with a spring-loaded intake valve which permits the one-way flow of oil from the reservoir to the cylinder. When the suspension spring is compressed, the cam rotates counterclockwise, carrying both pistons to the right, thereby creating pressure in the right cylinder. When the pressure reaches a predetermined level, it forces the compression relief valve open and permits oil to flow into the rebound cylinder. At the same time, the intake valve in the rebound piston may open to permit the flow of make-up oil from the reservoir into the rebound cylinder.

During the suspension spring rebound, both pistons are forced to the left by the cam and the oil is forced through the rebound relief valve, back to the compression cylinder. The resistance to flow of the orifices in the relief valves determines the damping forces. Normally, the orifice in the rebound valve is smaller than the orifice in the compression valve, resulting in greater rebound damping forces.

Some shock absorbers of this type incorporate

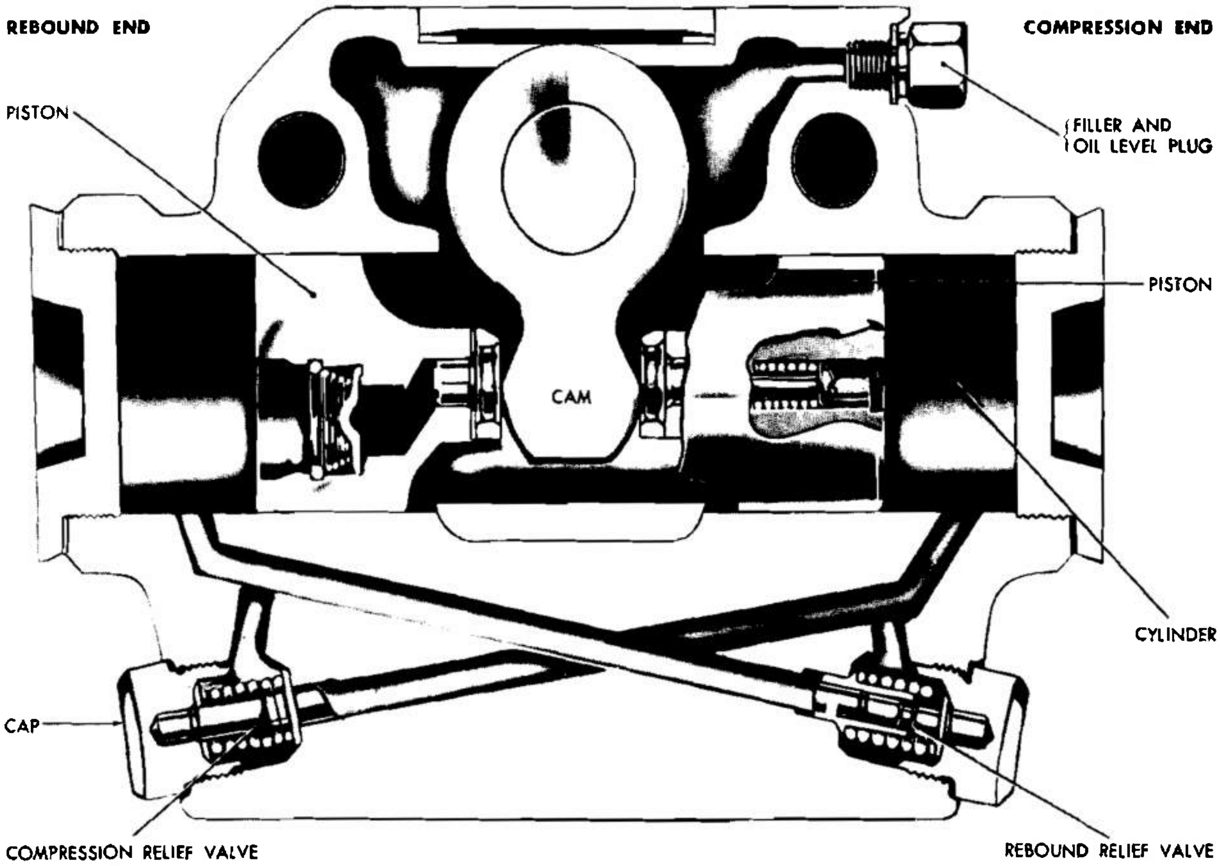

Figure 10-36. Single Acting, Cam-Operated Piston Shock Absorber

a low pressure relief valve in series with the compression relief valve in an attempt to simulate the light friction damping desirable for suppressing wheel dance. However, the effect is inferior to friction damping because of the slower response to high-frequency reversals.

10-21.2.3 Vane Type, Rotary

One example of a vane-type shock absorber is shown in Figure 10-38. The housing of this unit is divided into two working chambers by stationary partitions, each of which contains a check valve. The central shaft, connected to the unsprung mass by means of an arm and link, has a pair of vanes attached to it which extend into each working chamber. As the suspension spring is compressed, the central shaft rotates, and the vanes develop pressure in the chambers which causes oil to flow, unrestricted, through the open check valves in the stationary partitions. On the rebound stroke, the vanes develop pressures on the opposite sides, closing the check valves. Since oil cannot flow through the check valves, it is forced through the needle valve in the center of the shaft, thereby producing a resistance to motion. The vane-type is not widely used, owing to its relatively high cost and the difficulty in achieving adequate sealing.

10-21.2.4 Direct-Acting

A typical direct-acting shock absorber is shown in Figure 10-39. It consists of three concentric cylinders. The innermost of these contains a double-acting piston which divides the cylinder into upper and lower chambers. The annular vol-

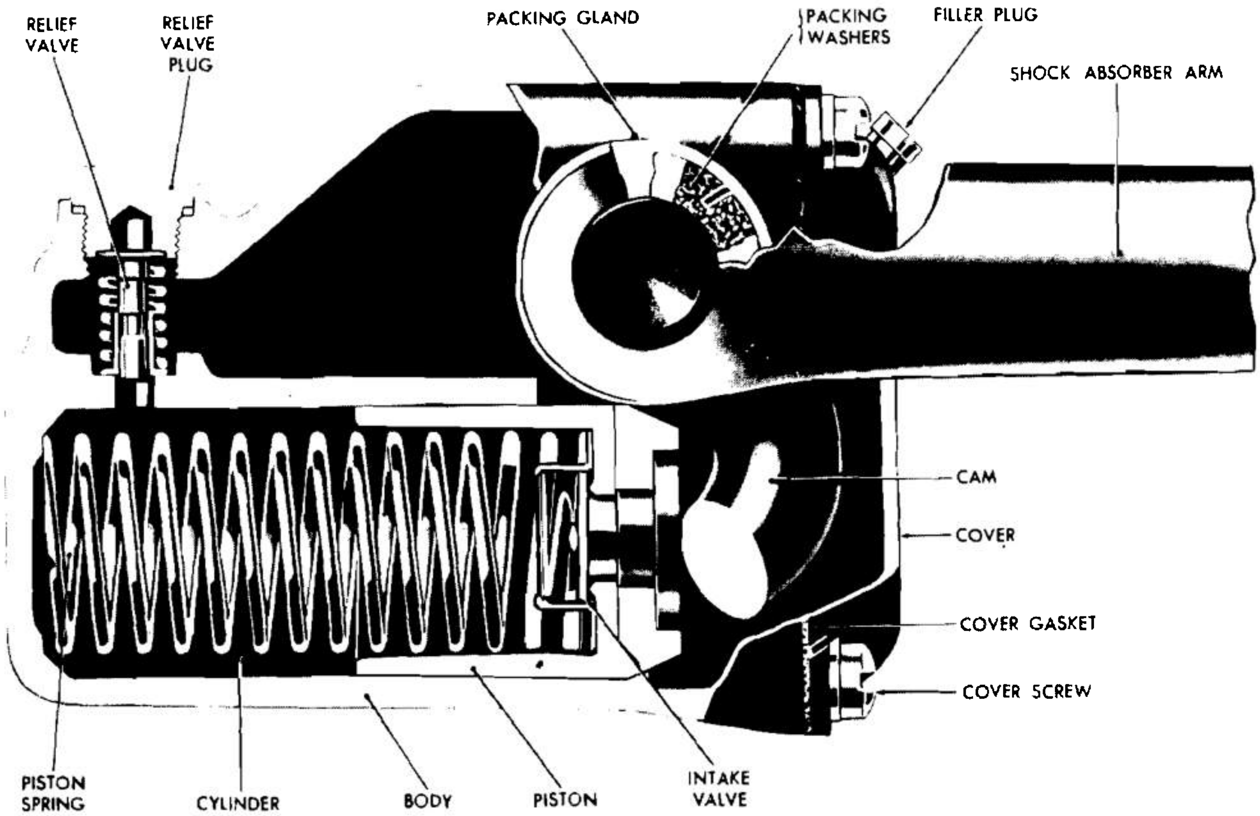

Figure 10-37. Double-Acting, Cam-Operated Shock Absorber (Opposed-Cylinder-Type)

ume between the first and second cylinders serves as an oil reservoir. The outer cylinder serves as a shield for the piston rod. The piston, which is fastened to the actuation rod, contains the compression and rebound orifices and check valves.

During compression of the suspension spring the piston which is connected to the sprung mass, moves downward and forces oil through the compression orifices in the piston to the rod end of the cylinder. Because of the volume occupied by the piston rod, the volume change above the piston is not as rapid as that below; therefore, some of the oil is forced into the reservoir through the reservoir check valve. On the rebound stroke, oil is forced from the rod end to the opposite end of the cylinder through the rebound orifice. Some oil flows from the reservoir to the lower chamber through the reservoir check valve to compensate for the differential volume change caused by the piston rod.

The configuration of the direct-acting shock absorber lends itself well to vehicular applications. During recent years, direct double-acting hydraulic shock absorbers have been used on the majority of military wheeled vehicles and tanks.

Because of the greater piston travel of direct-acting shock absorbers compared with vane- and cam-operated types for a given level of damping, their operating pressure is lower and the sealing problem is less severe. The pressures in the direct-acting types are about 400 psi compared with 1500 psi for the other types. However, the "t" direct-acting shock absorber experiences a greater temperature rise than the other standard hydraulic types. This is because the bodies of the cam-operated and vane-types are in intimate contact with substantial structural members which improve their heat dissipation rates, whereas, the direct-acting type is attached to the vehicle by insulating rubber

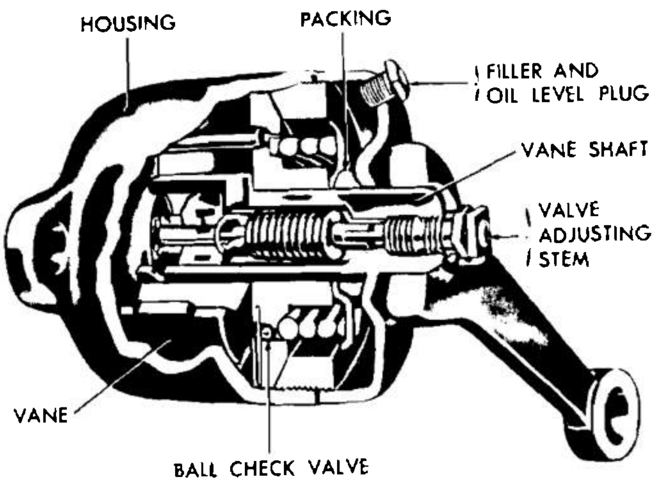

Figure 10-38. Double-Acting, Vane-Type Shock Absorber

bushings and depends almost entirely upon the surrounding air for its heat dissipation.

10-21.3 FIXED-ORIFICE TYPES

As stated in paragraph 10-21, the behavior of dampers utilizing constant-area metering orifices depends on the relative sizes of the jets. The most practicable type of fixed orifice shock absorber produces a resisting force which is proportional to the square of the velocity of the piston (or fluid). Characteristics of dampers having this type of resisting force are discussed in paragraph 10-21.1.2.

10-21.4 VARIABLE-ORIFICE TYPES (Ref. 57)

Two basic kinds of variable orifice shock absorbers are used; namely, (a) spring loaded valve types and (b) metering rod types.

Spring load valve dampers have been discussed in paragraph 10-21.1.2. A metering rod damper achieves a variable orifice effect by utilizing a fixed orifice and a tapered needle or rod that fits into and moves relative to the orifice. The position of the tapered rod, with respect to the orifice, determines the fluid-discharge area which is carefully calculated to give a desired pressure-deflection response.

If a constant pressure device is desired, the orifice area A_o required for any point in the stroke is given by the following equation

$$A_o = \left[\frac{wA_p^3(V_1^2 - 2aS)}{288\, c^2 Wa} \right]^{1/2}, \text{ sq in.} \tag{10-160}$$

where,

w = specific weight of fluid, lb/cu ft
A_p = piston area, sq in.
V_1 = initial piston velocity, fps
a = acceleration of the mass to be damped, fps^2
S = piston travel, ft
c = orifice discharge coefficient, dimensionless
W = weight of the moving mass to be damped, lb

10-21.5 CONTROLLED-ORIFICE TYPES (Ref. 58)

Except for the suppression of wheel dance, damping of the suspension spring is undesirable when the sprung mass does not experience significant acceleration. For this reason, it would be desirable to vary the degree of damping of a particular suspension system to suit the various conditions encountered during operation. An attempt in this direction has been made by incorporating an inertia-sensitive valve into the hydraulic shock absorber body, in parallel with the rebound relief valve. This valve is normally open for average shock conditions. It is closed by a spring-suspended weight when the sprung mass experiences substantial acceleration during rebound. This action automatically increases the damping momentarily. This system is not widely employed, however, since the added complexity of the shock absorber adversely affects its reliability and cost.

Another approach toward improving the damping behavior of a military vehicle is expressed by the experimental systems using terrain sensor-actuated variable orifices. These electro-hydraulic systems attempt to preset the dampers to suit the impending terrain conditions by means of electrical or electronic sensors.

10-22 FRICTION DAMPING (Ref. 59)

10-22.1 CHARACTERISTICS OF FRICTION DAMPING

Idealized dry friction damping, as expressed by Equation 10-101, is defined as a constant resistance

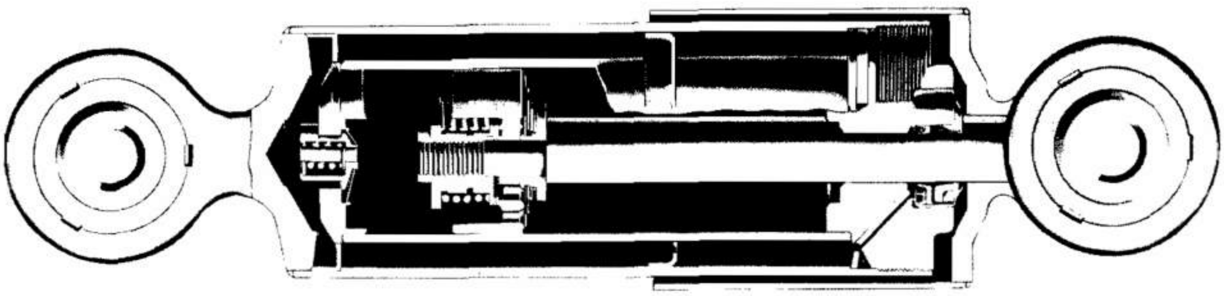

Figure 10-39. Direct-Acting Shock Absorber

to motion, regardless of relative velocity. It is rarely found in practice as the result of rubbing between dry surfaces. Actual dry friction is characterized by resistance to motion that is inversely proportional to the velocity. The force-resisting motion is a function of the force holding the two surfaces together, the materials in contact, and the nature of the surfaces, as well as the velocity.

For most operational conditions experienced, there is a direct relationship between the amplitude of deflection, shock input, and relative velocity of sprung to unsprung components. It is possible to design a friction shock absorber that will produce the desired damping for a given set of conditions; however, it is difficult to obtain the proper degree of damping for all conditions which the vehicle will experience. A friction damper adequate for extreme deflections would be too harsh for average deflections.

Another disadvantage of the friction shock absorber stems from the fact that static friction is substantially greater than sliding friction. This phenomenon contributes to a harsh ride, especially when operating on terrain having only moderate irregularities and with a vehicle equipped with double-acting shock absorbers. The shock must exceed the break away frictional force before the suspension spring will deflect. As soon as relative motion begins the vehicle body experiences a vertical lurch resulting from the sudden decrease in resistance.

Although friction damping characteristics are not suited to the full range of damping requirements, a small amount of friction damping is desirable to suppress wheel dance.

10-22.2 GENERAL DESCRIPTION

Two basic types of dry friction shock absorbers have been developed to satisfactory levels of performance. A design widely used in the automotive field in past years consists of a pair of arms, each enlarged at one end to form flat disks. One side of each of these disks is covered with a suitable friction material and placed in contact with each other so as to rotate about a pivot through their center. A clamping force is applied to the friction surfaces through a spider spring which, in turn, is loaded by a threaded connection on the pivot pin. The opposite ends of the arms are hinged, one to the sprung portion of the vehicle and the other to the unsprung portion. Damping occurs whenever there is relative motion between these bodies.

A recent application of a friction shock absorber to a medium tank has proven successful. This device derives its damping force from the friction of a brake-lining type material pressed against the inside surface of a steel tube. Although the ride quality over the entire speed range of the tank was inferior to that of similar tanks equipped with other types of shock absorbers, the greater durability of the friction type warranted its selection.

10-23 PNEUMATIC DAMPING

Pneumatic damping is feasible in a suspension system using pneumatic springs. The application of pneumatic damping in this type of elastic support system is discussed in Section VII of this chapter.

10-24 ELECTROMAGNETIC DAMPING

10-24.1 AIR-GAP TYPE (Ref. 60)

Air gap or eddy-current damping produces a damping force directly proportional to velocity. The eddy-current damping force can be attained by moving a short-circuited coil through a magnetic field. Both the direction of current flow in the conductor and the direction of motion of the conductor are perpendicular to the direction of the lines of flux of the magnetic field.

For a device of this type, the damping force is directly proportional to the velocity of the conductor in the magnetic field, and is defined quantitatively by the coefficient c of viscous damping.

$$c = \frac{5.6\, B^2\, l\, S}{\rho} \times 10^{-6}, \text{ lb-sec/in.} \quad (10\text{-}161)$$

where

B = flux density, gauss
l = length of the conductor, in.
S = cross sectional area of the conductor, sq in.
ρ = resistivity of the conductor, ohms-circular mil/ft

Equation 10-161 applies only if the entire conductor is in the magnetic field.

10-24.2 MAGNETIC-FLUID TYPE

It is known that a slurry composed of fine iron particles in a fluid medium will stiffen substantially when subjected to a magnetic field. The application of this phenomenon to the regulation of flow rate and resistance force associated with hydraulic-type dampers suggests itself. Research programs have been conducted to investigate the feasibility of developing attenuator systems which would use magnetic fluids. Experiments performed during one program indicate when a magnetic fluid device is configured and energized to generate the required maximum force at low velocity, the force generated at high velocity is excessive. The density of the unmagnetized fluid is the principal cause of these conditions.

The dynamic response of the magnetic fluid system was found to be unfavorable for impact attenuation when used in devices based on controlled fluid flow.

10-25 DYNAMIC DAMPING

A dynamic vibration absorber is an auxillary mass-spring system which tends to neutralize vibration of a structure or body to which it is attached. The basic principle of operation is vibration out-of-phase with the vibration of the main structure or body, thereby creating a counteracting force.

As applied to vehicles, the dynamic absorber is a small spring-mass system mounted on the unsprung mass. It is turned in such a manner that it vibrates in antiphase to, and therefore opposes, an exciting force of a specific frequency. The frequencies at which the absorber must be effective are (a) the natural frequency of the sprung mass, and (b) the natural frequency of the unsprung mass. The small spring mass system usually has its own small frictional damper.

The advantages claimed for the dynamic absorbers are simplicity, isolation of the absorber from the sprung mass, and reduction of the power consumption and overheating problem present with conventional dampers. The dynamic absorber does, however, increase the unsprung mass, thereby offsetting, to some extent, the advantages gained.

10-26 FREQUENCY-CONTROLLED SHOCK ABSORBER

Another approach toward improving the damping characteristics of hydraulic fluid shock absorbers depends upon the fact that both the free vibrations and the resonant vibrations occur at the same frequency. By incorporating a damping valve in the shock absorber that is sensitive to the natural frequency, ideal damping is approached. However, a suitable frequency-controlled shock absorber has not been developed at present.

10-27 SELECTION AND DESIGN OF VIBRATION DAMPERS

Current military vehicles, excluding experimental types, use leaf springs, coil springs, or torsion bars as the elastic medium of the suspension system. These springs—in conjunction with the various linkages, supporting and drive members, as well as the spring mass—result in very diverse dynamic behavior characteristics for different vehicles. The specific damping requirements for the different vehicles may vary considerably; however, the general purpose of the shock absorbers is to damp out, as quickly as possible, each vertical impulse given by the terrain without increasing unduly the maximum accelerations applied to the sprung mass.

The amount of damping required for a given vehicle can be calculated approximately by taking into consideration the weights, spring rates, and other factors, as well as the anticipated road profile. However, for the final optimization of the damping requirements, extensive testing of the given vehicle is necessary.

The analytical determination of the damping requirements can be facilitated by analog computor simulation. This technique is outlined in Chapter 8.

SECTION IX BUMP STOPS

10-28 PURPOSE

Bump stops are provided in the suspension systems of tracked vehicles to limit the vertical displacement of the road wheels. Some of the various types of bump stops have impact isolation capabilities which reduce the dynamic loading on the structural members.

10-29 TYPES

The bump stops that have been used for tracked vehicles can be classified into two types—mechanical and hydraulic. The mechanical systems include (a) metal-to-metal contact between the road wheel arm and the hull, (b) volute springs mounted on the hull and contacted by the road wheel arms, (c) elastic pads mounted on the hull and contacted by the road wheel arms, and (d) a structural design that causes the track to be pressed to the sponsor by the road wheels. The hydraulic systems include (a) liquid springs, (b) strut-type shock absorbers (dashpot or tapered rod type), and (c) hydraulic bumpers.

The volute spring and elastic pads using rubber in compression possess nonlinear stiffness characteristics. The nonlinear spring is discussed in Section VII of this chapter. The hydraulic bump stops are either energy absorbers—e.g., liquid springs—or euergy dissipation devices, e.g., the dashpot. The former are discussed in Section VII, and the latter are discussed in Section VIII of this chapter.

SECTION X SUSPENSION LOCKOUTS (Refs. 61, 62)

10-30 PURPOSE

The basic function of a suspension lockout system is to convert a vehicle's sprung mass into unsprung mass. This change permits guns of large recoil impulse to be fired from relatively light vehicles since the reaction is transmitted directly to the ground with minimum carriage movement. Suspension lockouts also confer advantages to a carriage when the vehicle is used with an overhanging boom (mobile crane configuration). Without a lockout system, a vehicle with a boom under load experiences an increase in the overturning moment arm and a decrease in the stabilizing moment of the vehicle as the suspension deflects. This effect takes place regardless of the position of the load with respect to the longitudinal centerline of the vehicle.

10-31 TYPES

10-31.1 HYDRAULIC SUSPENSION LOCKOUTS

Two types of hydraulic suspension lockouts have been developed—an internal or combined unit,

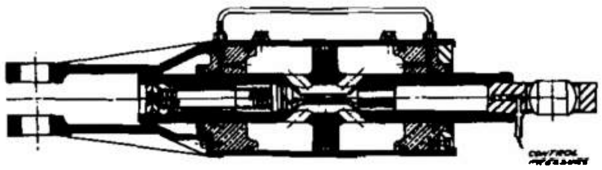

*Figure 10-40. Hydraulic Lockout Cylinder** (Ref. 61)

and an external device. The internal unit utilizes a hydraulic cylinder designed to combine lockout, damping, and bump-stop functions for each road wheel arm. An experimental system of this type has been used on the T236 carriage. While illustrative of the general design, Figure 10-40 is a prototype unit. The construction is that of a double-rod-end cylinder with passages into the valve region in the rod from opposite sides of the piston. In the open position shown, the valve spool is preloaded by the spring against a hollow sleeve stop. To lock the cylinder, control pressure is applied to the spool, depressing the spring until the spool land closes to the right-hand ports in the rod, and trapping the fluid in both ends of the cylinder. A bump-stop action occurs as the piston rod and trapping the fluid in both ends of the end of the stroke.

The lockout cylinder, as pictured, was satisfactory for gun carriage use, as the slight leakage past the sealing land of the spool during the recoil interval was inconsequential; however, when used in the wrecker chassis to lift sustained loads, no leakage could be tolerated. For this service, the spool and cylinder rod bore were modified to install seals and to permit them to ride across the ports without extrusion.

The bypass tube shown was part of a system to relieve the rod seals of excess pressures by bleeding fluid after its passage through the rod bushing to the low pressure side of the piston. This too provided a leakage path and was found, by experience, to be unnecessary.

Piston and rod gland seals are now T-rings. A porous bronze spring seal excludes dust from the spring chamber. The spool has been lightened and its passages modified to reduce acceleration effects and Bernoulli forces.

The control pressure is supplied from a hydraulic accumulator through a three-way control valve mounted in the driver's compartment and through a reducing valve to a common line which supplies all of the lockout cylinders.

Experience with this lockout system has shown it to be very effective. Hop records taken with the system inoperative record motions of $11\frac{1}{2}$ inches rise and $4\frac{1}{2}$ inches dip at the front of the hull. Total motion with the suspension locked ranges from 2 to 4 inches. The maximum cylinder pressures recorded during firing were 1220 psi in the rear cylinder and 860 psi in front.

As a shock absorber, the system is satisfactory for cross-country operations as flow velocities are great enough to make the orifice porting effective. High speed travel on gently undulating surfaces fails to produce sufficient flow for effective velocity-squared damping. Unfortunately design modifications of the spool, incorporating valve-controlled restraint, have not provided durable components.

The hydraulic means of suspension lockout fulfills the basic requirement of securing the wheels while they conform in position with the terrain. It integrates well with thin-skinned hull structures and torsion spring suspensions, and it is amenable to convenient control in vehicles having compatible hydraulic systems.

The external type of hydraulic lockout system is basically a separate unit which is added to a standard suspension system. In this case, lockout is accomplished by entrapping fluid in a hydraulic cylinder.

10-31.2 HYDROMECHANICAL SUSPENSION LOCKOUTS

The friction hydraulic suspension system shown in Figure 10-41 is a hydromechanical system which incorporates a suspension lockout. Basically, the damping device is a frictional damper (disc brake) to which a hydraulic actuating cylinder is added. The hydraulic-actuating cylinder or "brain box" supplies and controls the pressure to the frictional damper where the energy absorption takes place. The device performs three functions (a) damping, (b) lockout, and (c) bump stop.

Computer studies conducted by the Detroit Arsenal Computer Section have established the following lockout requirements

(a) Each lockout shell to be capable of resisting

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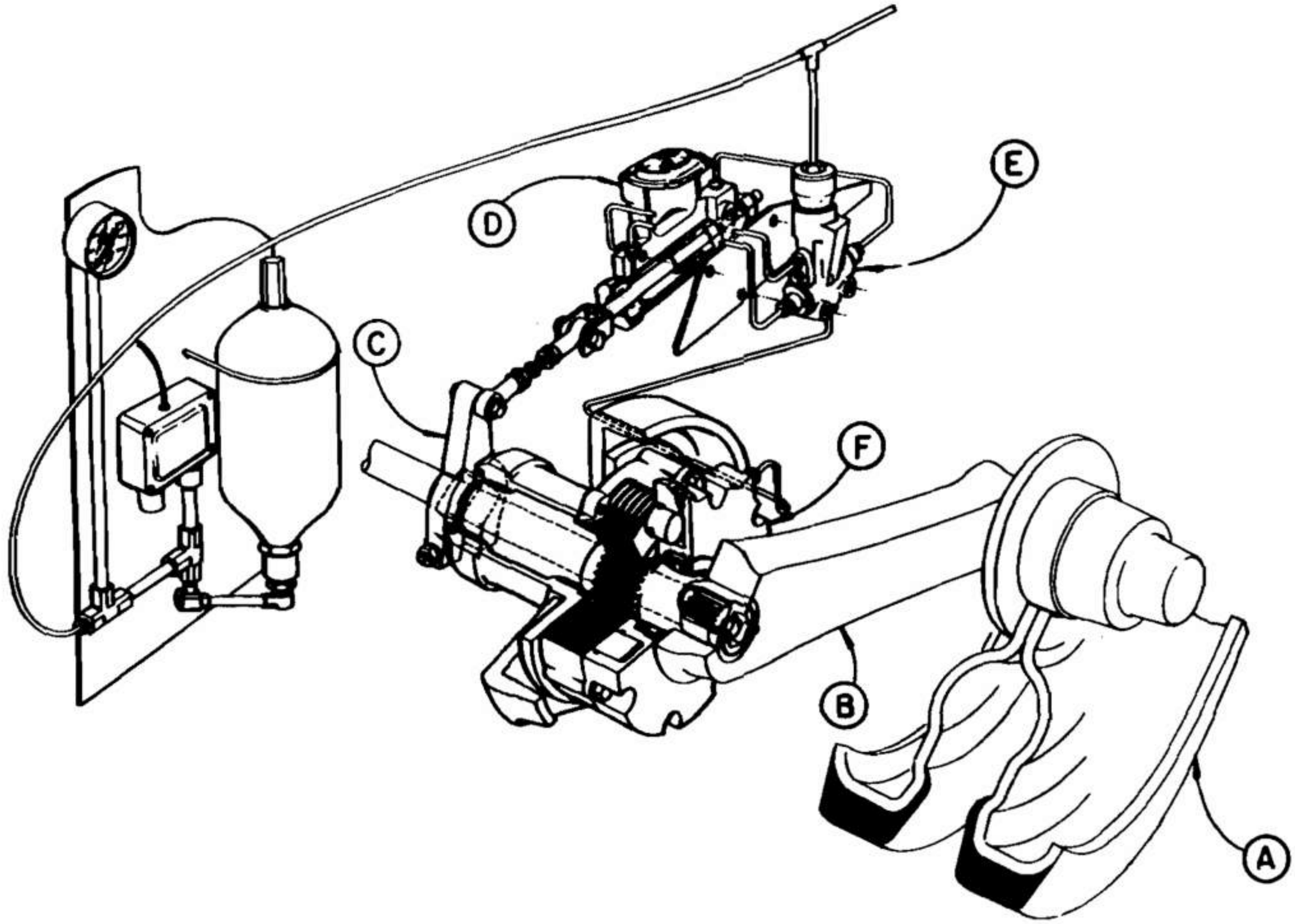

*Figure 10-41. Friction-Hydraulic Suspension System** (Ref. 62)

a force of 260,000 in.-lb torque when applied at the torsion bar trunnion.

(b) Each of the ten (10) torsion bars to be equipped with a lockout.

In Figure 10-41 the component parts of one wheel location are illustratively shown as well as the device for rigididizing the complete vehicle suspension system. This system was applied to the T-117 Universal Carrier for evaluation tests.

The basic operation is as follows

(a) On the assumption that the tank is operating over rough terrain, the road wheel *A* will strike a surface obstruction tending to drive the wheel upwardly relative to the tank, thus moving the road wheel arm *B* in a counterclockwise direction as viewed in Figure 10-41. The road wheel arm *B* carries the arm *C* also in a counterclockwise direction, and thus pulls to the left upon the piston rod of the "brain box" *D*. Movement of the piston rod creates pressure which is transmitted through suitable lines to the flutter valve *E*. The flutter valve is so actuated as to transmit the fluid pressure to the eight (8) pistons in the frictional damper *F*. Pressure on these pistons in the frictional damper forces the stators and rotors together creating a frictional resistance to this motion and retarding the upward movement of the road arm *B*.

(b) If the extreme limit of travel is reached, the piston in the "brain box" enters the lockout area creating maximum pressure which in turn exerts a maximum force on the pistons in the frictional device, thus stopping the upward movement of the road wheel. This compression bump stop feature is accomplished by having a reduced cylin-

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der bore at the end of the compression chamber in the "brain box." A high pressure is developed when the piston is forced into this area, thus producing maximum resistance in the frictional device.

(c) Conversely, if the road wheel *A* encounters a pothole or recess in the surface over which the tank is traveling, it tends to fall downward, moving the arm clockwise, as viewed in Figure 10-4, and driving the piston to the right, thereby pressurizing the opposite side of the "brain box" *D* and the flutter valve *E*. This pressure on the eight (8) pistons in the frictional device forces the stators and rotors together thus retarding the downward movement of the road arm ***B***.

Though the above is an over-simplified explanation of how the damping system functions, it is important in order to fully understand the function of the components.

Under static conditions, the hydraulic damping system is at zero atmospheric pressure but when movement of the piston rod occurs, pressure is created in the sending unit. The amount of pressure is determined by the valving used and the velocity at which the piston rod travels.

As a result of using shock absorber components in the "brain box," velocity control characteristics are approached with the frictional device.

Resistance characteristics are very important in the final solution of the damping problem. In nearly all cases, two-way control is required to properly damp the suspension. The two-way control may be balanced to equal control on both compression and rebound or may be valved in such a way as to have more compression control than rebound control or vice versa, depending on the requirements of the particular system. The "brain box" valving can be adjusted to meet any desired conditions.

A test to determine fade characteristics of the damping system, under continual operation, shows that the friction-hydraulic system is not affected by high temperature as are conventional appurtenances for damping.

The frictional damper is responsive to fluid pressure from the "brain box" while the actual energy absorption is accomplished between the frictional plates. Since the hydraulic system does not actually absorb the energy, but functions as a control device, the system is less dependent upon the viscosity of the oil and therefore on temperature. The resistance actually increases with temperature. The frictional plates using a cerametallic lining have a 0.2 coefficient of friction.

Military vehicles, operating in all types of environments and climatic conditions, dictated the use of MIL-H-5606 oil in the hydraulic system.

Fire power accuracy, we are told, depends on a stationary gun platform. To accomplish this end it is necessary to immobilize the torsion bars, thus cancelling the gun recoil influence. This means sufficient pressure would have to be applied to the stators and rotors of the frictional device to withstand a force of 260,000 in.-lb applied at the torsion bar trunnions. The lockout pressure is applied by the use of an electrically operated hydraulic motor pump assembly including reservoir, relief valve, check valve, pressure switch, accumulator, and control valve.

The lockout requirement of 260,000 in.-lb is accomplished with a system pressure of 1700 psi. Inasmuch as the system has been operated at 2700 psi (equivalent to 460,000 in.-lb) it is felt that the system may be a little over designed.

SECTION XI DRIVING SPROCKET ASSEMBLIES (Ref. 63)

10-32 PITCH INCREASE OF TRACKS RELATIVE TO THE SPROCKET

The life of a well-designed track is determined by increase of pitch in the track relative to the sprocket. When a new track is first fitted, there is a comparatively rapid initial increase of pitch as the pins wear down any irregularities in the pin holes and bed themselves in. After this the increase

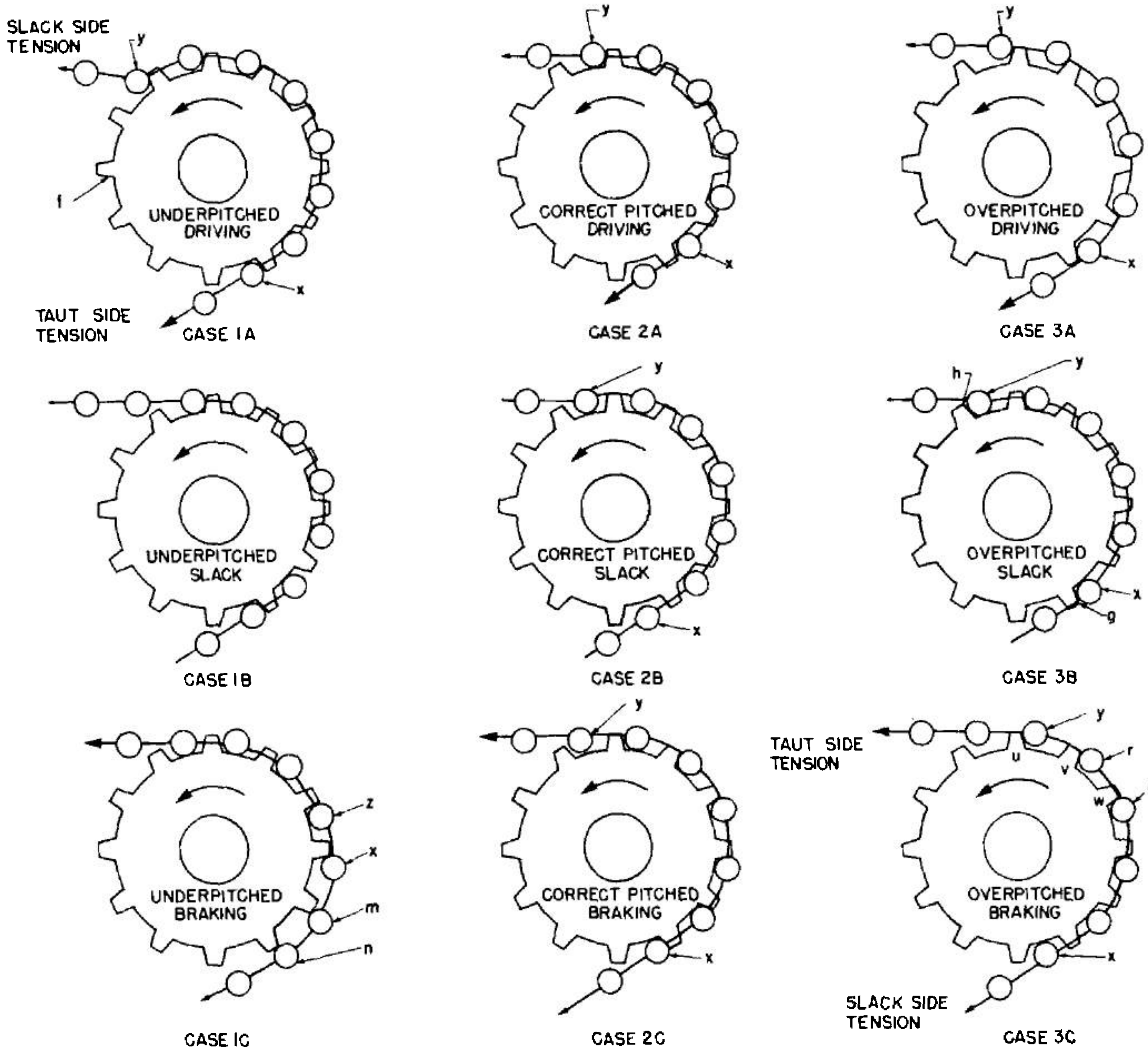


Figure 10-42. Meshing of Tracks and Sprockets

of pitch is comparatively gradual; although near the end of track life it may become more rapid again if the case-hardening of the link or pin wears through to softer metal. As the pitch increases, readjustment becomes necessary to keep the proper value of track tension. When the idler or other adjusting wheel has been moved to the tight limit, it is shifted back again and a link is removed from the track. As far as maintaining the track tension is concerned, this process could be carried on indefinitely; but a limit is set by other factors such as lug wear. In general, however, the lugs can be thickened to prevent this factor from limiting the life of the track. Therefore, the limit of life in tracks occurs when the increase of pitch is such that the links no longer mesh properly with the sprocket teeth and track throwing takes place. Actually, it is found possible to continue using tracks until the pitch increase is of the order of six to eight or even ten percent before they become unserviceable due to track throwing.

Track life may be increased by implementation of two general design methods (a) wear reduction at the track joints, and (b) sprocket design allowing maximum pitch increase before the condition of incompatibility occurs between track and sprocket.

The first method requires careful selection of materials. Considerable material hardness or case-hardening is needed for pins; lug requirements are hardness, for wear resistance (including wear due to dynamic effects of track flexing under tension), and toughness to maintain strength. For lugs, a work hardening material, e.g., manganese steel, is preferred over case-hardening because of wear through. In general, wear can be reduced by increasing road wheel, sprocket, and idler size; and by eliminating the separate idler, or both. Reduction of track flexing will also be helpful in reducing wear.

The second method of increasing track life—proper sprocket design—will be discussed in the paragraphs that follow.

10-33 MESHING OF TRACKS AND SPROCKETS

Various tooth profiles are used on the driving sprockets of tracked vehicles. Since rapid tooth wear is the universal problem of sprocket design, techniques aimed at reducing or minimizing wear should be emphasized in generating tooth profiles. The meshing of tracks with sprockets must be considered under a number of different modes of operation. These modes are shown in Figure 10-42.

Three general cases of track-sprocket relationships can exist, namely: an underpitched, an overpitched, and a correctly pitched track with respect to the sprocket. The pitch of the track for these three cases is smaller, larger, and equal to the pitch of the sprocket, respectively. The pitch relationships change with time as wear increases track pitch and, at the same time, slightly decreases sprocket diameter. In effect, an initially underpitched track will pass through the condition of correct pitch and continue to increase in pitch as a result of wear until it becomes useless due to excessive overpitch.

Consideration must also be given to the conditions of sprocket driving, sprocket idling (or slack track), and sprocket being driven by the track (or braking—due to such actions as steering, deceleration, and descending a slope). These various combined conditions will be discussed in more detail.

Case 1A. Underpitched track, sprocket driving.

A sprocket starting to drive an underpitched track (Figure 10-42(1A)) encounters no difficulty at the beginning of contact since the arriving link x fits easily into a tooth space and is free to gradually slip backward in the tooth space and contact a driving tooth face (such as f) as the sprocket rotates. But at the end of sprocket contact, the leaving link y tends to hang up on the sprocket tooth due to the wedging action of the underpitched condition and the fact that the pin of the leaving link is experiencing a driving force. End of contact is a troublesome portion of the track-sprocket mesh zone for all pitch relationships if tooth profiles are almost radial, as shown in Figure 10-43.

With an underpitched track, high contact load (essentially all the tractive effort) exists between a leaving link and its driving tooth. Therefore, the shape of the sprocket tooth can determine the amount of effort that will be needed to part the track and sprocket. The leaving link y in Figure 10-43 requires considerable force to disengage from its adjacent tooth because of the hindering effect of the tooth profile. The action of parting causes link y to pivot about link z, resulting in interference with the shaded portion of tooth a. This situation has been greatly exaggerated in the figure in order to illustrate this point more clearly.

A minimum design consideration should be to insure that tooth profiles do not interfere with arcs, such as m-n in Figure 10-43, determined by the geometry of track-sprocket disengagement for a particular vehicle. Actually, for a track having correct pitch, it is recommended that sprocket tooth profile be considerably inside the limiting arc in order to permit a rapid parting of track links from sprocket teeth. Furthermore, a tooth profile that makes an angle with the limiting arc (at point of contact) greater than the angle of

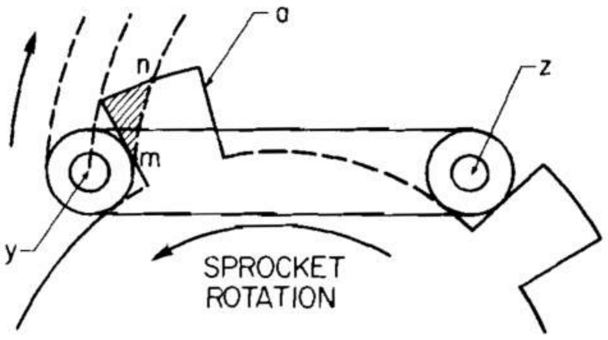


Figure 10-43. Tooth Interference

friction, will disengage without the application of force—even with some track underpitching.

Great accuracy of profile is not necessary; a straight line inclined enough to avoid the interference area of the tooth (shaded area in Figure 10-43) gives the required angle at point of contact. Because of rapid parting, no contact will be made with other portions of the tooth surface. An advantage of the straight line profile is that the angle of inclination at the point of contact—i.e., the angle made between a radial line of the sprocket and the tooth profile—remains relatively constant as the track wears toward the tips of the teeth. In contrast, a curved profile tends to make disengagement difficult when the track is new and increases the possibility of track throwing when the track is old.

Case 1B. Underpitched track, track slack.

The relation of an underpitched track on a sprocket not subjected to tractive effort is self-explanatory and is shown in Figure 10-42(1B).

Case 1C. Underpitched track, sprocket braking.

Conditions here are much more serious than in Case 1A. The arriving link x contacts the rear face of a sprocket tooth at a considerably greater radius than is normal (Figure 10-42(1C)). The only force available to counteract this—i.e., pull the link toward the tooth root—is the tension on the slack side of the track minus the friction component. As any one link (such as z) slides out to a larger contact radius, the amount of underpitching is exaggerated. Each succeeding link tends to slide further toward the tooth tip and aggravates the condition, as is evidenced by z, x, and m. Actually, only a few improperly contacting links are needed to cause loss of contact between sprocket and track (track throwing). Link n (Figure 10-42(1C)) has moved out so far radially that it has already lost contact with the sprocket. Therefore, to prevent track throwing, tracks must be kept very tight and the maximum braking force must be limited to one to two times the tension on the slack side. It is largely to minimize this condition that new tracks are designed to have only a limited underpitch to extend their useful life. Generally, the amount of underpitching used is such as will result in correctly pitched track after the comparatively rapid initial pitch increase due to bedding down of the pins has taken place.

Cases 2A, B, and C. Correctly pitched track; conditions of sprocket driving, track slack, and sprocket braking.

The track-sprocket relations of these three cases are shown in Figure 10-42(2A), (2B), and (2C), respectively, where x represents the arriving link and y represents the leaving link. The only comment necessary here is that, in the slack condition (2B), the track can remain in a driving or braking position with respect to the sprocket, rather than assume the theoretical, neutral position of Figure 10-42(2B).

Case 3A. Overpitched track, sprocket driving.

This track-sprocket relation is shown in Figure 10-42(3A). Tractive effort and inclination of tooth faces cause links to move out radially on the sprocket and, unless some constraint is offered, the links will ride over the tooth tips, resulting in a thrown track. Constraint for a given link is the tension exerted by the link which arrived ahead of it at the sprocket. This tension is aided by the friction force when the sprocket is driving.

If friction is neglected, the ratio between the taut side and slack side track tensions (T_2/T_1) for a given link can be shown to be a function of two angles (Figure 10-44). Angle φ depends on the slope of the tooth surface to a radial line such as x-x, while angle Θ depends on the track pitch and sprocket radius at contact. Figure 10-44 shows schematically the contact of a track link with a sprocket tooth. The tooth surface slopes in such a

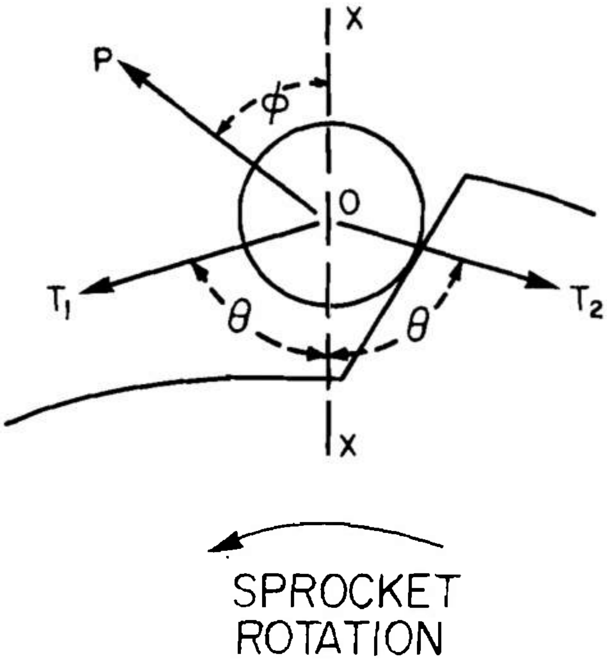


Figure 10-44. Tensions in Link

way that force P exerted on the link makes an angle φ with the radial line, x-x. Also, track tensions T_2 and T_1, imposed on the link on the taut and slack sides, respectively, make angles Θ with the radial line. Therefore, the forces can be resolved radially and tangentially.

Radial forces give

$$P \cos \varphi = (T_1 + T_2) \cos \Theta \qquad (10\text{-}162)$$

Tangential forces give

$$P \sin \varphi + T_1 \sin \Theta = T_2 \sin \Theta \qquad (10\text{-}163)$$

Rearrangement of these equations shows

$$P = (T_2 + T_1) \frac{\cos \Theta}{\cos \varphi} = (T_2 - T_1) \frac{\sin \Theta}{\sin \varphi}$$

$$(T_2 + T_1) \tan \varphi = (T_2 - T_1) \tan \Theta$$

Further algebraic manipulation yields the desired taut side to slack side tension ratio.

$$\frac{T_2}{T_1} = \frac{\tan \Theta + \tan \varphi}{\tan \Theta - \tan \varphi} = \frac{\tan \varphi}{\tan \Theta} - \frac{\tan \Theta}{\tan \varphi} \qquad (10\text{-}164)$$

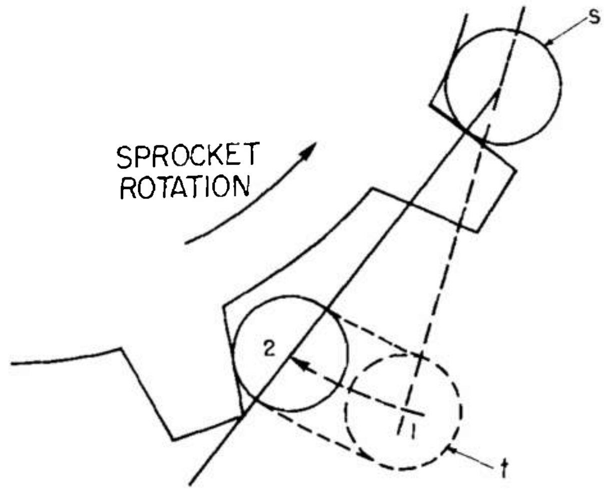


Figure 10-45. Overpitched Link

Ratio T_2/T_1 gives the value of tension an arriving link can take (compared to the tension on the link directly ahead of it) before the arriving link will tend to ride radially outward on its contacting tooth.

Friction may allow a given link s to remain firmly seated in its position at a tooth root (Figure 10-45). However, as the next link t arrives, it wants to share in taking tractive effort. Since the track is overpitched, initial contact with a tooth will be at some point radially out from the root as the link swings in toward the sprocket from position 1 to 2. Furthermore, link t will remain in position 2, due to friction and the fact that link s is carrying part of the tractive effort. If link t were to slide toward the root, link s would be displaced from its contacting tooth with the result that link t would take the entire tractive effort. Such an action would cause link t to slide radially outward and reposition itself.

If the links position themselves radially outward from the roots of the teeth, the result is an increase of sprocket diameter or an increase in effective pitch. Since this action reduces overpitching, an arriving link will not contact a tooth much farther toward the tip of the profile than did the previous link. As transmission of the tractive effort continues, the links quickly attain a position where the original overpitch is offset by an increased effective sprocket diameter (Figure 10-42(3A)). The illustration of Case 3A shows an extreme condition with a high tractive effort and a very low tension on the slack side. In general, links tend to slide back toward the roots of the teeth as they progress around the sprocket from x to y. This has been verified by photographic observations (Ref. 63). However, in either the general or extreme condition, the tractive effort will be distributed over a number of teeth and links, but not equally distributed.

Case 3B. Overpitched track, track slack.

This track-sprocket relationship in which the track meshes easily with the sprocket is shown in Figure 10-42(3B). The arriving link x rests against the leading face of its driving tooth g while each link ahead of it is pushed progressively farther from the leading face of its corresponding driving tooth. Depending upon the amount of overpitch and the diameter of the sprocket, this advancing of the links relative to the sprocket teeth can continue to the point where a link contacts the trailing face of the tooth ahead of it, as is shown at h, with respect to the leaving link y.

Case 3C. Overpitched track, sprocket braking.

This case is similar to an overpitched track driving (Case 3A), but the arriving link x in Figure 10-42(3C) is positioned into the bottom of a tooth space with friction tending to keep it there. As a result, the entire braking effort can be taken by tooth u, adjacent to leaving link y. However, when the braking effort is many times greater than the tension on the slack side of the sprocket and the sprocket teeth have very tapered profiles, several links—such as y, r, and s—in the vicinity of the leaving link y may overcome friction and slide toward the tips of the teeth. Braking effort will then be distributed to the several teeth (u, v, w) that are in contact with links, as is shown in Figure 10-42(3C).

10-34 SPROCKET TOOTH DESIGN

The effect of torque reversals must be considered when relating sprocket tooth spacing to the geometry of track engagement surfaces. If the space between teeth is considerably longer circumferentially than track boss (or trunnion) thickness, significant play between the meshing parts results. When torque is reversed, a trunnion loses contact with one tooth face and impacts on the back of the next. A correctly pitched track-sprocket combination represents the worst condition from the standpoint of impact due to torque reversal, since each link within the mesh zone can travel the full distance of circumferential looseness toward impact. In the underpitched and overpitched cases, impact contact takes place in less than the full dimension of looseness which results in less severe shock. Some clearance is necessary from the standpoint of assembly as well in order to prevent an overpitched track from contacting the backs of teeth directly ahead near the slack side of the sprocket.

Tooth profile design is also affected by torque reversals; consequently, both the front and backs of teeth must counteract loads. Although most vehicles under consideration will be traveling predominantly in a forward direction resulting in wear predominantly on one side of the tooth, no direct application of this principle has been made in designing sprocket tooth shapes. However, sprocket rings are sometimes reversed after a specified mileage in order to make the back of a tooth the front.

Sprocket teeth must have sufficient strength to withstand the tractive effort and possess an adequate reserve for shock loading. Due consideration should also be given to the fact that wear will reduce the tooth thickness, resulting in an overpitched condition and a tooth loading nearer the tip.

The preceding considerations apply to simple toothed sprockets meshing with an approximately cylindrical surface (or boss) on a link. Substantially different conditions exist in sprockets having rollers that engage teeth on links—here tracks with considerable underpitching can be successfully used.

Four basic types of sprockets have been developed for tracked vehicles. These are (a) the dual metallic sprocket, with and without support tires, (b) the single metallic sprocket, center drive, with support tires, (c) the roller tooth sprocket, center drive, and (d) the nonmetallic sprocket rubber or rubber-covered.

TABLE 10-3
TRACK APPLICATION CHART

Track Model	Description	Application	Track Width, in.	Track Pitch, in.	Shoe Weight, lb	Pin Size, in.
T72E1	Cast steel, single-pin	Lt. TK. M24	16	5-1/2	26	0.828
T80	Steel and rubber, double-pin	M. TK. M4 Series, with Hor. Sus.	23	6	56.5	1.25
T80E1	Steel and rubber, double-pin	M. TK. M26 and 46	23	6	55	1.25
T80E5	Steel and rubber, double-pin	M. TK. M4 Series, with Hor. Sus.	23	6	59	1.25
T80E6	Steel and rubber, double-pin, 1-1/2 inch grouser	M. TK. M47, M46, and T42	23	6	57.5	1.25
T80E7	Steel and rubber, double-pin, 1-1/2 inch grouser and 2-1/2 inch end connector	H. TK. T43	28	6	60	1.25
T84	Rubber chevron, double-pin	M. TK. M4 Series, with Hor. Sus.	23	6	51.7	1.25
T84E1	Rubber chevron, double-pin	M. TK. M47, M26, and T42	23	6	50.6	1.25
T85E1	Rubber chevron, double-pin	Lt. TK. M24	14	5-1/2	24	1.00
T91E3	Cast steel and rubber, single-pin, detachable rubber pad	Lt. TK. T41E1	21	6	40	0.875
T95	Forged steel and rubber, single-pin, detachable rubber pad	M. TK. T42	24	6	48.44	0.875
T96	Steel and rubber, double-pin	H. TK. T43 and M. TK. M48	28	6-15/16	67.5	1.25
T97	Rubber chevron, double-pin	H. TK. T43 and M. TK. M48	28	6-15/16	59.4	1.25
T96E1	Steel and rubber, double-pin	H. TK. T43 and M. TK. M48	28	6-15/16	67.5	1.25
T97E1	Rubber chevron, double-pin	H. TK. T43 and M. TK. M48	28	6-15/16	59.47	1.25

SECTION XII TRACK ASSEMBLIES AND COMPONENTS

10-35 INTRODUCTION (Refs. 64, 65)

The primary function of the tracks is to transmit tractive effort to the ground while providing a continuous road-wheel surface for traversing various types of terrain. This function has been performed best with a strong, flexible, and preferably light arrangement of continuous linked blocks or shoes (continuous bands of rubber-covered steel cables have been used successfully on half-track vehicles, though not on tanks). The continuous linked-shoe track must contain durable shoes with strong flexible joints, provide means for engaging the sprocket and retaining the track on the suspension, and offer a maximum permissible degree of traction and flotation.

Early tanks employed tracks consisting of large, flat metal plates hinged together with simple pins. Subsequent tank development resulted in higher vehicular speeds and consequent need for improved tracks. Prior to World War II, only minor track

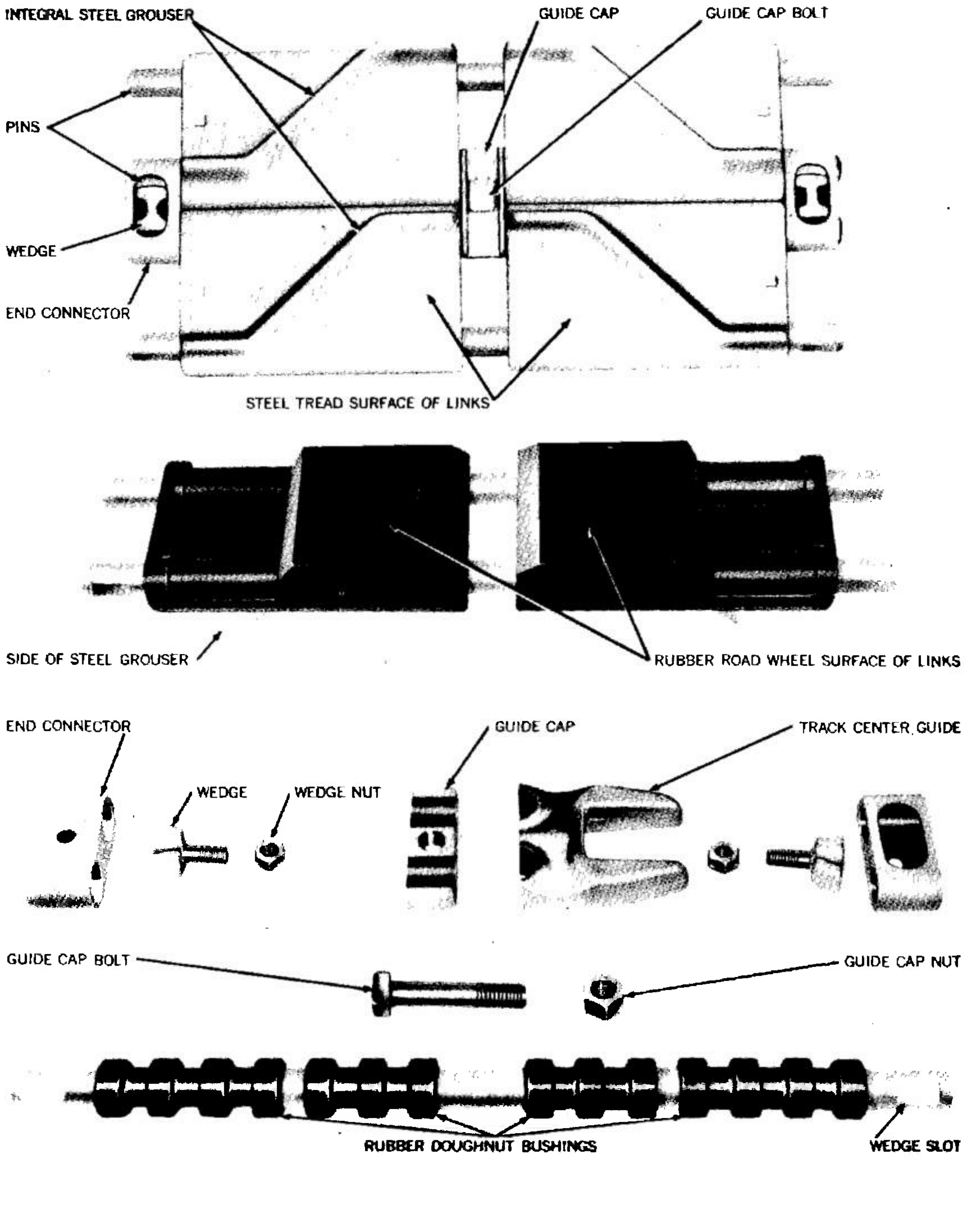


Figure 10-46. Double-Pin Shoe Assembly (T96 Track)

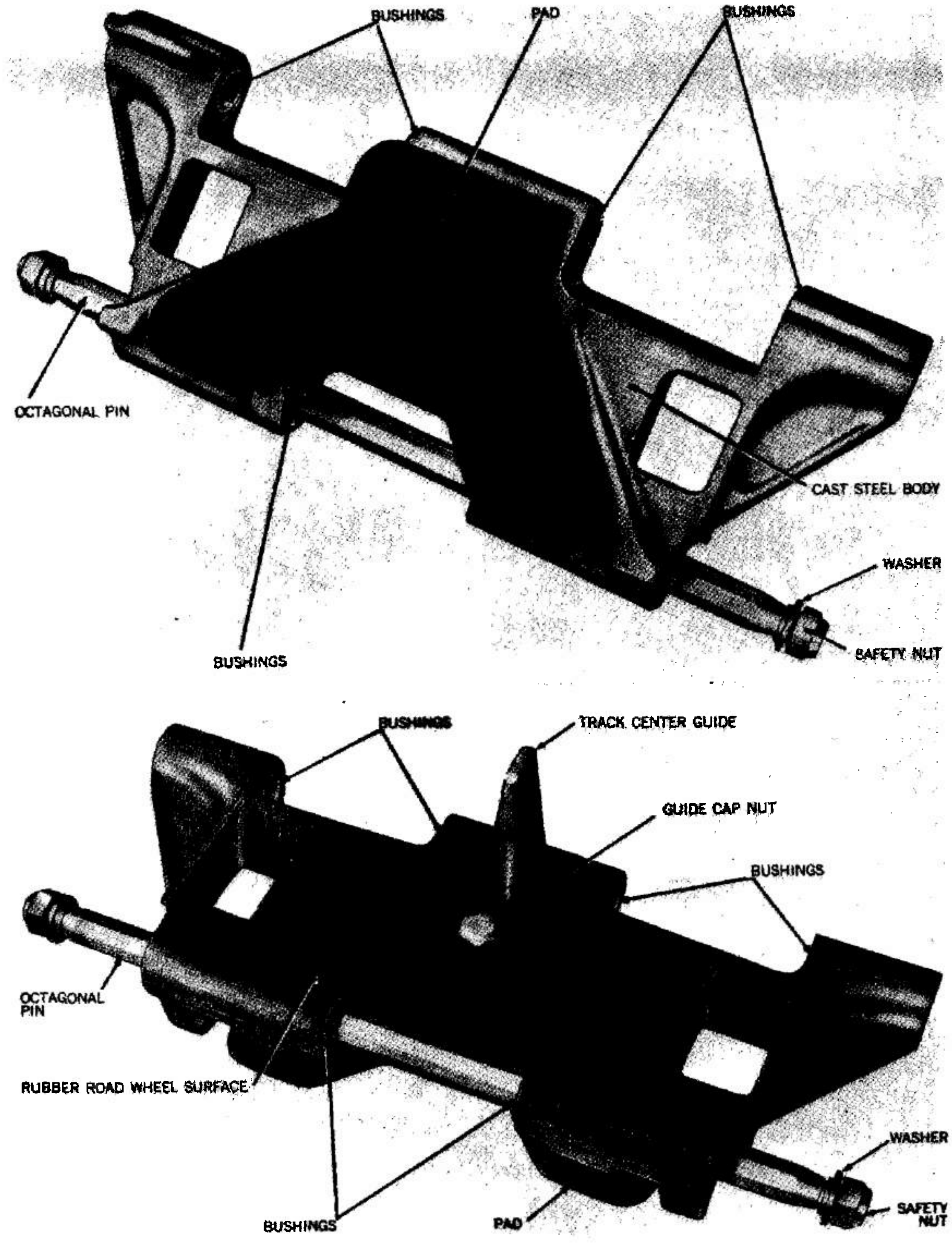


Figure 10-47. Single-Pin Shoe Assembly (T91E3 Track)

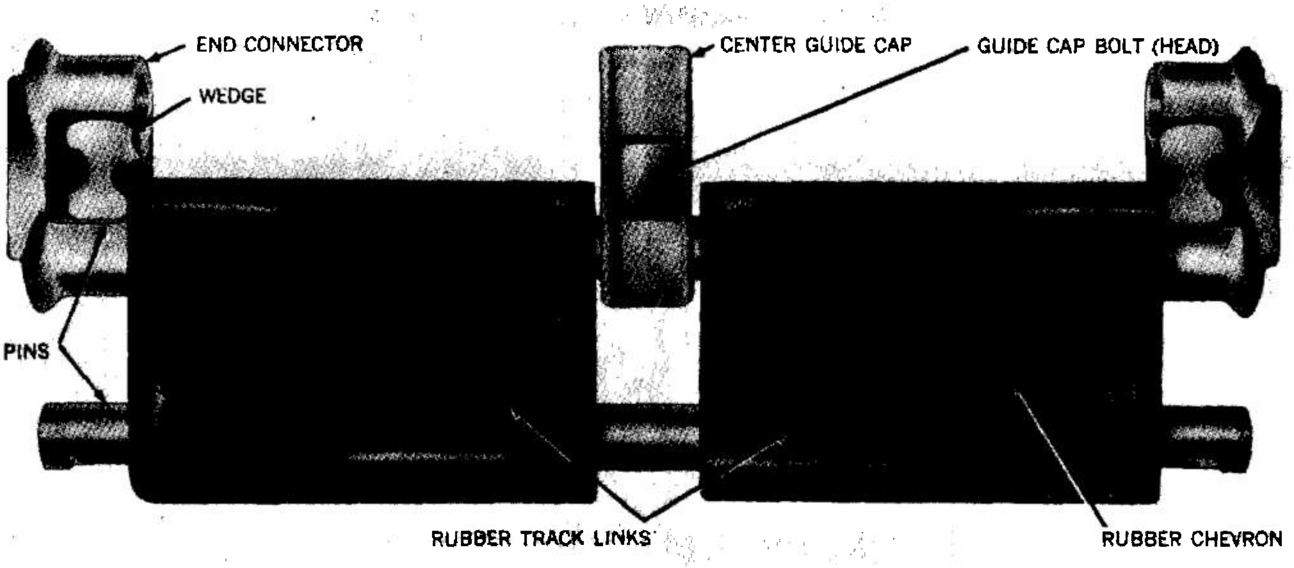


Figure 10-48. Grouser on Road Surface Side (T84E1 Track)

development was pursued. However, during World War II, there was a vast amount of activity in this field. The initial phase of the accelerated wartime track program placed emphasis on steel tracks as a result of the critical supply of natural rubber, the lack of any suitable synthetics, and durability of steel tracks in active theaters.

Two basic double-pin, rubber-bushed steel tracks evolved. One type incorporated a binocular (double-pin type) frame as the tension link, with various types of metal surfaces on the tread and road wheel sides. The other type employed nonbinocular steel shoes, and was designed in several ways in an effort to provide structural strength and wearing properties. None of the nonbinoculars designs, however, was very successful.

Concurrently, the major rubber companies experimented with numerous synthetic rubber compounds to duplicate the characteristics of natural rubber required in tank track use. Generally, synthetic tracks proved inferior to natural rubber tracks in resistance to cutting, cracking, and shelling-out, though superior in abrasion resistance. At the end of World War II, average endurance life of synthetic rubber tracks was approaching that of natural rubber tracks. Recently, polyurethane compounds have been tested as track pads. The polyurethane pads showed greater wear than the standard SBR compound pads but the polyurethane was superior with respect to resistance to chunking, chipping, ozone cracking, and bonding failures.

Parallel efforts to obtain stronger, more durable, noncritical materials was the search for improved design methods to solve such problems as weight reduction, track jumping, and increased traction and flotation. Some problems were partially solved at the expense of other considerations, depending on the immediate criteria of basic design. Actually, durability was the overall yardstick for all track design and remains such today.

10-36 BASIC TYPES OF TRACKS

Currently, two basic track types are used on American tanks. These are the *single-pin type* track and the *double-pin type* track. With the former, a single hinge pin is used with each basic link, while with the latter each basic link has two pins which are in turn connected to the adjacent links. Figures 10-46 through 10-51, show detailed component features of four of these tracks. Table 10-3 is a partial list of some modern track types, their application, and pertinent specifications. A discussion of the components of the basic types of tracks is presented in the immediately following paragraphs while other types of tracks are discussed in later paragraphs of this section. A more

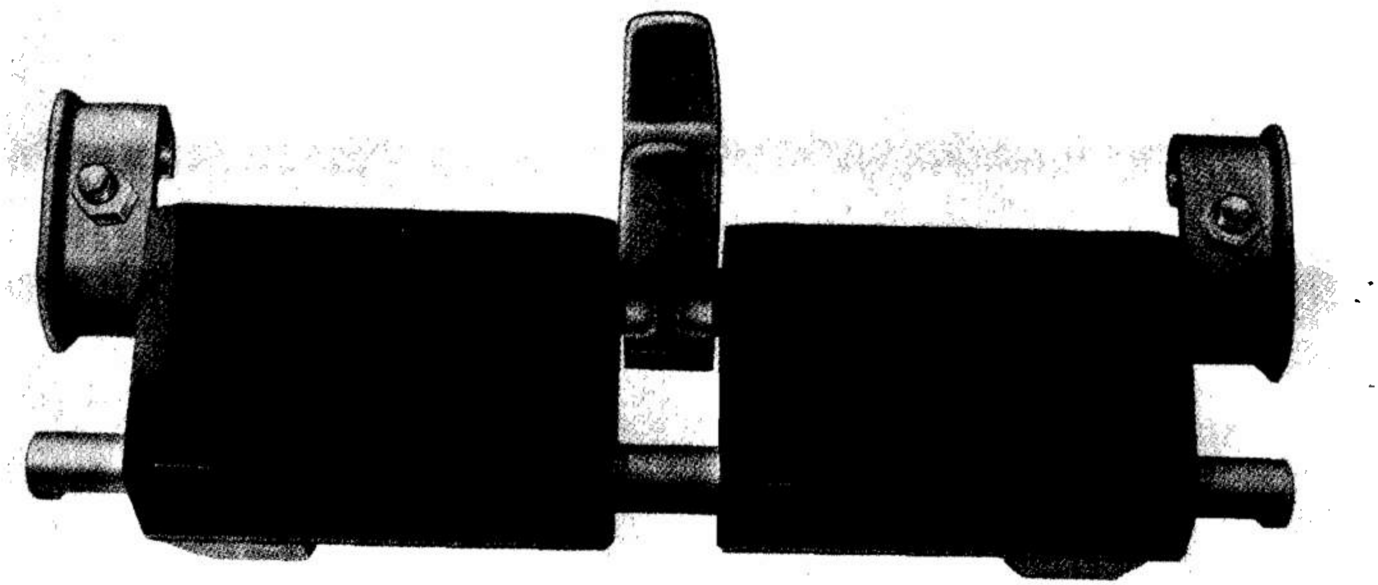


Figure 10-49. Road Wheel Side (T84E1 Track)

detailed tabulation of standard track types and their design characteristics is given in Appendix II.

Blocks, as used in double-pin tracks, compose the basic track body. When two blocks are assembled on the bushings and track pins, as shown in Figure 10-46, the unit is known as a link. The link, end connectors, wedges, and capped center guide are called the shoe assembly. Blocks may consist of rubber on both sides (Figures 10-48 and 10-49), of a steel face and rubber back (Figure 10-46) or of an all-steel construction. A grouser (chevron) is a raised segment on the tread surface of the block, which affords increased traction over a smooth tread surface. In the T91E3 track the detachable pad functions as a grouser. All current tracks are rubber-bushed with 0.187-inch-thick, doughnut-type bushings.

In the T91E3 single-pin track the basic metal-rubber structure is called the body. When bushings are added, the assembly is known as a link and is comparable to the link of the double-pin rack. The link, pin, nuts and washers (Figure 10-47) make up the shoe assembly.

Center guides are either bolted to or integral with the top of the body (T91E3 track), or attached by caps to the pins (T96 and T84E1 tracks). In the latter case the capped guide not only functions as a track retaining component but as an additional connecting unit. Outer guides are sometimes incorporated at the extremities of the links, and are made integral with end connectors of double-pin tracks and integral with the body or link of single-pin tracks.

Binocular frames are metal tubular units with webbed ends, and are designed to contain the bushings and pins of double-pin tracks. The binocular frame provides support for all-rubber blocks or fabricated rubber-steel combinations. In Figure 10-51 the rubber road wheel surface is removed to show binocular construction.

10-37 OTHER TYPES OF TRACKS

10-37.1 FLEXIBLE PIN-JOINTED TYPE

Most foreign tank tracks, at least through the World War II period, were of the flexible pin-jointed type. This track consists of a chain formed of a series of rigid links connected by hinged joints. These hinges are formed by passing a pin through interlocking links, or through adjacent rigid links. The links may be identical or may be of two different types, alternating around the track. This

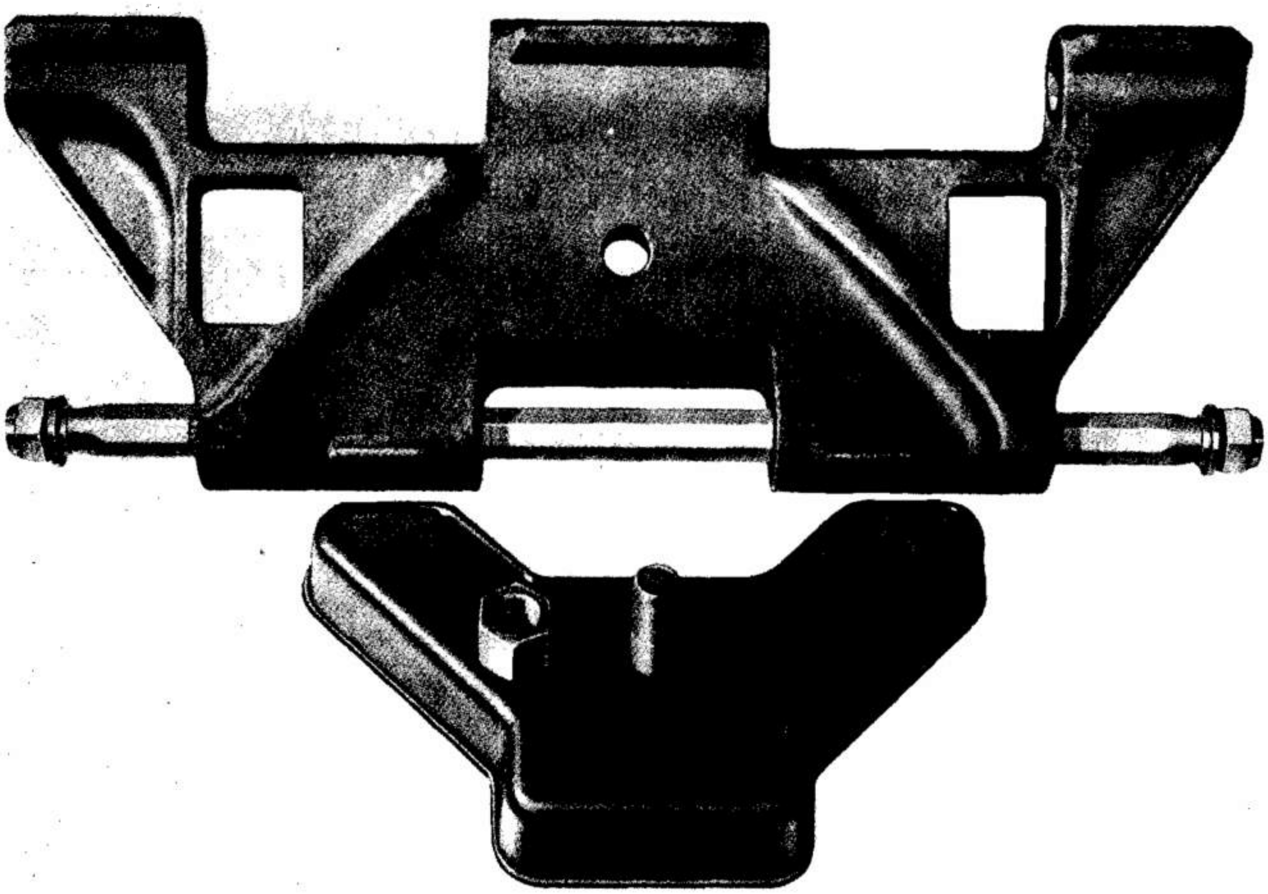


Figure 10-50. Grouser Side of T91E3 Track With Pad Detached

latter construction is referred to as a "two-piece link".

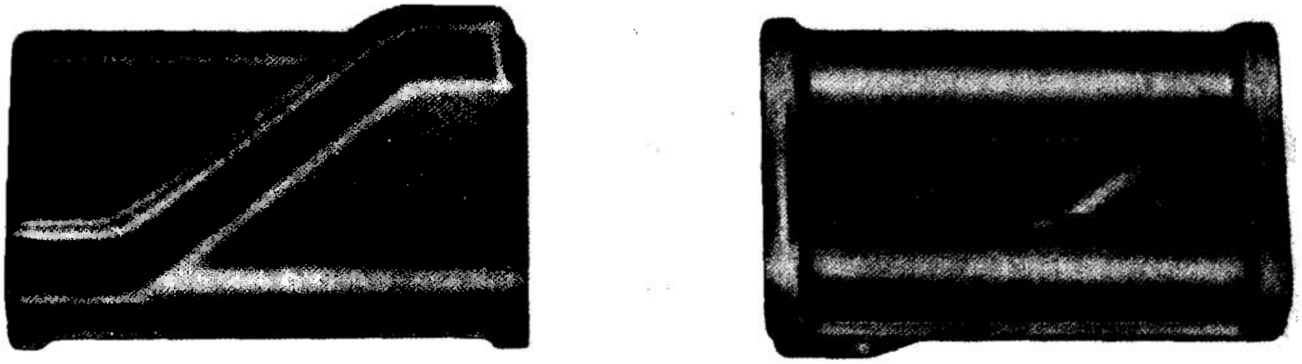
Characteristics of pin-jointed tracks are strongly dependent upon track pitch (distance between corresponding points of neighboring joints). Pitches of different tracks vary from about $1\frac{3}{4}$ in. to 10 in. These tracks bend into a polygon, rather than into a smooth curve, when deformed.

Flexible pin-jointed tracks may be subdivided into "dry pin" and "lubricated" types of track. In most modern tracks, the pin is without lubrication and friction arises as the metal surfaces move with respect to each other. Track power losses are high and this type of track wears out rapidly. In the past, designs involving lubricated and sealed pins have been produced; however, such systems are not in general use today because of the good service obtained from the rubber-bushed track.

10-37.2 IRREVERSIBLE TYPES (ELASTIC GIRDER AND RIGID GIRDER)

All of the track types discussed previously were free to bend in either direction. In the rigid girder track, the links are so interlocked as to permit the track to form a convex, but not a concave, curve. The ground pressure under a short-pitched pin-jointed track, or a continuously flexible track, is greatest directly under each road wheel because the track is relatively free to bow upward between them. A rigid girder track is designed to prevent such upward bowing, and the portion of the track between the road wheels remains in contact with the ground.

The rigid girder track cannot be used with a sprung suspension system, which permits road wheels to displace vertically, since this would place an overload upon the unsupported girder when



GROUSER SIDE

BINOCULAR CONSTRUCTION

Figure 10-51. Binocular Construction (T80E6 Track)

passing over hard obstacles. Its application has been restricted to very slow-moving vehicles such as life-boat carriers and log trailers. It has been suggested that a rigid-girder tracked vehicle be built with the road wheels all mounted on a single rigid unit with suspension springing between this unit and the vehicle hull.

The elastic girder track is similar in principle to the rigid girder track; however, in this case, the joints incorporate rubber blocks which tend to make the track assume a slightly convex curvature. Normal track loads cause this track catenary to flatten out so that ground pressure is more evenly distributed. A slight amount of flexibility is present which permits the track to adjust to rough, hard terrain.

10-37.3 CONTINUOUSLY FLEXIBLE OR ENDLESS BAND TYPE

This type of track consists of a continuous band which is flexible at any point along its length rather than a series of rigid links that are flexible only at their pin-connected joints. The band is usually formed of fabric or steel-reinforced rubber. The use of the band track has not been as widespread as the block and pin type; therefore, the development of this type is not as advanced. However, the recent emphasis for greater speed and air transportability of military vehicles makes the use of the band track more attractive.

There are three basic types of band tracks (a) flexible friction drive track, (b) rubber band track, and (c) band-block track. These are discussed in the paragraphs which follow.

10-37.3.1 Flexible Friction Drive Track

The distinguishing characteristic of this track is that stiff lateral members are not employed. This track, like most band tracks, is usually made up of steel and fabric-reinforced rubber. Since lateral stiffeners are not employed, the drive force is transmitted by means of V-groove-type drive wheels which engage raised V-belt-like rails on the track. This light flexible type of track has many desirable characteristics for achieving high flotation for light vehicles and cargo trailers. It is not used on combat-type tracked vehicles because of its strength limitations.

10-37.3.2 Rubber Band Track

This term applies to the type of reinforced rubber track used in the American half-track vehicles of the 1930's and World War II. Lateral stiffness is obtained in this track by means of rigid cross members vulcanized into the rubber. The driving force is transmitted by the sprocket through these rigid cross members. In order to restrict the twisting of the track along its longitudinal axis, the track guide members are interlocked (see Figure 10-52).

10-37.3.3 Band-Block Track

The band-block track has either rubber blocks or metal track bars bolted or riveted to the flexible,

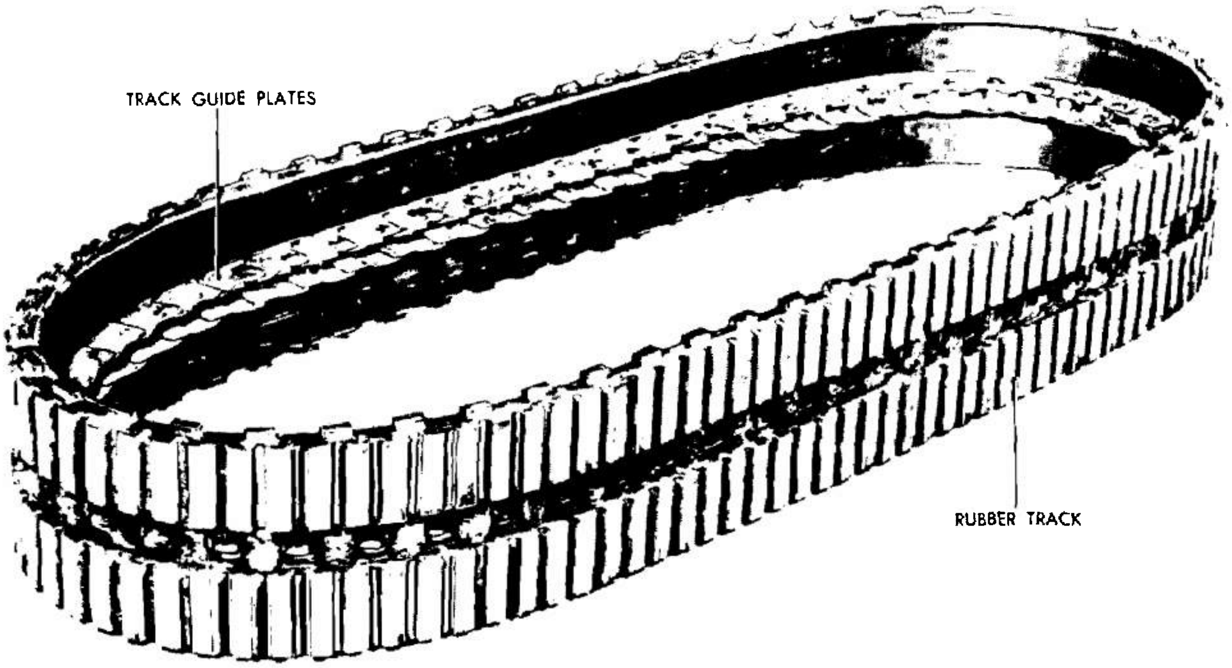


Figure 10-52. Rubber Band Track

reinforced rubber bands. It is this type that has undergone the most development and is now referred to simply as the band track. Several of the more pertinent design details and functional advantages of this track are discussed in the paragraphs below.

10-37.3.4 Vehicle Speed

The block and pin track induces substantial vibrations into the suspension system and vehicle hull when traveling at high speed. This is because the track bends into a polygon rather than into a smooth curve. The slapping of the track blocks as they strike the terrain produces high ground pressures and shortens the life of the track block. The polygon effect can be minimized by reducing the track link pitch but this results in increased track weight and compounds the problems associated with track pins. The continuously flexible band track with narrow track bars approaches the desired smooth bending curve with a reasonable track weight. It is the light weight that is considered by some to be the chief advantage of the band track. The low weight permits the use of very wide tracks to achieve the low ground pressures necessary for the high flotation vehicles.

Data obtained by towing 18- to 25-ton tracked vehicles, with their drive shafts removed, indicate that rolling resistance of band-tracked vehicles is comparable to other types up to 15 mph. At 15 to 30 mph, the rolling resistance of the band track is substantially lower than that of block and pin tracks. The reduced rolling resistance combined with the superior vibration characteristics of the band track are the factors that permit higher vehicle speeds.

10-37.3.5 Sectional Construction

One of the chief disadvantages to the early band tracks was the endless belt type of construction. Damage to a small section made it necessary to discard the entire band track. Since it was impractical to carry an entire spare track on the vehicle, the vehicle was completely immobilized. Techniques have been developed, however, for the sectional construction of band tracks to eliminate this disadvantage.

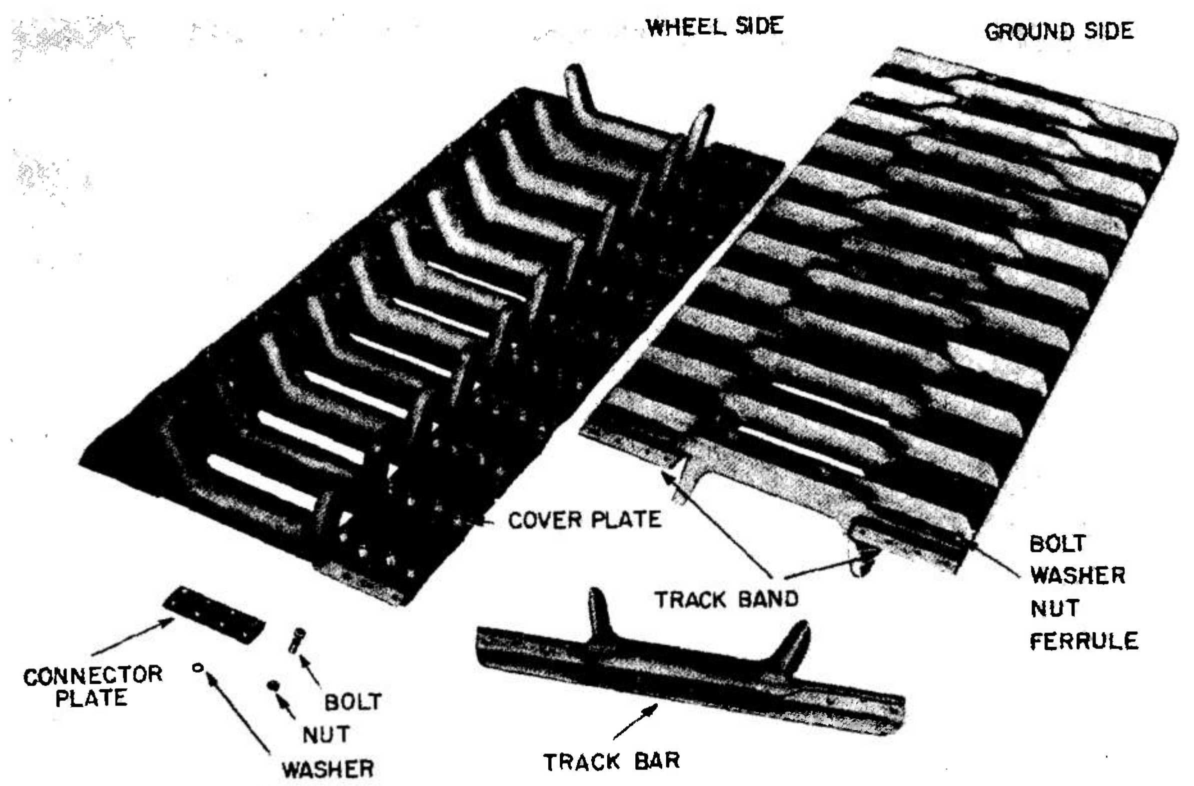


Figure 10-53. Assembly of Band Track Section

10-37.3.6 Band Track Design

Figure 10-53 shows a typical band track assembly for a vehicle with a single-road-wheel type of suspension. Band track construction for a dual-road-wheel suspension is quite similar. The parallel cable reinforced bands, referring to the nomenclature on Figure 10-53, are clamped between the cover plates and track bars by means of through bolts and self-locking nuts. The road wheels run on the track bars between the guides. The drive sprocket engages the center portion of the track bar. Track alignment is provided by means of steel track guides welded to the steel track bars. These guides straddle the road-wheel tires, idler wheel, and sprocket. Each track section is made to a convenient length for ease of handling and is connected to adjacent sections by connector plates. The joint is made between the track bars. Some designs provide for the joint to be made at the track bar.

Rubber pads are bonded into the V-section of the stamped steel track bars. These inserts greatly reduce wear of the track bars and increase traction on hard surfaces. The rubber pads protrude one-eighth of an inch beyond the metal edge of the track bar. The pad continues to protrude during wear of the metal track bar. Although the track bars shown are steel stampings; aluminum and magnesium castings, forgings, and stampings have been successfully used with the band track on various types of vehicles.

Since it is desirable to have all the track sections

TABLE 10-4
PRODUCTION AND EXPERIMENTAL TRACK BANDS

Part No.	Vehicle	Weight, tons	Nominal Length, in./Section	Width, in.	Thickness, in.	No. of Cables	Cable Type	Cable Size, in.	Track Band Weight, lb/Section	Swaged Cable Strength,* lb	Band Design Strength, lb	Mfg. Min. Specified Strength, lb
PRODUCTION VEHICLES												
8740998-9	M56	8	44.0	5.370	.430	14	7x7x7	5/32	4.86	2,300	32,200	25,535
8727043	M50	9	59.90	5.375	.430	11	7x19	5/32	9.63	2,800	30,800	29,100
7976864	M76	6	44.0	3.5	.360	7	7x7x7	5/32	3.05	2,300	16,100	15,530
7977329	M76	6	44.0	7.0	.360	14	7x7x7	5/32	6.04	2,300	32,200	29,950
PILOT VEHICLES												
37858	T116	4	30.50	4.0	.468	8	7x19	3/16	3.5	3,775	26,864	25,790
T-700AS	T107	3	58.37	3.5	.375	8	7x19	3/16	5.1	3,775	26,864	25,790
EXPERIMENTAL BANDS — GM DYNAMOMETER TEST VEHICLES												
5-18919	TDTV	25	39.62	7.00	.770	8	7x7x7	11/32	14.29	10,000	80,000	76,800
5-19141	TDTV	20	38.36	6.40	.590	11	7x7x7	1/4	9.52	5,150	56,650	53,530
5-19142	TDTV	18	38.36	6.40	.590	11	7x19	1/4	10.23	6,200	68,200	64,450
5-19391	TDTV	18	35.88	5.14	.710	7	7x19	5/16	9.70	9,700	67,900	65,520
5-19392	TDTV	18	35.88	5.14	.710	7	7x19	5/16	9.69	9,700	67,900	65,520

*Based on swaged length of 3 diameters.

of the same length, a design compromise between track bar spacing and total track length is necessary. The width and number of steel-reinforcing cables are a function of the vehicle size. Table 10-4 lists some typical track bands currently in production for vehicles up to 25 tons.

Two of the more important design parameters and techniques that should be considered by the vehicle designer are

(a) Track-bar spacing relative to road-wheel spacing so as not to induce severe disturbances in the suspension by simultaneous climbing and dropping of all the road wheels on the track bars.
(b) Drive sprocket tooth pitch to facilitate track bar engagement and disengagement and thereby preclude reverse bending of the cable which reduces the fatigue life.

10-37.3.7 Spaced-Link Track

The spaced-link track applies the principles of soil mechanics to develop the maximum shear strength of the supporting soil while, at the same time, producing a minimum of resistance to motion. This track concept and the principles upon which its design is based are discussed in Chapter 6, paragraph 6-1.

10-38 BASIC DESIGN AND PERFORMANCE CONSIDERATIONS

Current track design is based on the following primary design requirements and preferred performance characteristics

(a) Low ratio of track weight to vehicle weight.
(b) Low power loss in track (also sprocket and suspension).
(c) High static and dynamic flotation.
(d) High traction values (under such broad considerations as improved road; secondary road; general cross-country including sand, gravel, shale, rocky terrain, mud, and clay; also ice and snow).
(e) Durability.
(f) Resistance to track throwing.
(g) Minimum source of secondary vibration.
(h) Ease of maintenance.
(i) High-speed performance.

The track designer must also consider such secondary requirements as the use of low priority metal alloys; the use of synthetic rubbers; the use of available low priority, high-production manufacturing facilities and techniques; and minimum material, equipment, and man-hours for manufacture.

10-38.1 TRACK WEIGHT

Excessive track weight affects power-to-weight ratio of engine and tank, resulting in decreased acceleration, increased fuel consumption, unsatisfactory riding characteristics, shorter suspension life, and increased maintenance and logistics.

Weight reduction was the principal goal of many of the experimental track programs conducted during World War II. Skeletonizing of grouser plates in many double-pin steel tracks was carried to the point of bolting or welding a steel grouser directly to the binocular frame, with only slight flaring pads added to the bare chevron to distribute the load. In some cast steel, nonbinocular, double-pin tracks, the web between pin tubes, which formed the bogie surface, was reduced to a thickness that proved inadequate. In single-pin tracks, structural strength was obtained by webs between the hinge lugs, and the sections between webs were held to a minimum consistent with adequate rigidity and load-carrying ability. This practice led to the coring of lightening holes through the track shoe but these generally had a detrimental effect on bogie tire life.

Soundness of castings, forgings, and welds, as well as quality of the material, became important factors in these minimum-weight designs. All too frequently, well-designed shoes lacked sufficient durability as a result of poor manufacturing or heat-treating techniques, or the use of low-grade steel. Here, as in nearly all phases of design practice, a compromise was dictated—in this case between desirable high-quality properties and the undesirable high cost of high-grade steels for track shoes.

Application of rubber, both on the wheel and tread surfaces of track shoes, somewhat reduced the weight of tracks. Since this weight reduction is not relatively large in double-pin types, the metal parts are revised constantly in experimental units to further reduce track weight. The metal construction of present single-pin rubber tracks constitute the bulk of the shoe weight, making reduction of weight an even more difficult problem. Relative weights of current steel, rubber, and steel-rubber combinations are listed in Table 10-3.

10-38.2 TRACK POWER LOSSES

A vast amount of test data recorded prior to 1946 indicated that high power losses relative to gross engine horsepower occurred in the tracks and suspension system of tracked vehicles. However, the resistance-to-towing method (a tank equipped with test tracks is towed by a dynamometer) used in most of the tests was inadequate for a quantitative evaluation of the various factors contributing to track and suspension propulsion losses. Not until the development of the road-load method of determining propulsion losses was a definite approach to the problem possible.

The road-load method is based on the use of electrically driven tanks which operate under their own power. By proper instrumentation, the voltage, temperature, and speed of the traction motors may be recorded, in addition to the acceleration of the tank. With these data and proper correction factors, the horsepower available at the sprocket teeth can be determined. When the tank is operated alone at constant speed, the horsepower at the sprocket is completely dissipated in the track and suspension.

One project (Ref. 66) employed several electrically driven tanks. Each tank was equipped with a 500 hp gasoline engine; a six pole, direct-current generator; and two direct-current traction motors mounted in the final drive housing. The power lost between sprockets and ground was estimated to be 40 to 50 percent chargeable to the road wheels and 50 to 60 percent chargeable to the tracks. During the hundreds of tests, the precise amount of horsepower loss varied over a wide span because of the various speed ranges, terrain, and type of track and suspension. The tests established that the three factors which were the most important contributors to track losses are (a) track tension, (b) hysteresis in rubber blocks, and (c) track block construction.

As an attempt to reduce high *track tension,* which causes high power losses, the *flat-track* system was developed. In this system, the track support rollers and idlers are eliminated by increasing the diameter of the road wheels to support the top portion of the track. In addition to permitting satisfactory operation at lower track tension, wheel losses of the idler and the first and last wheels are reduced.

Evaluation of the test data indicated that the power losses resulting from hysteresis in the track and road wheels could be reduced by decreasing tread thickness and by developing low hysteresis elastomers. Steel tracks did not contribute as much to power loss as did all-rubber tracks.

Track block construction is considered a very important factor because of its interrelationship to other factors such as vibration, track weight, and wheel and block impact. The effect of track block vibration can be decreased with reduction of track weight by providing smoother rolling surfaces for the road wheels and by preventing the track blocks from rocking. In addition, the tests determined that a considerable amount of power is lost as a result of impact of the track shoes on the ground, center guide friction, and track vibration.

The project also revealed that type of terrain was perhaps the most important external factor contributing to track suspension power losses. Off-the-road operations were found to require up to 270 percent more power than operations on paved surfaces.

Experimental studies of the type discussed are valuable as guides in the development of improved vehicular systems. Based on these studies, suspensions can be modified to reduce power losses. The gain in usable power means increased mobility in adverse terrain and is reflected in improved tractive effort, greater speed and maneuverability, or greater load-carrying ability.

10-38.3 FLOTATION AND TRACTION

As indicated in the introductory paragraph, two of the most important functions of a track are to provide support for the vehicle (flotation) and traction under widely varying ground conditions. Flotation and traction depend on the properties of the soil as well as the design of the tracks and related suspension components. These factors are discussed in Chapters 4 and 5.

10-38.4 COMPONENT DURABILITY

10-38.4.1 Steel Blocks or Links

The durability of steel track blocks or links is a function of structural design, type of steel, and method of fabrication; other factors being equal. Structural design, in turn, is based essentially on a compromise between allowable weight and structural strength. All-steel, rubber-bushed blocks or links, either double-pin or single-pin, generally have an average endurance life of 3000 miles as compared with 1500 miles for various rubber-faced, rubber-backed constructions. The endurance life of steel rubber-backed blocks is similar to that of all-steel blocks.

10-38.4.2 Rubber Blocks or Links

The application of rubber on both tread and road-wheel surfaces of track shoes has presented several endurance problems resulting from characteristics inherent in rubber construction. In this respect, the following four problems are the most important to the track designer

(a) Bonding of rubber to steel.
(b) Resistance to blowout caused by internal heat build-up.
(c) Excessive hysteresis resulting in power loss.
(d) Resistance to cutting, chipping, shelling-out, and abrasion.

A cold rubber process has resulted in synthetic rubber tracks of satisfactory endurance characteristics. The best synthetic of this type, CPR 41°F, is produced in Government synthetic rubber plants, and is compounded and processed in commercial rubber plants. Since the fall of 1950, CPR 41°F, has been used exclusively in tank tracks (except in pin bushings which are natural rubber compounds) because it is superior to both natural rubber and GR-S (Government Rubber-Styrene) synthetic rubber in most characteristics.

The rubber must be firmly bonded to the metal structure of the shoe to prevent separation under severe operating conditions. Formerly, a relatively good bond was obtained by vulcanizing the rubber to the metal surface which had been brass plated. Another method incorporated a rubber latex material (Ty-ply) between the metal and rubber. Currently, rubber-to-metal cements are used extensively in track shoe bonding processes. Some samples have shown a resistance to separation far above the 150 lb per inch of width required by recent engineering change orders.

Resistance to blowout is generally an important consideration in desert operations or in any area

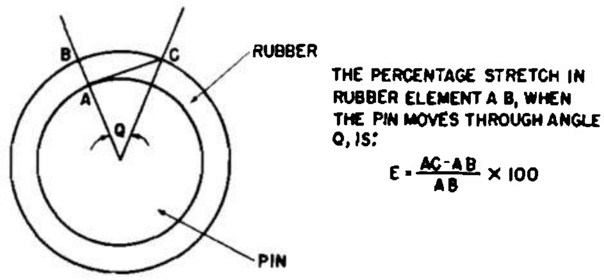


Figure 10-54. Diagram Showing Stretch in Rubber Bushings

with high ambient temperatures. High internal temperatures build up and cause the rubber to blowout, or separate internally, decreasing the usable life of the block. During rubber track qualification tests, a careful check of the internal temperature of the rubber is made; temperatures exceeding 300°F are indicative of low resistance to blowout.

Resistance to wear includes resistance to chunking, chipping, cracking, and shelling-out, as well as abrasion resistance. If operating conditions are assumed to be equal, resistance to wear is primarily a function of type of rubber compound employed.

Another characteristic of rubber, which depends on the type of compound used, is hysteresis—the energy lost or absorbed in flexing rubber due to internal friction. A fairly large amount of engine horsepower is required in performing this flexing action, and may be considered wasted over and above that necessary to cushion the suspension system. Low hysteresis synthetic compounds that have high durability are the aim of present track development programs.

10-38.4.3 Wheel Tread Surface

The rubber wheel tread surface which supports the road wheels is far less subject to wear than the road surface. In fact, when flat, symmetrical rubber block tracks were used extensively, the blocks were turned over when the road side became badly worn. The problem of finding a satisfactory wheel tread compound is not acute and is usually solved satisfactorily by employing the same type of stock used in the tread. The use of rubber-backed tracks has, in many cases, nearly doubled the life of road wheel tires.

10-38.4.4 Track Pins

Two basic types of pins are used in present tracks—cylindrical for double-pin tracks, and octagonal or hexagonal for single-pin tracks. Each basic type of pin presents an important design problem which is not found in the other. Hexagonal or octagonal pins offer difficulty in retention. Cylindrical pins present a load distribution problem which, if not controlled properly, results in excessive bushing wear from pin bending.

A considerable amount of development work has gone into solving the problem of single-pin retention and wear, resulting in the use of series bushing in each lug or eye of the hinge sections and safety nuts and washers on the ends. Typical safety or lock nuts and washers are shown in Figure 10-50. The deflection of cylindrical pins varies as the cube of the pin length; therefore, the distance between the supports must be held to reasonable values. With excessive pin bending, the rubber at the extreme ends of the pin becomes very highly loaded and life of the rubber decreases. Hinge-type tracks should be used where unsupported pin lengths exceed 8 or 10 inches. This arrangement is necessary to afford multiple supports and to obviate the necessity of using large-diameter pins which add excess weight to the tracks.

10-38.4.5 End Connectors and Grousers

Two main problems arise in the design of end connectors (a) securing the pins within the connector and (b) wear at the sprocket surface. Wedge and lock nuts (Figure 10-46) have, for the most part, solved the first problem. The second problem has been partially solved by improved metallurgical properties and heat-treating methods, and improved sprocket tooth contour.

Steel grousers are either integral parts of the block or are attached by some method such as welding. Each method presents fabrication problems. Hard alloy surfacing and heat-treatment are also required to effect maximum wear characteristics.

In rubber grousers, the height and surface area are considered in the durability rating and have an effect on tractive effort.

10-38.4.6 Double-Pin Track Bushings

Rubber doughnut-type bushings have been employed in double-pin tracks. Each grommet or doughnut is bonded to the pin and is compressed 30 to 40 percent of the initial height of the doughnut when assembled in the track block. Various rubber compounds, both natural and synthetic, have been developed and tested in an effort to obtain maximum durability. The tests resulted in the selection of a low-set, high-tensile-age-resistant natural rubber compound of 60-70 durometer hardness. This type of stock is also compounded to resist "cold flow" or "creeping" in service. Softer and harder stocks were tried and found unsatisfactory with respect to fatigue life. Substitution of GR-S (Government Rubber-Styrene) synthetic rubber was found unsatisfactory, attaining only one-quarter the life of natural rubber.

During the development of doughnut bushings, attention was given to the Harris-type bushing. In this type, the rubber is stretched approximately 120 percent by a reducing die and is trapped between the pin and the track shoe bore, providing high frictional engagement with the metal. The Harris-type bushing is not as satisfactory as the doughnut type and is not used in current track application.

The trend toward increased pitch in tracks during the past few years has inspired experimental work on still other types of bushings (increased pitch often reduces bushing life). Various powdered metals and a lubricant-impregnated sleeve of heavy canvas duck have been tried. An experimental track in which needle bearings replaced the bushings also has been tested. Although some of these materials and methods show promise, none possess the overall satisfactory characteristics of rubber doughnut bushings. The Germans developed a "luxury" bushing which incorporated bearings and seals but discarded it during the pressure of war-time production.

10-38.4.7 SINGLE-PIN TRACK BUSHINGS

The early designs of single-pin tracks were based on the dry-pin type used in British tracks. Wear and the consequent increase in pitch of the dry-pin track were excessive, necessitating development of rubber-bushed, single-pin tracks. However, before the application of rubber bushings could be effected, the problem of durability and pin retention methods had to be solved. Since pins must be removed to disconnect the track for maintenance, the permanently installed rubber-doughnut bushed pins used in double-pin tracks were not suitable. To permit pin removal from bushed track lugs, rubber-bushed pin sleeves were developed. These sleeves were installed in the hinge lugs on one or both sides of each track shoe.

Sleeves installed in one side only were keyed to the track pin either by a full-length key formed in the sleeve and a keyway in the pin or by making both sleeve and pin octagonal. In the former construction, the pins were locked in the unbushed lugs of each joint by various methods—such as square keys, taper pins, and set screws—none of which was very successful. The minimum clearance necessary for installation of the pins in the sleeves permitted a torsional lash between sleeve and pin that quickly wore out the keys. The octagonal pin and sleeve design was more successful but still subject to excessive wear.

Sleeves installed in both sides of each track shoe had serrated ends in the manner of a face clutch. The track pin, with nuts or cap screws served to hold these face-serrated sleeves in engagement with each other and prevent relative turning. Half of the torsional deflection at each joint was taken by the bushings of each adjacent shoe instead of the entire deflection occurring in one set of bushings. The superior bushing life obtained with this design was somewhat offset by a tendency of the bushings to creep together in service, making track maintenance difficult.

Serrated bushings have been superseded by bushings installed in the hinge eyes. Each eye contains a doughnut-bushed, internally octagonal sleeve which holds the octagonal pin. The pin is secured at the ends by a washer and safety nut. This arrangement, as well as the double-pin arrangement, generally outlasts the shoe.

Besides rubber composition and pin arrangement, the most important factors related to bushing durability are the unit pressure on the projected area of the pin and the unit stretch in the rubber. With other factors equal, fatigue life depends on these two factors. The diagram in Figure 10-55 demonstrates that, as the pin diameter increases,

TABLE 10-5
TYPICAL PIN-TYPE TRACK AND RUNNING GEAR DESIGN COMBINATIONS

Track Type	Block Type			Pin Type		Link Type		Wheel*	
	All-Rubber	Rubber-Back, Metal-Face	All-Metal	Rubber-Bushed	Dry-Pin	Single-Pin	Double-Pin	Rubber Tire	All Steel
1	X			X			X	X	
2		X**		X			X	X	
3		X**		X		X		X	
4			X**	X			X	X	
5			X**	X		X		X	
6			X		X	X		X	
7			X		X	X			X

*Single, dual, or staggered multiple.
**Removable rubber grouser pad for highway or training operation.

Miscellaneous general or variable factors:

(a) Pitch
(b) Sprocket drive types: front or rear
(c) Grouser pattern: shape or height
(d) Track guide type: center or outer
(e) Track support type: roller or wheel return.

the wall thickness of rubber must increase to maintain the same percentage stretch in the rubber. Tests show that the fatigue life of the rubber varies nearly as $\left(\frac{1}{E}\right)^{1.5}$, where E is the percentage stretch.

Bushings are intentionally assembled in the track blocks such that the circumferential stretch in the rubber is at a minimum. For example, if the angular movement of the pins relative to the track blocks is 14 degrees while going around the sprocket, the pins and bushings must be so inserted in the blocks that the rubber flexes seven degrees either side of the neutral position. This presetting arrangement tends to make rubber-bushed tracks curl at the ends when stretched full length on the ground.

The deflection of the rubber bushings is an advantage rather than a disadvantage, provided, excessive stretch does not result in track throwing. Due to its elasticity, the rubber softens impacts and shock loads on the track, idlers, sprockets, suspension system, and final drive; and contributes to the increased durability, quietness, and high speeds of rubber-bushed tracks.

10-38.5 STATIC SUPPRESSION

During tank movement, static electrical charges build up on tracks as a result of friction at the tread and road wheel surfaces. These charges remain on the tracks as they leave the tires and ground, and are discharged at the sprockets, idlers, and support rollers. Such discharges set up static interference in the communication system of the tank.

To eliminate this condition, various methods of electrical bonding have been tried. Braided copper or bronze strips wrapped around the pin near the center were designed to contact the pin sleeve in the shoe. Generally, these units failed in fatigue or lost sufficient spring action to maintain contact with the sleeve. Rings of conductive rubber also were used but the electrical resistance of the rubber was found to change with flexing during operation. At the present time, piano wire is bonded to the pin and bushing. This arrangement satisfactorily connects the sleeve in the shoe and the pin for the life of the shoe.

10-38.6 TRACK THROWING

Track throwing is essentially a function of such factors as track tension and stretch, pitch, sprocket engagement and conture, and guide design. With present rear drive torsion bar suspensions, track jumping is most prevalent in longer tanks employing double-pin tracks. The track tends to "belly out" and shorten the pitch, causing improper sprocket tooth engagement and subsequent track jumping. The application of a torsion bar-sprung tensioning roller between the last road wheel and sprocket has greatly reduced the tendency to belly out and climb the sprocket. In light tanks, with single-pin tracks, little or no difficulty is experienced with track throwing. Track throwing tendencies that do occur in these tanks are generally eliminated by correcting sprocket contour.

10-39 EVALUATION OF TRACK TYPES

Each basic type of track due to design and material application possesses certain advantages and disadvantages not found in the others. Knowledge of comparative performance characteristics is not only necessary in selecting the best track for a particular operation, but is important in development of present designs and as a guide to future designs. Descriptive data of typical track and running gear design combinations are listed in Table 10-5. For convenience, each track is designated by a type number and should be referred to in reading the text which follows. Data on dry-pin tracks used on many foreign tanks are included for comparison.

Type 1

The all-rubber, rubber-bushed, double-pin block track is one of two basic types of rubber tracks in current use, and is exemplified by the T84E1 track (Figure 10-48). The all-rubber block was used in quantity on M3 and M4 series tanks during World War II but was discarded for general combat use in favor of an aggressive steel-shod block.

The principal advantages of this type of track are

(a) Lighter than equivalent steel tracks.
(b) Higher coefficient of friction than steel on dry hard road surfaces.
(c) Reduced noise level.
(d) Increased tire life.

The principal disadvantages are

(a) Relatively low durability (average endurance life of current synthetic rubber tracks is 1500 miles compared with 3000 miles for steel tracks of the same type).
(b) The rubber road surface wears out long before other components have reached their potential terminal lives.
(c) High power loss due to resilience of rubber blocks.
(d) Poor wheel hop and induced vibration characteristics.
(e) Traction in mud reduced as chevron wear is increased.

Type 2

Type 2 track is a metal-faced, rubber-bushed, double-pin block type. It has been used in the past because of long life and its provision of protection for road wheel tires and support rollers. The T80 and T87 series tracks exhibited lives of approximately 4000 miles.

The principal advantages of this type of track are

(a) Incorporates a permanent aggressive steel grouser.
(b) Provides a resilient path of tire protection (tire life is increased 50 percent under average conditions including a debris laden track, as compared with all steel blocks).
(c) Relatively long endurance life (four times that of all-steel, dry-pin tracks).
(d) Protects suspension components because of resilience.
(e) Processes and materials are readily available for manufacture.

The principal disadvantages are

(a) Power loss due to resilience of rubber blocks.
(b) Difficulty of incorporating a detachable rubber pad.
(c) Relatively high weight.
(d) Relatively high initial manufacturing cost.

Type 3

Type 3 track is similar to Type 2, except that it employs a single-pin link. A removable rubber grouser pad can be readily incorporated as required. Average life is somewhat less than Type 2, essentially because of wear and nonreplaceability of sprocket drive surfaces. Machining of cast track bodies is somewhat difficult, while hammer facilities for the more easily machined forgings are limited when large parts are employed. Typical examples of this type track are T91E3 (cast steel) and T95 (forged steel).

TABLE 10-6
PHYSICAL PROPERTIES OF RUBBER COMPOUNDS

Property	Natural Rubber	Synthetic GR-S	Synthetic CPR 41°F
Hardness, Shore Durometer (Type A)	65–80	65–80	65–80
Tensile Strengths:			
Before aging, psi (min)	3000	1800	2750
After aging in air oven, psi (min)	2250	1500	2250
After aging in oxygen bomb psi (min)	1650	1500	2250
Elongation:			
Modulus at 400%, psi (min)	2000	—	1800
Before aging, percent (min)	435	350	400
After aging in air oven, percent (min)	325	300	300
After aging in oxygen bomb, percent (min)	260	300	30
Specific Gravity (max)	1.25	—	1.25

Type 4

The all-metal, rubber-bushed, double-pin block track has the advantages of lower wheel hop and induced vibration, and lower power loss due to rubber resilience. Disadvantages include reduced tire life and poor icing characteristics. Examples of this type track are the obsolete T49 and T54 series used on M3 and M4 series tanks.

Type 5

The all-metal, rubber-bushed, single-pin block track requires a minimum of strategic materials and offers most of the desirable characteristics of dry-pin tracks while still avoiding the short life and higher maintenance requirements of the latter. A typical track of this type is the T82E1 which was used on Cargo Tractor T43.

The principal advantages of this type of track are

(a) Reduced power loss.
(b) Reduced block height or thickness.
(c) Use of low cost materials and processes.
(d) Applicable to detachable rubber pad construction.
(e) Flame-resistant.
(f) Reduced wheel hop and induced vibration.
(g) Approximately three times the life of dry-pin types.

The principal disadvantages are

(a) Reduced tire life.
(b) Reduced track life.
(c) Servicing difficulties inherent in some types of single-pin bushing designs.
(d) Poor icing characteristics.

Types 6 and 7

Types 6 and 7 are foreign-type tracks. The selection of these design combinations is based on simplicity of design and manufacture, with corresponding economies in initial investment, manufacturing equipment, materials, and man-hours of production.

Specific advantages of these tracks are

(a) Dry-pin and metal-block design offer low initial cost, low weight, manufacturing simplicity, ease of maintenance, low elastic stretch, low wheel hop and vibration incidence, and low power loss.
(b) Steel-rimmed wheels offer low power loss, low initial cost, elimination of tire deficiencies.

Specific disadvantages are

(a) Dry-pin and metal block design produce short track life (particularly when operating in abrasive sand or mud), no elastic protection for tires, poor icing characteristics, frequent track adjustment due to pin-eye wear, no elastic protection for

TABLE 10-7
DURABILITY OF SYNTHETIC AND NATURAL RUBBER IN MILES OF VEHICLE OPERATION

Vehicle Weight, lb	Average Speed, mph	Synthetic Rubber, miles	Natural Rubber, miles
Under 60,000	25	2000	3000
60,000 to 100,000	20	1500	2000
Over 100,000	15	1000	1500

driving train, poor high speed stability in track circuit, and high noise level.

(b) Steel rimmed wheels produce low life, high noise level, no protection from shock and high frequency vibration to the running gear, and poor high speed characteristics.

10-40 TESTING

At the present time, all production tank tracks are of rubberized construction on both tread and wheel faces. Steel rubber-backed tracks are carried in stock and are a part of some experimental programs but are not used on production tanks. The relationship of certain physical properties of rubber and the durability of rubber has been established during the extensive rubber track development program. The physical requirements necessary to obtain the maximum possible performance characteristics of rubber at the present stage of rubber development are set forth in Military Specification MIL-S-11891 (ORD), 28 March 1952.

Besides being inspected dimensionally and visually for such obvious defects as excessive rubber flash and cracks, the specific type of rubber—natural, GR-S, or CPR 41°F—must meet the requirements listed in Table 10-6. A sufficient number of samples are taken from a lot to establish the general quality, and tests are conducted in accordance with MIL-R-3065. If shoe design is such that specimens of the desired size cannot be obtained, the specimens may, with the approval of the procuring agency, be taken from slabs of the rubber stock intended for bonding to the metal.

The quality of adhesion of rubber to metal is of the greatest importance in securing satisfactory mileage of shoes and pads. Since the method used to obtain this adhesion contains the most variables of any in the manufacturing process, frequency of inspection is increased whenever test samples barely meet minimum requirements. The method used to determine bonding strength consists of "starting" a one inch wide strip of rubber at a specified section and then pulling it on a tensile testing machine. The load required to separate the rubber from the metal should be not less than 60-100 pounds, depending on the section and shoe type tested. Recent engineering change orders have indicated the desirability of 150-pound minimum loads in many instances.

Even though bonding qualities and physical properties of the rubber meet laboratory requirements, the assembled track must be installed on a tank and pass a durability test. The number of satisfactory miles of operation required with either natural or synthetic rubber for a specific tank weight are listed in Table 10-7. One-third of the test is conducted on paved roads (concrete, asphalt, or macadam); one-third, on gravel composed mostly of small stones; and the other one-third over cross-country terrain. Not more than 150 miles of continuous operation is maintained on one type of terrain. The temperature at the center of the rubber parts during these operations must not exceed 300°F, as measured by a thermocouple or other suitable means.

Before the durability test, the track shoes and/or pads are broken in by operating the test vehicle for 45 miles on a paved road.

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GLOSSARY

acceleration. A vector quantity that expresses the time rate of change of velocity and is, thus, the first derivative of velocity and the second derivative of displacement of a moving body with respect to time.

Ackermann steering. The standard system of steering in which the front wheels are mounted on pivoted knuckles and are interconnected by a linkage. During a turn, the inner wheel rotates through a larger angle than does the outer wheel.

actual ground pressure. *See*: **ground pressure.**

adhesion. The shear strength of the bonding between soil and other materials under zero externally applied pressure.

amplitude. When applied to vibrations, it is the maximum value of a sinusoidal quantity.

analogy. A recognized relationship of consistent mutual similarity between the equations and structures appearing within two or more fields of knowledge, and an identification and association of the quantities and structural elements that play mutually similar roles in these equations and structures, for the purpose of facilitating transfer of knowledge of mathematical procedures of analysis and behavior of the structures between these fields.

angle of approach. The maximum angle of an incline onto which a vehicle can move from a horizontal plane without interference, e.g., from front bumpers.

angle of departure. The maximum angle of an incline from which a vehicle can move onto a horizontal plane without interference, e.g., from rear bumpers.

angle of internal friction. Angle φ between the abscissa and tangent of the curve representing the relationship of shearing resistance to normal stress acting within a soil. Tan φ is analogous to a coefficient of friction.

angular frequency. *See*: **circular frequency.**

articulated steering. The system of steering used by tracked or wheeled vehicles consisting of two or more powered units in which the turning maneuver is accomplished by yawing the units with respect to each other about a pivot system not located over an axle of either unit.

articulated vehicle. A tracked or wheeled vehicle system consisting of two or more powered units.

athey wagon. A classification of heavy duty, unpowered, tracked cargo trailers designed to be towed behind track-laying prime movers (tractors) for the transport of general cargo over soft or rough terrain.

automobile. A self-propelled, wheeled vehicle, generally commercially designed, for transporting less than ten (10) passengers on highways and/or roads. Excludes bicycles, motorcycles, and motor scooters.

automotive vehicle. A general category of mechanical land vehicles that contain a means of propulsion within themselves. They are generally considered to be either wheeled or track-laying; but, in the broad sense, this category includes all types of walking and jumping vehicles as well as self-propelled sleds and various air-cushion supported vehicles. They may have the ability to negotiate deep water barriers by swimming on the surface, in which case they are amphibious automotive vehicles, or by swimming submerged, in which case they are submarine automotive vehicles.

auxiliary mass damper. A system consisting of a mass, spring, and damper which tends to reduce vibration by the dissipation of energy in the damper as a result of relative motion between the mass and the structure to which the damper is attached.

axle assembly. A device suspended between and connecting opposite wheels which consists of the housing and driving differential mechanism. The assembly also supports the weight of the vehicle.

axle load. The total load transmitted to the road by all wheels whose centers are included between two parallel transverse vertical planes 40 inches apart, extending across the full width of the vehicle. (From AR 705-8)

axle tramp. The sustained vibration of the axle of a solid axle suspension in a vertical plane.

band track. A continuously flexible track usually comprised of an endless band of rubber reinforced with steel cables.

bearing capacity. The average load per unit area required to produce failure by rupture of a supporting soil mass.

binocular frames. Track units with webbed ends designed to contain the bushings and pins of double-pin tracks.

block and pin track. *See*: **jointed track.**

body. *See*: **hull.**

bogie (tracked vehicles). A suspension assembly in which road wheels (bogie wheels) are interconnected in tandem by a system of arms, walking beams, cranks, springs, etc., in such a manner that when one wheel experiences a vertical force or displacement, a corresponding change in loading or position is reflected in the other wheels of the bogie unit.

bogie wheel. *See*: **bogie (tracked vehicles)**; also **road wheel.**

bogie (wheeled vehicles). A suspension assembly consisting of tandem axles, interconnected by walking beams which pivot vertically about a cross member (trunnion axle). Also, a tandem axle assembly without a distinct walking beam but interconnected by a system of cranks and links in such a manner that when one axle experiences a vertical force or displacement, a corresponding change in load or position is reflected in the other axle.

bottom roller. *See*: **road wheel.**

bounce. The upward movement of the sprung mass of a vehicle, away from the unsprung mass, in response to suspension system disturbances.

bounce distance. The maximum upward travel of the sprung mass of a vehicle, away from the unsprung mass and measured from the free standing position, before further upward deflections of the suspension mechanism are rigidly restrained.

brake fade. A temporary failure in a braking system due to excessive temperature. The term is used especially in connection with automotive vehicles.

camber. A setting of the front or rear wheels of a vehicle, closer together at the bottom than at the top.

car, armored. A wheeled self-propelled vehicle with protective armor plate designed for combat use and usually equipped with armament.

carrier, personnel. A self-propelled vehicle, sometimes armored, used for the transportation of troops and their equipment.

circular frequency. The frequency of a periodic quantity in radians per unit of time. It is the frequency multiplied by 2π.

clay. Soil particles of 0.002 mm and smaller which exhibit plasticity when wet and considerable strength when air-dry.

coefficient of permeability. The rate of discharge of water under laminar flow conditions through a unit cross sectional area of a porous medium under a unit hydraulic gradient and standard temperature conditions.

cohesion. The portion of the shear strength of soil which is independent of normal pressure. It is indicated by the term c in Coulomb's equation: $s = c + p \tan \varphi$.

cohesionless soil. A soil which, when unconfined, has little or no strength when air-dried, and that has insignificant cohesion when submerged.

cohesive soil. A soil which, when unconfined, has considerable strength when air-dried, and that has considerable cohesion when submerged.

combat vehicle. A land or amphibious vehicle, with or without armor or armament, designed for specific functions in combat or battle. The installation of armor or armament on vehicles other than combat vehicles does not change their original classification.

compaction. The densification of a soil by means of mechanical manipulation.

complex periodic vibration. Any periodic vibration that is not a pure sinusoid. It is considered as a sum of multiple sinusoids that are harmonically related to each other.

complex vibration. Any vibration that is not a pure sinusoid and whose sinusoidal components are not harmonically related to each other.

condual tire. A tire consisting of two tubes or carcasses, the major outside diameter of the smaller equal to and nested within the major inside diameter of the other. Each carcass is permitted a maximum deflection compatible with acceptable wear rates. The allowable deflection permitted in this type of tire is approximately double that of conventional tires. The larger deflection obtained leads to a long, thin contact area considered desirable from a soft soil mobility viewpoint.

cone index. An index of the shearing resistance of a soil obtained with the cone penetrometer. It is a dimensionless number representing the resistance to penetration into the soil of a 30 degree cone having a ½ sq inch base area (is actually the load, in lb on the cone base area in sq inches).

consistency. The relative ease with which a soil can be deformed.

consolidation. The gradual reduction in the volume of a soil mass resulting from an increase in compressive stress.

cornering force. The force, in pounds, measured normal to the longitudinal plane of a wheel or track, which is exerted by the ground-contacting area in resisting the centrifugal force developed when a vehicle moves in a nonlinear path.

Coulomb damping (dry friction damping). The dissipation of energy that occurs when a particle in a vibrating system is resisted by a force whose magnitude is a constant independent of displacement and velocity, and whose direction is opposite to the direction of the velocity of the particle.

coupled modes. Modes of vibration that are not independent but which influence one another because of energy transfer from one mode to the other.

critical damping. The minimum viscous damping that will allow a displaced system to return to its initial position without oscillation.

cross-country. Proceeding on a course directly over countryside, as across fields, hills, marshes, woods, and not by roads or paths.

cumulose. Consisting chiefly of accumulated organic matter.

cycle. The complete time-history of a periodically varying quantity—from any starting point, through a complete sequence of values, and back to the starting point.

damped natural frequency. The frequency of free vibration of a damped linear system. The free vibration of a damped system may be considered periodic in the limited sense that the time interval between zero axis crossings in the same direction is constant, even though successive amplitudes decrease progressively.

damping. The process of dissipating energy with time or distance from an oscillatory system.

deep fording. *See*: **fording.**

density. Weight of solids divided by the volume, including voids between the soil particles.

deterministic function. A function whose value at any time can be predicted from its value at any other time.

dilatation. Increase in volume of soil during shearing deformation.

dimensional analysis. Method of analysis based on relationships that must exist between pertinent variables involved in a phenomenon because of their dimensions.

direct shear test. A shear test in which soil under an applied normal load is stressed to failure by moving one section of the soil container (shear box) relative to the other section.

displacement. A vector quantity that specifies the change of position of a body or particle and is usually measured from the mean position or position of rest.

drawbar pull. The amount of tractive effort developed by a vehicle in excess of motion resistance (net tractive effort).

drawbar pull-weight ratio. An index of the efficiency of a vehicle system similar in concept to the lift-drag ratio for an aircraft. The drawbar pull-weight ratio indicates the effort available for hill climbing, vehicle acceleration, load towing, etc.

dry friction damping. *See*: **Coulomb damping.**

dual tires. A pair of tires mounted on dual wheels.

dual wheels. Two wheels, bolted together and mounted on the same end of an axle shaft. For purposes of vehicle nomenclature, dual wheels are counted as one wheel.

durability. That characteristic, pertaining to an object, device, or system of devices, related to the period of time of satisfactory operation on a comparative basis. If two or more comparable items are subjected to the same operating conditions, the one that operates satisfactorily for the longest period of time is the more durable. Ability to withstand abuse is also a characteristic of a durable unit.

dynamic axle reaction. The motion-induced effective axle loading. Acceleration, braking, air resistance, and drawbar loads affect the axle loading.

elastic girder track. A track in which adjacent links are interlocked by elastic components, such as rubber buffers, to limit reverse bending. Cf. **rigid girder track,** also **flexible track.**

elastic wheel. A resilient wheel such as the pneumatic-tired wheel. The ground-contact area of an elastic wheel on rigid ground is relatively large. Cf. **rigid wheel.**

exoskeletal construction. A construction technique in which the body is a major stressed member. This is the principle of "unit construction" used by some automotive manufacturers and can result in a sizable reduction in vehicle weight.

fighting compartment. Portion of a fighting vehicle in which the occupants service and fire the principal armament. It occupies a portion of the hull and all of the turret, if any.

flat track suspension system. A suspension system on a tracked vehicle wherein the track returns on the top surfaces of the road wheels without the use of supplementary support rollers.

flexible track. A track that can flex in either direction about a horizontal transverse axis. Cf. **rigid girder track.**

floating. The ability of a vehicle to negotiate water obstacles without being in contact with the bottom. Self-propulsion while in the water is not implied in this definition.

flotation. The ability of a vehicle to traverse soft soil without undue sinkage.

forced vibration. A vibration that takes place due to the excitation of external forces. It occurs at the frequency of the exciting force and is independent of the natural frequencies of the system.

fording. The ability of a vehicle with its suspension in contact with the ground to negotiate a water obstacle of a specific depth. *Shallow fording* is fording without the use of special waterproofing kits, while *deep fording* is fording of greater depths with the application of a special waterproofing kit.

frame. A structure, separate from the body or hull, that supports the various components of the automotive assembly and maintains their spatial relationship. The frame provides strength and rigidity to the vehicle.

free vibration. A vibration that occurs due to the action of forces inherent within an elastic system and in the absence of external impressed forces. The system will vibrate at one of its natural frequencies.

frictional soil. Same as **cohesionless soil.**

full-track vehicle. Vehicle entirely supported, driven, and steered by means of tracks.

Goer type vehicle. A four-wheeled vehicle having a combination of distinguishing features—large diameter tires, exoskeletal construction, powered wagon wheel steering, power to all wheels, and suspension system consisting of tires only.

gradeability. The slope-climbing ability of an automotive vehicle.

gradation. Proportion of material of each grain size present in a given soil.

grade. *See*: **slope.**

grade resistance. The motion-resisting force acting on a vehicle traveling up a grade. For a vehicle going down a grade, the grade resistance force becomes negative.

gravel. Particles of rock that pass a 3-in. sieve but are retained on a No. 4 standard sieve.

gross tractive effort. The maximum propelling force that can be developed by the ground-contacting elements of a vehicle on a given type of support.

gross vehicle weight. The chassis or the hull weight, plus the weight of the entire body, fully equipped and serviced for operation, plus operating personnel.

ground-contact area. The area of the ground-contacting element of a suspension system that is in contact with the ground and has a function in supporting the weight of the vehicle. On soft ground, it is assumed to be the product of the overall length and width of the area in contact, including all open spaces between components of the ground-contacting element. On rigid surfaces, it is the actual area in contact with the ground, exclusive of the open areas.

ground pressure. The force exerted by a vehicle on the ground, usually expressed in pounds per square inch. *Mean ground pressure* equals the gross weight of the vehicle divided by the ground contact area in soft ground. *Actual ground pressure* can be obtained only from complex calculations that take into consideration the unequal wheel loading, flexibility, form, and dimensions of the ground-contacting element. The actual ground pressure is usually nonuniformly distributed beneath the ground-contacting element. Cf. **ground-contact area.**

grouser (spud). A detachable or integral projection (often chevron-shaped) on a track shoe, normal to the tread surface, provided for improved traction in off-the-road operations.

guide horn. *See*: **track guide.**

half-track vehicle. A vehicle in which some wheels (usually the front steered wheels) run without tracks while the others run on tracks.

handling. The subjective evaluation of the maneuvering and course-keeping characteristics of an automotive vehicle. It includes (a) the ease and precision with which a vehicle can be steered or maintained on a desired course, (b) its overall speed of response to control, and (c) the stability of the vehicle.

harmonic. A sinusoidally varying quantity having a frequency that is an integral multiple of the frequency of a periodic quantity to which it is related.

harmonic motion. A motion such that the displacement is a sinusoidal function of time.

helical coil spring. Round, square, or rectangular wire, wound in the form of a helix, offering a resistance to a force applied along the axis of the coils. When wound with space between coils, they may be loaded in compression. When the force is applied in a manner that separates the coils, it is termed a helical tension spring.

helical torsion spring. Round, square, or rectangular wire, wound in the form of a helix, offering a resistance to a moment applied in a plane perpendicular to the coil axis.

horizon. The geological deposit of a particular time, usually identified by the presence of distinctive fossils. Also one of the layers of a soil profile distinguished principally by its texture, color, structure, and chemical content.

Hotchkiss drive. In automotive vehicles, a method of drive by which the torque reaction is transmitted to the frame through the springs rather than through a torque tube or a torque arm.

hull. The main structure which forms the passenger, cargo, and component compartments. The term *body* is usually applied to wheeled vehicles, while the term *hull* is applied to amphibious and tracked vehicles.

hydraulic spring. A sealed plunger working in a highly finished cylinder, against an enclosed volume of liquid. The resiliency of the spring is derived from the compressibility of the liquid at high pressure.

hydropneumatic. Pertaining to, or operated by means of, a liquid and a gas; used with recoil and equilibrator mechanisms which provide variable absorption of energy or thrust.

hydropneumatic spring. A self-contained spring and shock absorbing unit comprised of an enclosed volume of gas and fluid separated from each other usually by a flexible diaphragm or a piston. The system derives its elasticity from the compressibility of the gas, while the fluid provides system damping, vehicle leveling, and ground clearance control.

hydrospring. Pertaining to, or operated by means of, a liquid and springs; used with recoil and equilibrator mechanisms which provide variable absorption of energy or thrust.

idler. On track-laying vehicles, the wheel at the end of the vehicle opposite the driving sprocket, over which the track returns. It maintains track tension and reduces track skipping.

impedance, mechanical. The ratio of a force-like quantity to a velocity-like quantity when the arguments of the real (or imaginary) parts of the quantities increase linearly with time. Impedance is the reciprocal of mobility.

impulse. The product of a force and the time during which the force is applied. More specifically, it is $\int_{t_1}^{t_2} F dt$, where F is time dependent and equal to zero before t_1 and after t_2.

impulse shock. A particular type of shock for which the waveform can be approximated by assuming a waveform of simple shape.

independent suspension. A system of arms, springs, wheels, etc., for elastically supporting the sprung mass of a vehicle, which permits the deflection of any of the supporting wheels without substantially changing the load or position of the remaining wheels (as distinguished from solid axle or bogie suspension systems).

inertia resistance. As applied to an automotive vehicle, the resisting forces opposing the linear and angular accelerations of the various masses of the vehicle.

internal friction. The portion of shear strength of soil which is proportional to the normal stress on the shearing surface. It is indicated by the term $p \tan \varphi$ in Coulomb's equation: $s = c + p \tan \varphi$.

internal motion resistance. Resistance to motion of a vehicle due to forces acting within and upon the vehicle such as friction between moving parts, hysteresis, inertia, vibrations, etc.

isolation. A reduction in the capacity of a system to respond to an excitation, attained by the use of a resilient support.

jerk. A vector quantity that specifies the time rate of change of acceleration. It is the third derivative of displacement with respect to time.

jointed track (block and pin track). A track comprised of rigid links connected by joints at which flexing occurs. Cf. **band track.**

jounce. The downward movement of the sprung mass of a vehicle, toward the unsprung mass, in response to suspension system disturbances.

jounce distance. The maximum downward travel of the sprung mass of a vehicle, toward the unsprung mass, and measured from the free standing position, before downward deflections of the suspension mechanism are rigidly restrained.

leaf spring. A flat bar spring that is relatively thin in proportion to its length and width, designed to be loaded in bending. In vehicle suspensions, leaf springs usually are a lamination of several leaves of unequal lengths.

longitudinal slope. A slope that is in line with the fore and aft axis of a vehicle. *See*: **side slope.**

L/T ratio. A steering ratio in which L represents the length of track in contact with the ground and T represents the lateral distance between the centerlines of the tracks.

maintainability. The combined qualitative and quantitative characteristics of material design and installation which enable the accomplishment of operational objectives with minimum maintenance expenditures including manpower, personnel skills, test equipment, technical data, and facilities under the operational environmental conditions in which scheduled and unscheduled maintenance will be performed.

maintenance. All action taken to retain materiel in a serviceable condition or to restore materiel to serviceability. It includes inspecting, testing, servicing, classification as to serviceability, repairing, overhauling, modifying, modernizing, and rebuilding.

mean ground pressure. *See*: **ground pressure.**

mechanical shock. A nonperiodic excitation—e.g., a motion of the foundation or an applied force —of a mechanical system that is characterized by suddenness and severity, and usually causes significant relative displacements in the system. The duration of the excitation is for a relatively short time with respect to the fundamental natural period of the equipment concerned.

military characteristics. Those characteristics of equipment found desirable or necessary to the performance of a military mission, either combat or noncombat. Military characteristics are prescribed by the using arms and usually form the basis of initiating development of a new item.

mobility. The competence of a vehicle to perform its mission as measured by its best average speed over a route representative of the terrain where it will operate.

mobility index. A dimensionless number obtained from a mathematical interrelationship of certain characteristics of a vehicle. It can be used to determine the **vehicle cone index.**

model. A device so related to a physical system that observations on the model may be used to predict performance characteristics of the physical system.

module. A functional and structural assembly or subassembly so designed that, if it becomes unserviceable, it may be readily replaced by a like serviceable unit.

moisture content. Ratio, expressed as percentage, of the weight of water in a soil mass to the weight of solid mineral particles.

natural frequency. The frequency of free vibration of a system.

particle size. A particle dimension, customarily considered to be the diameter.

pedology. The science concerned with the classification of soils based upon their origin and development and their consequent physical and chemical characteristics.

penetrometer. Probe advanced into the ground to obtain a measure of soil strength.

percent of slope. Angle of ascent or descent expressed as a percent; the number of units a slope rises, or falls, vertically in a horizontal distance of 100 identical units.

period. The shortest time interval in which a periodically varying quantity repeats itself. It is the duration of one cycle.

periodic vibration. A vibration that repeats all of its characteristics after fixed time intervals.

pin-pointed track. A track of a track-laying vehicle in which the flexing occurs as angular oscillations of the journals about their pins, resulting in a sliding of the surfaces.

pitch. The angular displacement of a vehicle about an axis parallel to its lateral (horizontal) axis. Cf. **roll** and **yaw.**

plasticity. The property of a soil which allows it to be deformed beyond the point of recovery without cracking or appreciable volume change.

pneumatic spring. A self-contained spring assembly that derives its spring action from the compressibility and elasticity of an enclosed gas.

power spectral density. A measure of the amplitude of each frequency component in a random wave such as a ground profile; essentially the mean of the amplitude squared for each frequency.

prime mover. In a device of two or more moving parts, that unit considered to be the source, or principle source, of energy for movement, as with a tractor pulling a trailer.

random vibration. An oscillation whose instantaneous magnitude can be specified only by probability distribution functions giving the probable fraction of the total time that the magnitude will be within a specified range.

rating cone index. The measured **cone index** multiplied by the **remolding index**; it expresses the soil strength rating of a soil at the place tested.

reliability. The probability of a device performing its purpose adequately for the period of time intended under the operating conditions encountered. For a system with independent components, the overall reliability is based on the product of the individual reliabilities; e.g., three independent components with a 90-percent reliability each will have an overall reliability of $.9 \times .9 \times .9$ or 72.9 percent. Similarly, 100 components with a 99-percent reliability each will have an overall reliability of only 36.5 percent. Mechanical reliability as applied to automotive equipment also includes the capacity of a vehicle to perform its mission after sustaining failure or destruction of specific components.

remolding. The changing of the strength of a soil due to working by traffic or by a remolding test. Remolding may have a beneficial, neutral, or detrimental effect upon soil strength.

remolding index. The ratio of the remolded soil strength to the original soil strength, determined in accordance with established standard procedures.

residual soil. A soil resulting from the weathering in place of a parent rock and which was not moved during the weathering process.

resonance. Resonance of a system in forced vibration exists when any change, however small, in the frequency of excitation causes a decrease in the response of the system.

return roller. *See*: **return wheel.**

return wheel (top roller, return roller). One of a number of wheels that support the top run (return run) of the track between the drive sprocket and idler of a track-laying vehicle.

reverse bending. Flexing of a track in a direction opposite to that assumed when passing around the sprockets.

ride. A subjective evaluation of the roll, pitch, and bounce characteristics of a vehicle with respect to operator and passenger comfort.

rigid girder track. A track in which adjacent links interlock to form a girder that is rigid in one direction, thus preventing reverse bending. Cf. **elastic girder track** and **flexible track.**

rigid wheel. A wheel that deforms a relatively negligible amount on a hard surface, and in the limiting case has a line ground-contact pattern. A steel railway wheel is an example of a rigid wheel. Cf. **elastic wheel.**

roadability. A rating of the collective operating characteristics of an automotive vehicle that defines the quality of the vehicle's traveling performance. Included in this total rating are such factors as ease of steering, gradeability, acceleration, road holding, suspension stiffness, rebound control, directional stability, braking characteristics, skidding characteristics, etc.

road wheel (bogie wheel, bottom roller). One of a number of wheels which support the weight of a tracked vehicle and roll on the inside of the bottom run of the track.

roll. The angular displacement of a vehicle about an axis parallel to the vehicle's longitudinal axis. Cf. **pitch** and **yaw.**

roll axis. *See*: **roll center.**

roll center. The center about which a portion of the total sprung mass of a land vehicle rotates when a side force is imposed on the vehicle. The position of the roll center relative to the road surface depends on the type of suspension system used. The entire vehicle rotates about a roll axis which is generated by the positions of the major roll centers of the vehicle. For example, a four-wheeled vehicle will have a separate roll center for the front and rear suspension systems, these points being on the roll axis.

rolling resistance. The motion-resisting force developed by the interaction of the wheels or tracks of a vehicle and the ground. When the rolling resistance is subtracted from the gross tractive effort, the effective propelling force remains.

rubber-bushed track. A jointed track incorporating rubber bushings which permit flexing by annular shear, i.e., relative rotation between the inner and outer cylindrical surface of the rubber.

rubber torsion spring. A spring assembly generally consisting of a metal shaft bonded to an annular layer of rubber which is, in turn, bonded to an outer concentric metal shell. Spring action is derived by twisting the inner shell relative to the outer shell by applying a moment in a plane perpendicular to the shaft axis, thus loading the rubber in annular, or torsional, shear.

sand. Soil particles that pass a No. 4 but are retained on a No. 200 U.S. standard sieve.

semitrailer. A nonpowered vehicle having integral wheels at the rear only, and designed to carry material, supplies, or equipment and to be towed by a self-propelled motor vehicle that also supports the front end by means of a fifth-wheel coupling assembly. The front end can also be supported by a dolly that is provided with a fifth-wheel assembly for coupling to the semitrailer and a tongue and lunette for coupling to the prime mover.

shallow fording. *See*: **fording.**

shear strength. The maximum resistance of a soil to shearing stresses.

shimmy (wheel wobble). The vibratory oscillation of the steerable wheels of a vehicle about the kingpins.

shock. *See*: **mechanical shock.**

shock absorber, direct action. A damper, either frictional or hydraulic, designed to damp the shock of suddenly applied forces and/or to control spring rebound and oscillation; usually attached to the vehicle frame, body, or hull and connected to an axle, spring, spring support web, or pad, or between suspension arms of track-laying vehicles.

shock absorber, lever action. A damper, either frictional or hydraulic, designed to damp the shock of suddenly applied forces and/or to control spring rebound and oscillation, usually attached to the frame of a vehicle, with the arm connected by a link or linkage to the axle or spring.

shock isolator. A resilient support that acts to isolate a system from shock forces.

shock spectrum. A plot of the maximum response experienced by single-degree-of-freedom systems, as functions of their natural frequencies, in response to applied shocks. The response may be expressed in terms of acceleration, velocity, or displacement.

shoe assembly. Track unit consisting of link, end connectors and wedges, and capped center guide.

side slope. A slope that is in line with the lateral axis of a vehicle. *See*: **slope.**

silt. Soil particles above 0.002 mm which pass a No. 200 sieve. Silt has only slight plasticity and exhibits only slight strength when air-dry.

sinusoid. A mathematical curve whose ordinates are proportional to the sines of the abscissas, as defined by the equation $y = a \sin x$.

sinusoidal motion. *See*: **harmonic motion.**

slope. A natural or artificial incline to the plane of the horizon, as a hillside, terrace, or ramp. Its measure is usually expressed as a ratio of the vertical rise to the horizontal distance traveled. It is often expressed as a percent and termed "percent slope" or "percent grade."

soil classification system. Grouping of soils by characteristics and properties.

soil thrust. The total soil force capable of being developed into a propelling force for vehicles.

soil values. Parameters used to describe soil properties in equations for predicting performance of vehicles on soil such as sinkage, slip, rolling resistance, and thrust.

sprung weight. The total weight of all of the vehicle components that are supported by the vehicle spring system. This includes such major components as frame, body, power plant, transmission, clutch, cargo, etc. It does not include such items as wheels, axles, road wheels, etc.

spud. *See*: **grouser.**

stability. The property of mechanisms that causes them, when disturbed from conditions of static or dynamic equilibrium, to develop forces or moments to restore their original equilibrium conditions or, if the disturbance remains constant, to establish a new state of equilibrium.

static-steering torque. The torque required to turn the wheels of a stationary vehicle. Actual turning center of a steered wheel is the intersection of the king-pin axis with the ground. The steering motion of the wheel around this point is a combination of sliding and pure rotation.

steering system. The assembly of linkages and components which enables the driver to control the direction of the vehicle. Wheeled vehicles are normally steered by rotating the axes of rotation of two or more wheels with respect to the longitudinal center line of the vehicle, while tracked vehicles are usually steered by varying the speed of the tracks with respect to each other.

suspension system. The mechanical linkages and the elastic members that provide a flexible support for the sprung components of a vehicle.

swimming. The ability of a vehicle to negotiate a water obstacle by propelling itself across, without being in contact with the bottom.

tactical vehicle. Any vehicle designed for field requirements in combat and tactical operations, or for training personnel for such operations.

tank, amphibious. Vehicle mounting a howitzer or cannon, capable of delivering direct fire from the water as well as ashore, and used in providing early artillery support in amphibious operations.

tank, combat, full-tracked. A self-propelled, heavily armored, offensive vehicle having a fully enclosed revolving turret with one major weapon. It may mount one or more machine guns. Excludes self-propelled weapons.

tank transporter. Special purpose wheeled or tracked vehicle, or combination of vehicles, designed to transport tanks or other heavy vehicles over highways and natural terrain and incorporating integral provisions for loading and unloading disabled vehicles without supplemental assistance.

thixotropy. A characteristic of soil that enables it to stiffen in a relatively short time, on standing, but upon agitation or manipulation to change to a soft consistency, the process being completely reversible.

throwaway concept. A maintenance concept in which, upon failure of an assembly, subassembly, component, module, or end item, it will be discarded; no repairs to the item will be made.

top roller. *See*: **return wheel.**

torque rod, tandem axle. A metal device designed to insure correct spacing and alignment of truck and trailer axles.

torque tube. In automotive vehicles, a tube that encloses the propeller shaft and is designed to resist propelling and braking reaction forces while maintaining the spatial relationships between the various interconnected units.

torsion bar spring. A straight bar spring, usually cylindrical, employed as the elastic member in one type of vehicular suspension. One end of the bar is secured in torsion to the vehicle frame or hull while the other end is supported by and free to rotate in a hull mounted bearing. Torsional loads are applied to the bar by means of an arm fastened to the free end and rotated in a plane perpendicular to the longitudinal axis of the torsion bar.

track. The continuous band or segmented chain upon which a tracked vehicle runs. Cf. **track-laying vehicle.**

track body. The basic structural unit of a track link. On double-pin tracks it is the track block. Cf. **track shoe assembly** and **track link.**

track guide. The track projections on the road wheel side of a track that locate the road wheels on the wheel path and transmit lateral forces between the track and road wheels during steering and side slope operation.

track-laying vehicle. A vehicle that utilizes endless belts or tracks to distribute its gross load over the supporting ground to achieve more uniform ground pressure for improved traction and mobility on adverse soils.

track link. Each of the rigid units that are flexibly connected to form a jointed type track. On double-pin tracks it consists of two track blocks assembled with two track pins. On single-pin tracks it is the track body, with bushings but without pins. Cf. **track body** and **track shoe assembly.**

track pin. A pin that fits into track links to form the hinge about which flexing occurs in the jointed type track.

track shoe assembly. The assembly consisting of a track link, pins, end connectors, center guides, and bushings necessary to provide one complete unit of a jointed type track. Cf. **track body** and **track unit.**

tractive effort. *See*: **gross tractive effort.**

tractor. A track-laying vehicle designed to tow by means of a pintle hook or fifth-wheel coupling device.

tractor, cargo. Military track-laying vehicle designed to carry cargo, as well as to perform as a tractor.

trafficability. The capacity of a soil to withstand the traffic of military vehicles.

trailer. A wheeled or tracked vehicle, nonpowered, with all or most of its weight supported by its own integral wheels or tracks, designed to carry materials, supplies, or equipment and to be towed by a self-propelled motor vehicle. Excludes **semitrailer.**

transfer function. The function which describes the dynamic characteristics of a vehicle for use in statistical analysis of dynamic response.

transport vehicle. Vehicle primarily intended for personnel and cargo carrying, excluding combat vehicle.

transportability. The capability of an item of military equipment to be transported efficiently and effectively via railways, highways, waterways, oceans, and airways, either by carrier, by being towed, or by self-propulsion.

transported soil. A soil which was moved by the forces of nature away from its origin and redeposited.

tread. That part of a wheel, tire, or track shoe that makes contact with the ground as it moves. Also, the transverse distance between the central points of contact with the ground of the wheels, or tracks, of a vehicle.

triaxial shear test. A shear test in which a cylindrical specimen encased in an impervious membrane is subjected to a confining pressure and then loaded axially to failure.

truck, automotive. A self-propelled wheeled vehicle designed primarily to transport supplies and/or equipment and which may be used to tow trailers or other mobile equipment. Excludes **truck-tractor.**

truck-tractor. A short wheelbase wheeled vehicle designed to tow and partially support a semi-trailer through a fifth-wheel coupling device.

unit weight. Weight per unit volume.

unsprung weight. Unsprung weight is the total weight of all of the vehicle components that are not supported by the vehicle spring system. This includes such items as wheels, tracks, axles, road wheels, etc.

vehicle cone index. The index number assigned to a given vehicle that indicates the minimum soil strength, in terms of the **rating cone index,** that is required to permit 50 passes of the vehicle.

velocity shock. A particular type of shock motion characterized by a sudden (assumed instantaneous) velocity change.

vibration. The oscillation of a mechanical element about a given reference position. *See*: **forced vibration** and free **vibration.**

viscosity. In a liquid, the property of internal resistance caused by molecular attraction that makes the liquid resist flow.

viscosity index. A number given to a certain lubricating oil to indicate its performance, particularly as to change of viscosity with temperature variation—as compared with the average of two groups of test oils.

viscous damping. The dissipation of energy that occurs when a particle in a vibrating system is resisted by a force that has a magnitude proportional to the magnitude of the velocity of the particle and a direction opposite to the direction of motion of the particle.

void. Space in a soil mass not occupied by solid mineral matter.

void ratio. Ratio of the volume of void space to the volume of solid mineral matter in a soil mass.

volute spring. A form of conical compression spring usually made of flat spring stock and wound in a spiral helix with the successive coils telescoping into each other. It is characterized by its compactness, variable spring rate, and high friction damping. It is used as the spring element in certain bogie suspensions of tracked vehicles and as bottoming springs on vehicles with soft suspensions.

wagon steering. Steering of a vehicle consisting of one or more units by a single pivot system with the pivot point located over the front axle.

wheelbase. The distance between the centers of the front and rear wheels of a wheeled vehicle. In vehicles equipped with bogie axles, it is the distance from the center of the front wheels to a vertical line that is equidistant from the centers of the two bogie axles.

wheel dance (wheel hop). The vertical vibration of the unsprung mass of a suspension system occurring at the natural frequency of the spring-mass system, consisting of the primary spring elements, the unsprung mass, and the spring characteristics of the tires. The spring rate of the tire is the dominant elastic factor associated with wheel dance. Wheel dance is the principal source of secondary disturbances and vibrations of the sprung mass.

wheel wobble. *See*: **shimmy.**

yaw. The angular displacement of a vehicle about an axis parallel to its normal (vertical) **axis.** Cf. **pitch** and **roll.**

APPENDIX I

DESIGN DATA

A-1 INTRODUCTION

This appendix provides a practical approach to the determination of loads imposed on the suspension systems of vehicles—terrain, gun recoil, and handling loads—and discusses the application of safety factors associated with these loads.

A-2 LOAD DETERMINATION

Perhaps the most difficult problem area with which the military vehicle designer must cope is the one dealing with the determination of the magnitudes of the loads which act upon the vehicle—the so-called applied loads (limit load) which the various elements of the vehicle, and the vehicle itself, must withstand. These loads arise from the dynamics of motion as the vehicle traverses rough terrain (road loads), the recoil forces of large weapons mounted on the vehicle, forces arising from handling of the vehicle (hoisting, transporting, etc.), and forces due to the vehicle's weight and location of its center of gravity. Of these loads, the most significant to the suspension designer are the road loads. At the same time, these are the most difficult to evaluate, since they arise from an almost hopelessly complex dynamic system.

Nevertheless, a semi-empirical system does exist for approximating these loads. It is based upon data collected from actual vehicles by means of oil pressure readings, strain gage and accelerometer measurements taken while the vehicles were in motion, from the study of broken parts and applying stress calculations to determine the loads that obviously existed when the parts failed, and from studies based upon the performances of similar vehicles under tactical or simulated tactical conditions. This work was conducted by the U.S. Army Tank-Automotive Center (USATAC) and is still continuing. Tentative results have been compiled in a pamphlet entitled *Structural Design Criteria for Military Vehicles* by C. H. Winfree of USATAC. It is not a rigorous method, nor is it to be considered undisputably accurate. It is a simple procedure for approximating the ultimate values of the forces that act upon a military vehicle during typical tactical operations. The results have been shown to be reasonably accurate. The portion of this procedure that applied to estimating the loads acting upon suspension components is given here for the designer's convenience.

Figure A-1 is a simple monograph that is used to determine a basic load factor n for any vehicle under consideration. It shows seven vehicle classifications rated according to vehicle types (trailers, wheeled vehicles, tracked vehicles) and in accordance with their intended missions or operating environments (logistic vehicles, combat vehicles, vehicles limited to paved roads, etc.). The horizontal scale represents the gross vehicle weights in thousands of pounds while the vertical scale represents the basic load factors n.

The following procedure is used in estimating the principal road loads that are experienced by the main suspension components.

(a) Determine the vehicle classification and operations category, and select the appropriate curve on Figure A-1. The severity of the category increases as we go up on the chart. Since the classification curves are approximately parallel lines, intermediate categories can be created between the existing lines to suit special conditions. For example, if the vehicle is to be a tracked logistic type but is required to withstand greater abuse than is normal for a vehicle of this type and yet not quite as severe as that experienced by combat vehicles, a special classification line can be drawn between the lines representing these two types of vehicles. This is the signifi-

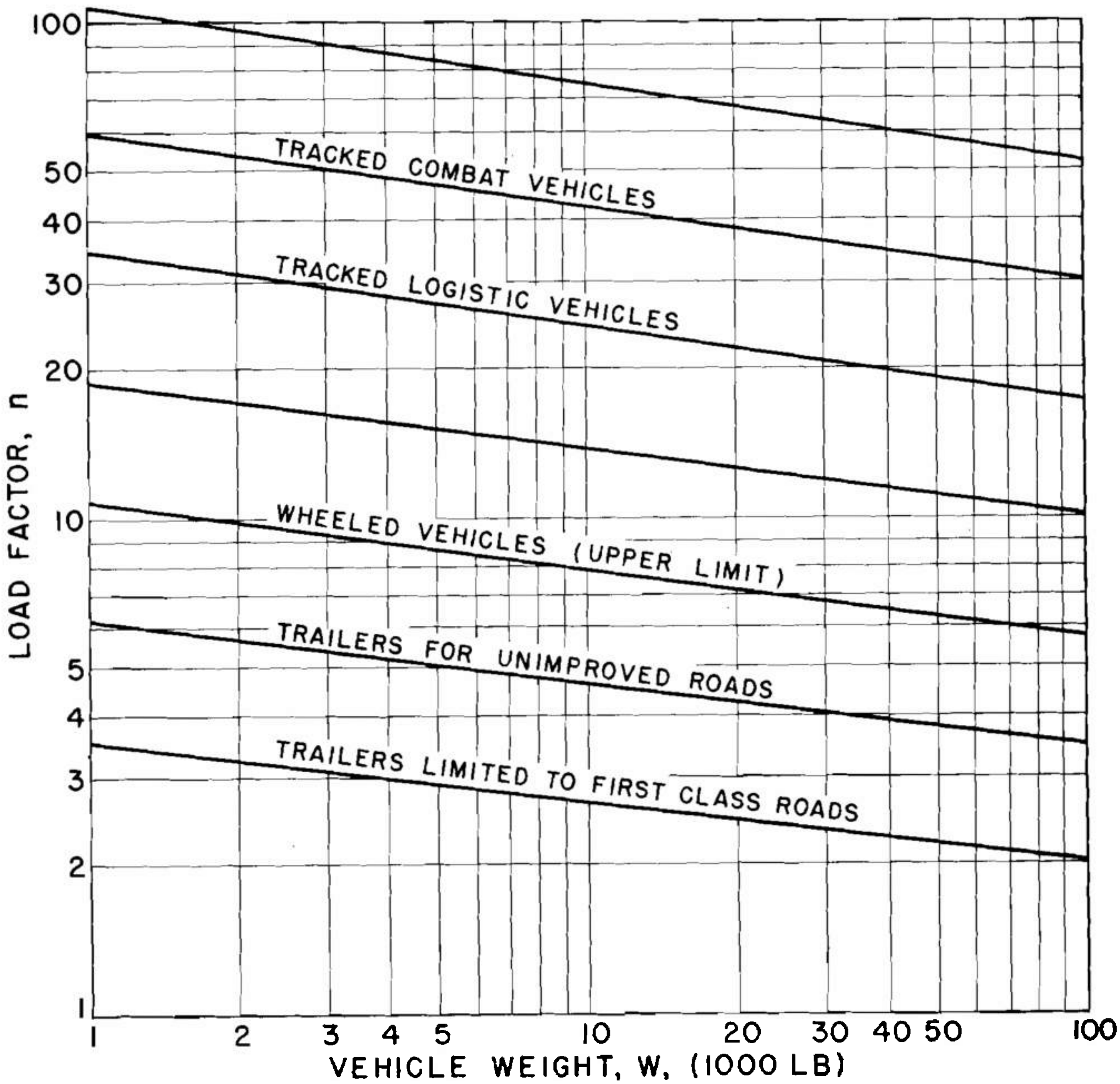

Figure A-1. Load Factors n for Estimating Road Loads

cance of the unlabled curves on the chart.

(b) Having selected an appropriate classification curve, determine the load factor n corresponding to the intersection of the selected curve with the estimated gross weight W of the vehicle.

For typical track-laying vehicles that have rigid hulls such as tanks, self-propelled artillery, and armored personnel carriers, this factor represents the maximum acceleration, in g's, that is experienced by the front end of the vehicle during severe tactical operations. For design purposes, the acceleration at successive stations along the vehicle length is assumed to decrease at a uniform rate (straight line function) to a value of $(n/2)g$ at the vehicle center of gravity (C.G.) and to remain constant at $(n/2)g$ at stations behind the C.G. When n has a value less than 40, however, the acceleration at the C.G. is assumed to have a flat value of $20g$ and to remain constant at this value for all stations beyond the C.G. When applied to trailers, the n factor represents a general load factor.

Design loads acting on the suspension elements of a tracked vehicle can be approximated by using the load factor n as a multiplier in the simple equations that follow. It should be understood, however, that these loads do not occur simultaneously but are the maximum values that occur under dynamic conditions at the point indicated.

(c) Horizontal and vertical design loads on forward sprocket, considered to act at centerline of track and directed rearward and upward, respectively,

$$= 0.15\, nW, \text{lb}$$

(d) Vertical load on No. 1 road wheel, considered to act upward on centerline of road wheel,

$$= 0.1\, nW, \text{lb}$$

(e) Vertical load on No. 2 and all successive road wheels,

$$= 0.05\, nW, \text{lb}$$

(f) Vertical and horizontal loads on rear sprocket, considered to act upward and forward, respectively,

$$= 0.07\ nW$$

Loads resulting from track tension must also be considered.

(g) Transverse load acting horizontally against bottom edge of road wheel, per road wheel

$$= \frac{W}{2}, \text{lb}$$

Example: Assume that a tracked combat vehicle is under consideration which is expected to have five road wheels per side and a gross weight of 25,000 lb.

From Figure A-1, $n = 37$

Loads on forward sprocket

$$= 0.15 \times 37 \times 25{,}000 = 138{,}750 \text{ lb}$$

Load on No. 1 road wheel

$$= 0.1 \times 37 \times 25{,}000 = 92{,}500 \text{ lb}$$

Loads on No. 2, 3, 4 and 5 road wheels

$$= 0.05 \times 37 \times 25{,}000 = 46{,}250 \text{ lb}$$

Loads on rear sprocket

$$= 0.07 \times 37 \times 25{,}000$$
$$= 65{,}000 \text{ lb}$$

Transverse load on each road wheel

$$= \frac{W}{2} = 12{,}500 \text{ lb}$$

A-3 GUN RECOIL LOADS

Gun recoil loads are obtained by established methods of calculation and confirmed by test firing. Calculations can be made according to AMCP 706-342 *Recoil Systems* or Detroit Arsenal Publication *Design of Hydraulic Orifice For Recoil Control.* Both methods give results having a reasonable degree of accuracy. Gun recoil loads are known more accurately than any other loads in combat vehicles. One question frequently asked is what effect short peaks or spikes in hydraulic recoil systems have upon hull and spade structures. Spikes of less than 3-milliseconds duration have no appreciable effect upon vehicles structure or spade load.

Recoil loads are transmitted through the gun mount, vehicle structure, and spade into the earth. Reactions from the earth may be distributed uniformly over the spade or concentrated due to large rocks. Investigations should be conducted for different angles of elevation and azimuth considering ledge rock in the earth. Ordinary dynamic and static equations provide satisfactory answers to spade loads. Additional discussion of this subject is given in Chapter 2 (paragraph 2-12.2.1) of this handbook.

A-4 HANDLING LOADS

Handling loads are determined by methods of transportation and a minimum strength factor dependent upon the abuse to which the vehicle is subjected. Small parts which a man can step on should withstand a load of 300 lb.

With the exception of air transportation requirements, design loads for transportation systems are not well defined by any military specification.

The following shock specification may be of some help.

(a) For rail transportation.

Vertical: $\pm 2g$

Longitudinal: $\pm 12g$

Transverse: ± $3g$

(b) For marine shipment a load factor of $2g$ is adequate.

(c) For air transportation.
Consult Spec. MIL-A-8421, *General Specification for Air Transportability Requirements.*

(d) For parachute delivery.
Loads imposed upon equipment during air drop depend upon the vehicle attitude at time of contact and shock-absorbing equipment incorporated in the delivery platform. Consult MIL-STD-669, *Air Delivery Loading Environment and Related Requirements for Military Materiel*, MIL-STD-814 *Requirements for Tiedown, Suspension, and Extraction Provisions on Military Materiel for Air Delivery*, and *Handbook of Instructions for Ground Equipment Designers* (HIGED) paragraph B-6-10.

A-5 SAFETY FACTORS

Concurrent with the establishment of basic loads and load distribution are the problems of selecting and applying factors of safety, consideration of stress concentrations, detailed environmental and life requirements, and the balancing of all these factors against "optimized" weight.

The more exact the knowledge of the basic loads, the more it is possible to design structures with minimum safety factors compatible with reasonable reliability. Design analysis should indicate the accuracy to which load factors are known and "factors of ignorance" should not be confused with safety factors.

Safety factors are used for the following reasons

(a) To provide a standard difference between applied load and structural failure. This is the classical meaning generally expressed as

Applied Load + Safety Factor = Design Load

(b) To provide a uniform criterion for failure in structures where deformation is unimportant.

(c) To account for unknowns in loading, analysis, material characteristics, and fatigue. These are truly "factors of ignorance" which are generally called safety factors.

Safety factors should not be confused by being indiscriminately placed on both the load and on the allowable stress level. Extreme care must be exercised to avoid multiplication of safety factors. Investigation has revealed many cases where safety factors were 5 to 10 times the quoted value. Common methods of introducing safety factors while determining strength of structural components are

(a) Working to yield stress in cases where the ultimate stress is the critical consideration.
(b) Failure to account for plasticity.
(c) Use of curved beam equations when they are not applicable.
(d) Incorrect use of allowable stress.
(e) Overlooking a primary loadpath.
(f) Undue simplication of analysis.
(g) And the fatal remark of the boss "it looks skimpy, better beef it up."

To avoid multiple safety factors, select a value before starting and apply it to the load; do not enter safety factors during stress calculations.

The relation comparing the allowable load with the design load is termed the "margin of safety" and is defined as

$$MS = \frac{\text{allowable load}}{\text{design load}} - 1$$

or

$$MS = \frac{\text{allowable stress}}{\text{design stress}} - 1.$$

The allowable load is usually determined by the shape of the part and the physical properties of the material. It must not be confused with design load. In case of a redundant structure, where failure of an individual element would result in the load being carried by other members, a negative margin of safety may be acceptable. In a well-designed structure, the margin of safety should be zero or have a slight positive value. The positive margin of safety is that overdesigned portion of a structure which cannot practically be eliminated.

APPENDIX II

SUSPENSION AND TRACK DATA

The following tables of suspension and track data are published to provide background information for those concerned with concept, engineering, and design systems and tracks. They apply to the various types of suspensions and tracks used on military track-laying vehicles. The tables are divided into two parts.

The tables of suspension data (Tables A-1 through A-10) provide information pertinent to suspension system design, type, and characteristics.

The tables of track data (Tables A-11 through A-20) provide information on track assemblies that have been or are currently being used on military vehicles. These tables include data on vehicle usage, track model, description, and design information.

TABLE A-1
SUSPENSION DATA

	Half Track	Light Tanks						
VEHICLE	M2, Car w/winch	T7E2	T9E1	T24E1	T37	T49, 90mm Gun	T71, 76mm Gun	T65, Flame Thrower
SUSPENSION TYPE	Front drive w/support roller and rear spring idler.	Front drive w/support rollers and trailing idler.	Front drive w/support rollers and trailing idler.	Front drive w/support rollers and rear compensating idler.	Rear drive w/support rollers, front comp. and track tensioning idler.	Rear drive w/support rollers and front compensating idler.	Front drive and trailing idler. No support rollers.	Front compensating sprocket w/support rollers and rear idler.
INSTALLATION DRAWING NUMBER	D6358730		31-243	8358000	D7715821			
VEHICLE WEIGHT (lb): TARE / GROSS	15,300 19,800	48,059 51,058	14,830 16,430	34,000 39,500	43,400 50,000	46,650 58,200	33,150 37,400	34,200 38,000
ROAD WHEELS: QUANTITY	8 dual	8 single	8 single	10 dual	10 dual	10 dual	8 dual	10 dual
ROAD WHEELS: TYPE	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber
ROAD WHEELS: SIZE (in.)	12 x 4-1/2	22 x 9	15 x 6	25-1/2 x 4-1/2	25-1/2 x 4-1/2	25-1/2 x 4-1/2	34 x 5	26 x 4-1/2
VEHICLE WIDTH ACROSS TRACKS (in.)	76-1/16	109	81-1/2	112	122-3/4	123-1/2	109-3/4	106-5/8
TRACK GROUND-CONTACT LENGTH (in.)	46-3/4	124	102	120	122	127	114	128-1/2
L/T RATIO	46-3/4/63-13/16	1.33	1.45	1.25	1.20	1.24	1.19	1.36
VEHICLE SPRINGING MEDIA	Vertical volute	Vertical volute	Vertical volute	Torsion bar	Torsion bar	Torsion bar	Torsion bar	Torsion bar
SHOCK ABSORBER: QUANTITY	None	None	None	8 (wheels 1,2,4,5)	8 (wheels 1,2,4,5)	6 (wheels 1,2,5)	4 (wheels 1,4)	10 (wheels 1 (2 ea), 2,4,5)
SHOCK ABSORBER: TYPE	Does not apply.	Does not apply.	Does not apply.	Direct-acting	Direct-acting	Direct-acting	Direct-acting	Double-acting cam and lever
SHOCK ABSORBER: CONTROL	Does not apply.	Does not apply.	Does not apply.	5636964	7716544	50 cpm, 3720 rebound & 5360 comp.; 100 cpm, 3350 rebound & 4650 comp.		C121535
SPROCKET: PART NUMBER	D6546406	D6540694	C106922	D5676210	D7527958	D7962691		C5621431
SPROCKET: TYPE	External tooth forging	External tooth casting	External tooth steel plate	External tooth steel plate	External tooth steel plate	External tooth steel plate	External tooth steel plate	External tooth plate welded to disc
SPROCKET: NO. TEETH	18	13	22	13	12	12	12	31
SPROCKET: PITCH DIAM. (in.)	22.916	25.038	22.945	22.979	23.182	23.422	21.492	25.234
SPROCKET: ENGAGEMENT	Center	End connector	Track shoe	Track shoe	Track shoe	Track shoe	Track shoe	Track pin
STATIC WHEEL LOADING (lb)		5106	1643	3603	5000	5320	4680	3800
SUSPENSION % VEHICLE WEIGHT				25.6	22.6	17.7		
SPRING RATE	5000 lb/in.			3338 in.-lb/deg	3838 in.-lb/deg	3220 in.-lb/deg		3338 in.-lb/deg
CENTER OF GRAVITY (in.): VERTICAL / HORIZONTAL			27.875 63 from c/l of sprocket		42 93.6 from c/l of sprocket		38.8 8.7 from c/l turret	30.5 78.5 from c/l of sprocket
NATURAL FREQUENCY (cpm): SUSPENSION / VEHICLE					305 98			
TRACK ANGLE (deg): APPROACH / DEPARTURE	45 45	30 Does not apply.	32 Does not apply.	35 30	30 27	37 30	41 Does not apply.	40 43
TRACK ENVELOPE (in.)	232	412-1/2	318	412-1/2	450	450	357-1/2	422-1/2

TABLE A-2
SUSPENSION DATA

		Medium Tanks							Heavy Tanks
VEHICLE		M4A3E2, 75mm Gun	M4A3E8	M48E1, 90mm Gun (T48)	T26	T33, Flame Thrower	T69, 90mm Gun	T95E2, 90mm Gun	M6, 3-in. Gun
SUSPENSION TYPE		Front drive w/support rollers and rear idler.	Front drive w/support rollers and rear idler.	Rear drive w/support rollers and front compensating idler.	Rear drive w/support rollers and front compensating idler.	Front drive w/support rollers and rear idler.	Rear drive w/support rollers and front compensating idler.	Rear drive and front idler. No support rollers.	Rear drive w/support rollers, adjustable front idler and fixed auxiliary idler.
INSTALLATION DRAWING NUMBER		21-235					K-7537165	DTA-25280	D46406
VEHICLE WEIGHT (lb): TARE GROSS		77,532 84,000	67,000 76,000	95,351 103,265	89,000 96,000	93,800 97,000	72,000 76,000	 79,400	119,830 129,338
ROAD WHEELS	QUANTITY	12 single	12 dual	12 dual	12 dual	12 dual	10 dual	10 dual	16 dual
ROAD WHEELS	TYPE	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber
ROAD WHEELS	SIZE (in.)	20 x 9	20 x 6-1/4	26 x 6	26 x 6	20 x 6-1/4	26 x 6	33 x 4	18 x 6
VEHICLE WIDTH ACROSS TRACKS (in.)		105-1/6	106	143	134	106	135	124	119-5/6
TRACK GROUND-CONTACT LENGTH (in.)		147	147	157-1/2	154	147	127	165.4	200
L/T RATIO		1.76	1.66	1.37	1.40	1.77	1.14	1.56	2.12
VEHICLE SPRINGING MEDIA		Vertical volute	Horizontal volute	Torsion bar	Torsion bar	Horizontal volute	Torsion bar	Torsion bar	Horizontal volute
SHOCK ABSORBER	QUANTITY	None	6 (Between suspension arms)	6 (wheels 1, 2, 6)	8 (wheels 1, 2, 5, 6)	6 (Between suspension arms)	6 (wheels 1, 2, 6)	4 (wheels 1, 6)	None
SHOCK ABSORBER	TYPE	Does not apply.	Double-acting	Direct-acting	Direct-acting	Double-acting	Direct-acting	Direct-acting	Does not apply.
SHOCK ABSORBER	CONTROL	Does not apply.	D6580067	7953032	5200 rebound 1925 compression	D6580067	5200 rebound 1925 compression	3000 rebound 5000 compression	Does not apply.
SPROCKET	PART NUMBER	D6547366	D6547366	K7364145	D6577076	D6547366	D7627958	DTA-15917	Center D46311 Outer D46310
SPROCKET	TYPE	External tooth steel plate	External tooth steel plate	External tooth-1-3/6 in. steel plate	External tooth forging	External tooth steel plate	External tooth steel plate	External tooth steel plate	External tooth casting
SPROCKET	NO. TEETH	13	13	11	13	13	12	10	14
SPROCKET	PITCH DIAM. (in.)	25.038	25.038	24.504	25.066	25.038	23.182	22.19	D46311-26.430 D46310-26.736
SPROCKET	ENGAGEMENT	End connector	End connector	End connector	Track shoe	End connector	Track shoe		Center and ends
STATIC WHEEL LOADING (lb)		7000	6333	8606	9717	8080	7200	7940	8084
SUSPENSION % VEHICLE WEIGHT		21	22.4	23.6	23.3	17.6		21.5	
SPRING RATE				7350 in.-lb/deg	7350in.-lb/deg		7350 in.-lb/deg	7350 in.-lb/deg	
CENTER OF GRAVITY (in.): VERTICAL HORIZONTAL				48.6 110.9 forward c/l of sprocket	77-13/16 from c/l sprocket		79-1/2 5.42 rear c/l turret		
NATURAL FREQUENCY (cpm): SUSPENSION VEHICLE						45-9/16 95-5/16 from c/l of sprocket			 86
TRACK ANGLE (deg): APPROACH DEPARTURE		32 30	32 30	36 37	40 30	32 30	36 29	24 18	33 35
TRACK ENVELOPE (in.)		474	474	546.062	492	474	456	555	600

TABLE A-3
SUSPENSION DATA

	Heavy Tanks			Self-Propelled Weapons				
VEHICLE	T28 (T95 GMC)	T32	T34	M19, M19A1, 40mm Twin	M36B1, 90mm Gun	M36B2, 90mm Gun	M43 (T89), 8-in. Howitzer	T27, 81mm Mortar Carriage
SUSPENSION TYPE	Rear drive w/support rollers and front idler.	Rear drive w/support rollers and front compensating idler.	Rear drive w/support rollers and front compensating idler.	Front drive w/support rollers and rear compensating idler.	Front drive w/support rollers and rear idler.	Front drive w/support rollers and rear idler.	Front drive w/support rollers and rear idler.	Front drive w/support rollers and trailing idler.
INSTALLATION DRAWING NUMBER	31-348	7073647	7379630	D7051460	8358149	8358072	7067329	
VEHICLE WEIGHT (lb): TARE / GROSS	180,600 190,000	120,000	132,000 141,000	83,775 38,500	68,000	66,000	74,000 83,000	28,100 31,500
ROAD WHEELS: QUANTITY	32 dual	14 dual	16 dual	10 dual	12 single	12 single	12 dual	8 single
ROAD WHEELS: TYPE	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber
ROAD WHEELS: SIZE (in.)	20-1/2 x 4-1/2	26 x 6	26 x 6	25-1/2 x 4-1/2	20 x 9	20 x 9	20 x 6-1/4	20 x 6
VEHICLE WIDTH ACROSS TRACKS (in.)	167	124	133	112	99-9/16	114-11/16	122	85
TRACK GROUND-CONTACT LENGTH (in.)	208	192	204-5/8	124-1/8	147	147	164	120
L/T RATIO	1.67	1.90	1.86	1.30	1.77	1.62	1.65	1.64
VEHICLE SPRINGING MEDIA	Double horizontal volute	Torsion bar	Torsion bar	Torsion bar	Vertical volute	Vertical volute	Horizontal volute	Vertical volute
SHOCK ABSORBER: QUANTITY	None	8 (wheels 1,2,6,7)	10 (wheels 1,2,3,7,8)	8 (wheels 1,2,4,5)	None	None	6 (Between suspension arms)	None
SHOCK ABSORBER: TYPE	Does not apply.	Direct-acting	Direct-acting	Double-acting	Does not apply.	Does not apply.	Double-acting	Does not apply.
SHOCK ABSORBER: CONTROL	Does not apply.		5200 rebound 1925 compression 4600 rebound 3000 compression	C136964	Does not apply.	Does not apply.	6580067	Does not apply.
SPROCKET: PART NUMBER	D7029276	7020098	7020098	D6576210	D6547366	D6578555	D6547366	D6559888
SPROCKET: TYPE	External tooth steel plate	External tooth steel plate	External tooth steel plate	External teeth steel plate	External teeth steel plate	External teeth steel plate	External tooth steel plate	External tooth steel plate
SPROCKET: NO TEETH	14	16	16	13	13	13	13	13
SPROCKET: PITCH DIAM. (in.)	26.964	28.890	28.890	22.979	25.038	25.038	25.038	22.876
SPROCKET: ENGAGEMENT	End connector	End connector	End connector	Track shoe	End connector	End connector	End connector	End connector
STATIC WHEEL LOADING (lb)	5937	8580	8813	3850	5666	5500	6917	3150
SUSPENSION % VEHICLE WEIGHT	23.6		21.2		22.8	23.5		
SPRING RATE		7350 in.-lb/deg	7350 in.-lb/deg	2436 in.-lb/deg				
CENTER OF GRAVITY (in.): VERTICAL / HORIZONTAL	130.25 from c/l of sprocket		56					
NATURAL FREQUENCY (cpm): SUSPENSION / VEHICLE			83					
TRACK ANGLE (deg): APPROACH / DEPARTURE	36 31	39 32	39 32	29 24	32 30	32 30	32 30	30 Does not apply.
TRACK ENVELOPE (in.)	612	612	612	440	474	474	516	363

TABLE A-4
SUSPENSION DATA

		Self-Propelled Weapons							
VEHICLE		T64, 4.2-in. Mortar Carriage	T64, 155mm Howitzer	T64E1, 155mm Howitzer	T84, 8-in. Howitzer	T88, 105mm Howitzer	T90, 155mm Mortar Carriage	T92, 240mm Howitzer	T93, 8-in. Gun
SUSPENSION TYPE		Front drive w/support rollers and rear compensating idler.	Front drive w/support rollers and rear idler.	Front drive w/support rollers and rear compensating idler.	Front drive w/support rollers and rear compensating idler.	Front compensating sprocket w/support rollers and rear idler.	Front drive w/support rollers.	Front drive w/support rollers and rear compensating idler.	Front drive w/support rollers and rear compensating idler.
INSTALLATION DRAWING NUMBER				D7051460				7321699	
VEHICLE WEIGHT (lb): TARE GROSS		41,000 43,500	36,190 40,000	38,500 41,500	75,580 82,600	34,000 40,000	64,000 68,000	120,600 125,000	125,500 130,000
ROAD WHEELS	QUANTITY	10 dual	12 single	10 dual	12 dual	10 dual	12 single	14 dual	14 dual
	TYPE	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber
	SIZE (in.)	25-1/2 x 4-1/2	20 x 6	25-1/2 x 4-1/2	26 x 6	26 x 4-1/2	20 x 9	26 x 6	26 x 6
VEHICLE WIDTH ACROSS TRACKS (in.)		108	105	112	134	108-1/2	100-1/8	133	133
TRACK GROUND-CONTACT LENGTH (in.)		115-3/4	136	124-1/8	150	116	147	182	182
L/T RATIO		1.33	1.46	1.30	1.36	1.23	1.76	1.66	1.65
VEHICLE SPRINGING MEDIA		Torsion bar	Vertical volute	Torsion bar	Torsion bar	Torsion bar	Vertical volute	Torsion bar	Torsion bar
SHOCK ABSORBER	QUANTITY	4 (wheels 1,6)	None	8 (wheels 1,2,4,5)	8 (wheels 1,2,5,6)	10 (wheels 1 (2 ea) 2,4,5)	None	8 (wheels 1,2,6,7)	8 (wheels 1,2,6,7)
	TYPE	Direct-acting	Does not apply.	Direct-acting	Direct-acting	Double-acting cam and lever	Does not apply.	Direct-acting	Double-acting
	CONTROL	7754637	Does not apply.	C136964	5200 rebound 1925 compression	C121535	Does not apply.	4500 rebound 3000 compression	4500 rebound 3000 compression
SPROCKET	PART NUMBER	D7952097		D6576210	D7058098	C5621431	D6547366	7321569	7321569
	TYPE	External tooth steel plate	External tooth steel plate	External tooth steel plate	External tooth steel plate	External tooth plate welded to disc	External tooth steel plate	External tooth steel plate	External teeth steel plate
	NO. TEETH	12	13	13	13	31	13	13	13
	PITCH DIAM. (in.)	23.422	20.380	22.979	25.068	25.234	25.038	25.038	25.038
	ENGAGEMENT	Track shoe	End connector	Track shoe	Track shoe	Track pin	End connector	End connector	End connector
STATIC WHEEL LOADING (lb)		4350	3333	4156	6883	4000	5666	8929	9286
SUSPENSION % VEHICLE WEIGHT				26.1					
SPRING RATE				3338 in.-lb/deg				7350 in.-lb/deg	7350 in.-lb/deg
CENTER OF GRAVITY (in.): VERTICAL HORIZONTAL		42-6/16 81-3/8 from c/l of sprocket	34 90 from c/l sprocket		45-1/2 103 from c/l of sprocket	38.6 4.7 from c/l center wheel		50-1/2 128-3/4 from c/l of sprocket	50-1/2 128-3/4 from c/l sprocket
NATURAL FREQUENCY (cpm): SUSPENSION VEHICLE									
TRACK ANGLE (deg): APPROACH DEPARTURE		45 30	25 23	29 24	28 29	29 25		37 36	37 36
TRACK ENVELOPE (in.)		420	434-1/2	440	498	422-1/2	474	564	684

TABLE A-5
SUSPENSION DATA

	Self-Propelled Weapons							
VEHICLE	T94, 250mm Mortar Carriage	T95, 105mm Gun (T28 Hvy Tank)	T100, Multiple Cal. .60 Gun	T101 (M56), 90mm Gun	T106, 105mm Rifle	T108 (M55), 8 in.-How.	T194 (M44), 155mm How.	T195, 105mm How.
SUSPENSION TYPE	Front drive w/support rollers.	Rear drive w/support rollers and front idlers.	Rear drive w/support rollers, front comp. idler and rear tensioning idler.	Front drive and rear compensating idler. No support rollers.	Rear drive w/support rollers and front ilder.	Front drive w/support rollers and trailing idler.	Front drive w/support rollers and trailing idler.	Front drive and trailing idler. No support rollers.
INSTALLATION DRAWING NUMBER		31-348				7393200	8385027	DTA-33000
VEHICLE WEIGHT (lb): TARE GROSS	76,000 80,000	180,600 190,000	38,750 44,000	12,500 15,750	4,731	88,000 96,000	55,000 61,000	35,500
ROAD WHEELS — QUANTITY	12 dual	32 dual	10 dual	8 single	16 dual	14 dual w/idlers	10 dual, w/2 idlers	14 dual inc/idlers
ROAD WHEELS — TYPE	Solid rubber	Solid rubber	Solid rubber	Smooth pneumatic	Solid rubber	Solid rubber	Solid rubber	Solid rubber
ROAD WHEELS — SIZE (in.)	20 x 6-1/4	20-1/2 x 4-1/2	25-1/2 x 4-1/2	7.50 x 12	8 x 1-1/4	26 x 6	25-1/2 x 4-1/2 idler-28 x 4-1/2	22 x 2
VEHICLE WIDTH ACROSS TRACKS (in.)	124	167	123-1/2	98	61	133	123-3/4	124
TRACK GROUND-CONTACT LENGTH (in.)	168	208	127	93.867	78	184	149-3/8	156
L/T RATIO	1.87	1.67	1.24	1.20	1.73	1.67	1.36	1.43
VEHICLE SPRINGING MEDIA	Horizontal volute	Double horizontal volute	Torsion bar	Torsion bar and tube	Leaf spring	Torsion bar	Torsion bar	Torsion bar
SHOCK ABSORBER — QUANTITY	6 (Between suspension arms)	None	6 (wheels 1,2,5)	4 (wheels 1,4)	None	4 (wheels 1,2)	8 (wheels 1,2,5 and idler)	4 (wheels 1,7)
SHOCK ABSORBER — TYPE	Hydraulic telescopic	Does not apply.	Direct-acting	Cam and lever	Does not apply.	Direct-acting	Direct-acting	Direct-acting
SHOCK ABSORBER — CONTROL	D6580067	Does not apply.	50 cpm, 3720 rebound & 5360 comp.; 100 cpm, 3350 rebound & 4650 comp.		Does not apply.	4500 rebound 3000 compression	5516398 7404209	DTA-F-33001
SPROCKET — PART NUMBER	D6547366	D7029276	D7527958	D7980550	C7324249	D6547366	D7982691	DTA-32988
SPROCKET — TYPE	External tooth steel plate	External teeth steel plate	External teeth steel plate	Internal tooth rubber-covered	External tooth plate riveted to disc	External tooth steel plate	External tooth steel plate	External tooth steel plate
SPROCKET — NO. TEETH	13	14	12	12	9	13	13	9
SPROCKET — PITCH DIAM. (in.)	25.038	26.964	23.182	19.099	12.892	25.038	23.422	17.550
SPROCKET — ENGAGEMENT	End connector	End connector	Track shoe	Internal duplex guide	Lugs on shoes	End connector	Track shoe	Track shoe
STATIC WHEEL LOADING (lb)	6666	5937	4400	1962	296	6857	5083	2535
SUSPENSION % VEHICLE WEIGHT		23.6	21.4					18.2
SPRING RATE						7350 in.-lb/deg		2435 in.-lb/deg
CENTER OF GRAVITY (in.): VERTICAL HORIZONTAL		86-1/4 from c/l		30 48.125 from c/l front wheel	24.4 47.9 from c/l sprocket	61 102 from c/l cross drive	57-1/2 100-1/4 from c/l of sprocket	44.9 101.88 from rear plate of vehicle
NATURAL FREQUENCY (cpm): SUSPENSION VEHICLE								
TRACK ANGLE (deg): APPROACH DEPARTURE		36 31	37 30	38 13	28 30	38 Does not apply.	37 Does not apply.	27 Does not apply.
TRACK ENVELOPE (in.)	503-1/2	612	450	352	274	L.-534 Rt.-540	L.-444 Rt.-450	L.-426 Rt.-432

TABLE A-6
SUSPENSION DATA

	Self-Propelled Weapons	Carriers						
VEHICLE	T235, 155mm Gun	M28, Cargo Carrier (T15)	M29C, Amphib. Cargo Carrier	M30, Cargo Carrier	M39 (T41E1), Armored Utility	M44E1, Armored Utility	M59, Armored Infantry	T22E1, T23E1, Cargo Carrier
SUSPENSION TYPE	Front drive and trailing idler. No support rollers.	Front drive w/support rollers and rear idler.	Rear drive w/support rollers and front idler.	Front drive w/support rollers and rear idler.	Front compensating sprocket w/support rollers and rear idler.	Front drive w/support rollers and rear compensating idler.	Front drive w/support rollers and rear compensating idler.	Front drive w/support rollers.
INSTALLATION DRAWING NUMBER	PCF-6263-496		8358373			7755478	K8340065	
VEHICLE WEIGHT (lb): TARE / GROSS	58,300	3,390 4,650	4,778 5,971	39,500 47,000	33,500 35,500	42,210 51,000	38,700 41,800	31,000 40,500
ROAD WHEELS — QUANTITY	10 dual inc/idlers	8 spaced duals	16 dual	12 single	10 dual	12 dual	10 dual	10 dual
ROAD WHEELS — TYPE	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber
ROAD WHEELS — SIZE (in.)	32 x 4	8 x 1-1/4	8 x 1-1/4	20 x 9	26 x 4-1/2	25-1/2 x 4-1/2	25-1/2 x 4-1/2	25 x 4-1/2
VEHICLE WIDTH ACROSS TRACKS (in.)	124	60	65	99	109-1/8	117	124	112
TRACK GROUND-CONTACT LENGTH (in.)	144	62-1/4	78	147	116-1/2	154-5/8	121-1/4	124
L/T RATIO	1.36	1.49	1.74	1.49	1.23	1.61	1.17	1.29
VEHICLE SPRINGING MEDIA	Torsion bar	Leaf spring	Leaf spring	Vertical volute	Torsion bar	Torsion bar	Torsion bar	Torsion bar
SHOCK ABSORBER — QUANTITY	10 (all wheels)	None	None	None	10 (wheels 1 (2 ea.) 2,4,5)	8 (wheels 1,2,5,6)	4 (wheels 1,5)	
SHOCK ABSORBER — TYPE	Direct-acting w/ lock out	Does not apply.	Does not apply.	Does not apply.	Double-acting cam and lever	Direct-acting	Direct-acting	
SHOCK ABSORBER — CONTROL		Does not apply.	Does not apply.	Does not apply.	C121535 C121233	7716544	60 cpm, 2300 rebound & 3750 comp.; 100 cpm, 2750 rebound & 4450 comp.	
SPROCKET — PART NUMBER	PCF-F6263-446		C7324249	D6547366	C5621431		D8369036	
SPROCKET — TYPE	External tooth steel plate		External tooth plate riveted to disc	External tooth plate	External tooth plate welded to disc	External tooth steel plate	External tooth steel plate	
SPROCKET — NO. TEETH	11	19	9	13	31	13	12	13
SPROCKET — PITCH DIAM. (in.)	21.350	17.000	12.892	25.038	25.234	25.038	23.422	23.108
SPROCKET — ENGAGEMENT	Track shoe	Center	Track driving lugs	End connector	Track pin	End connector	Track shoe	
STATIC WHEEL LOADING (lb)	5830	582	373	3917	3550	4250	4180	4050
SUSPENSION % VEHICLE WEIGHT	19.4			33.0				
SPRING RATE						3338 in.-lb/deg		
CENTER OF GRAVITY (in.): VERTICAL / HORIZONTAL						46.5 100 from c/l sprocket	43.5 85 from c/l of sprocket	
NATURAL FREQUENCY (cpm): SUSPENSION / VEHICLE								
TRACK ANGLE (deg): APPROACH / DEPARTURE	30 Does not apply.	32 11	20 27	32 30	29 25	24 30	30 28	
TRACK ENVELOPE (in.)	L.-450 Rt.-456	246	274	474	422-1/2	504	432	434-1/2

TABLE A-7
SUSPENSION DATA

	Carriers							Tractors
VEHICLE	T31, Cargo Carrier	T48, Amphib. Cargo Carrier	T55, Infantry Utility	T78, Quad-Track Cargo Carrier	T78, Quad-Track Cargo Carrier	T107, Amphib. Cargo Carrier	T113, Carrier, Armored Personnel	T2, Heavy
SUSPENSION TYPE	Front drive w/support rollers and rear idler.	Front drive and trailing idler. No support roller.	Front drive and trailing idler. No support roller.	Sprocket mounted on axle and roller type wheel w/ponton.	Chain driven sprocket w/support rollers and spring idler.	Rear drive and front idler. No support rollers.	Front drive and leading arm idler. No support rollers. (ALUMINUM HULL)	Front drive w/support rollers and rear idler.
INSTALLATION DRAWING NUMBER							DTA 27640	E-3964
VEHICLE WEIGHT (lb): TARE / GROSS	60,000 80,000	7,550 12,000	10,600 12,100	8,500 11,500	10,950 13,950	7,000	18,580	29,900 38,900
ROAD WHEELS: QUANTITY	12 dual	10 spaced dual	8 single	84	32	6 dual/8 single	10 dual	12 single
ROAD WHEELS: TYPE	Solid rubber	Pneumatic	Pneumatic			Solid rubber-dual or pneumatic-single	Solid rubber	Solid rubber
ROAD WHEELS: SIZE (in.)	26 x 6	6.00 x 12	6-10 low profile			22 x 3-3/6 10 x 4.50	22 x 2	20 x 9
VEHICLE WIDTH ACROSS TRACKS (in.)	134	96	93	66-1/2	88	71-1/2	96	95
TRACK GROUND-CONTACT LENGTH (in.)	150-1/4	119	103-3/4	144	120	77-1/2	108	145
L/T RATIO	1.37	1.76	1.42	144/47-1/4	120/57-1/2	1.51	1.29	1.80
VEHICLE SPRINGING MEDIA	Torsion bar	Torsion bar and tube	Torsilastic	Leaf spring	Leaf spring & torsilastic	Compound torsilastic	Torsion bar	Horizontal volute
SHOCK ABSORBER: QUANTITY	8 (wheels 1,2,5,6)	4 (wheels 1,5)	6 (wheels 1,2,3)	None	None	None	4 (wheels 1,5)	None
SHOCK ABSORBER: TYPE	Direct-acting	Cam and lever	Direct-acting	Does not apply.	Does not apply.	Does not apply.	Direct-acting	Does not apply.
SHOCK ABSORBER: CONTROL	5200 rebound 1925 compression		OMX-1847 OMX-2700	Does not apply.	Does not apply.	Does not apply.	3000 rebound 1000 compression	Does not apply.
SPROCKET: PART NUMBER	D77076		OMX-1809			930118	DTA 27652	D33955
SPROCKET: TYPE	External tooth steel plate	Rubber-covered casting	External tooth rubber-covered			External tooth rubber-covered	External tooth steel plate	External tooth forging
SPROCKET: NO. TEETH	13	11	15			10	10	13
SPROCKET: PITCH DIAM. (in.)	26.068	14.000	18.500	23.040	13.369	14.331	19.400	
SPROCKET: ENGAGEMENT	Track shoe	Track shoe (dual)	Track cross bar	Track chain link	Track chain link	Track cross bar	Track shoe	End connector
STATIC WHEEL LOADING (lb)	6660	1200	1512	179	436	675	1658	3070
SUSPENSION % VEHICLE WEIGHT		24.9					18.8	
SPRING RATE								
CENTER OF GRAVITY (in.): VERTICAL / HORIZONTAL		45 100-1/2 from nose	26-3/4 53-5/8 from c/l sprocket					
NATURAL FREQUENCY (cpm): SUSPENSION / VEHICLE								
TRACK ANGLE (deg): APPROACH / DEPARTURE		40 Does not apply.	40 Does not apply.			30 35	20 Does not apply.	40 30
TRACK ENVELOPE (in.)	480	352	296			292.5	L.-348 Rt.-342	498

TABLE A-8
SUSPENSION DATA

		Tractors							
VEHICLE		T13, Cargo	T20, High Speed	T22, Heavy	T31, Medium	T38, Snow	T39, Light	T42 (M8), Cargo	T43E1, Cargo
SUSPENSION TYPE		Front drive w/support rollers and trailing idlers.	Front drive w/support rollers and trailing idler.	Front drive w/support rollers and trailing idler.	Front drive and support rollers.	Front drive and trailing idler. No support rollers.	Front drive and trailing idler. No support rollers.	Front drive w/support rollers and rear compensating idler.	Front drive w/support rollers and rear compensating idler.
INSTALLATION DRAWING NUMBER						E-1428	E-1021	7098910	7754553
VEHICLE WEIGHT (lb): TARE GROSS		20,875 25,000	 20,665	60,000 75,000		11,500 14,500	14,000 18,000	35,500 48,000	26,000 33,600
ROAD WHEELS	QUANTITY	8 single	8 single, w/idler	12 dual	16 single	8 dual	8 single	12 dual	10 dual
	TYPE	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Pneumatic	Solid rubber	Solid rubber	Solid rubber
	SIZE (in.)	20 x 6		20 x 6	14 x 4-1/2	5.50 x 16	30 x 4-1/2	25-1/2 x 4-1/2	25-1/2 x 4-1/2
VEHICLE WIDTH ACROSS TRACKS (in.)		92-5/8	96-5/8	120	95-5/8	103-1/2	93-1/2	117	108
TRACK GROUND-CONTACT LENGTH (in.)		109	110-1/2	172	92-3/8	116	102-3/4	149-1/8	115-3/4
L/T RATIO		1.22	1.34	1.76	1.14	1.70	1.39	1.51	1.33
VEHICLE SPRINGING MEDIA		Coil spring	Vertical volute	Horizontal volute	Vertical volute	Torsion bar	Torsion bar	Torsion bar	Torsion bar
SHOCK ABSORBER	QUANTITY	None	None	None	None	8 (wheels 1,2,3,4)	4 (wheels 1,4)	8 (wheels 1,2,5,6)	8 (wheels 1,2,4,5)
	TYPE	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Cam and lever	Cam and lever	Direct-acting	Direct-acting
	CONTROL	Does not apply.	Does not apply.	Does not apply.	Does not apply.			C136964	7754637
SPROCKET	PART NUMBER		D6548406	D6578630 D6578555				7325804	D7762171
	TYPE		External tooth forging	External tooth casting & plate			External tooth steel palate	External tooth steel plate	External tooth steel plate
	NO. TEETH	14	18	13	20	10	9	14	13
	PITCH DIAM. (in.)	24.560	22.918	25.038	25.500	14.324	14.304	26.52	22.979
	ENGAGEMENT	End connector	Center	End connector & center guide		Track shoe	Track shoe	Track shoe	Track shoe
STATIC WHEEL LOADING (lb)		2500	2580	5347	3125	1450	2250	4000	3200
SUSPENSION % VEHICLE WEIGHT									23.7
SPRING RATE								3338 in.-lb/deg	
CENTER OF GRAVITY (in.): VERTICAL HORIZONTAL		34.5 91-13/16 from front bumper		46 100 from front bumper	42 88 from front bumper	24 47 from front wheel	30 72-1/2 from c/l of sprocket	46 100 from c/l of sprocket	
NATURAL FREQUENCY (cpm): SUSPENSION VEHICLE									
TRACK ANGLE (deg): APPROACH DEPARTURE		 Does not apply.		35 Does not apply		 Does not apply.	39 Does not apply.	24 32	36 30
TRACK ENVELOPE (in.)				498	296	360	335	504	418

TABLE A-9
SUSPENSION DATA

		Tractors						Trailers	
VEHICLE		T93E1, Amphib. Cargo	T94, Cargo	T116, Amphib. Cargo	T121, Snow	Fraudee Snoshu	Ground Hog	T79, Amphibian	T80, Cargo, 4-ton
SUSPENSION TYPE		Front drive w/support rollers and rear compensating idler.	Front drive w/support rollers and rear idler.	Front drive and fixed idler. No support rollers.	Front drive and spring loaded idler. No support rollers.		Rear drive w/support rollers and front idler.	Front sprocket (not driven) and rear idler. No support rollers.	Quadri-track.
INSTALLATION DRAWING NUMBER				PCF-K6319-1120	8358469				
VEHICLE WEIGHT (lb): TARE GROSS		29,000 42,750	68,000 72,000	8,500 loaded 6,000 empty	1,090 1,340	3,875		8,000 13,000	10,640 18,640
ROAD WHEELS	QUANTITY	10 dual	12 dual	10 dual	None	8	20 single	8 spaced duals	8 sets each of end and intermediate
	TYPE	Solid rubber	Solid rubber	Solid rubber	Does not apply.	Pneumatic	Steel	Pneumatic	Solid rubber
	SIZE (in.)	25-1/2 x 4-1/2	26 x 6	22 x 2	Does not apply.	6.00 x 16		15-6.60	15 x 6 & 9-1/2 x 6
VEHICLE WIDTH ACROSS TRACKS (in.)		124	124	75-3/4	30	97		98	91-1/2
TRACK GROUND-CONTACT LENGTH (in.)		128-1/4	175	98	50	98-1/2		98-5/8	130
L/T RATIO		1.25	1.73	1.76	2.50	98-1/2/64		1.45	130/80
VEHICLE SPRINGING MEDIA		Torsion bar	Torsion bar	Torsion bar	Track runner on leaf spring		Transverse leaf	Torsion bar and tube	Torsion bar
SHOCK ABSORBER	QUANTITY	4 (wheels 1,5)	8 (wheels 1,2,5,6)	4 (wheels 1,5)	None		None	4 (wheels 1,4)	None
	TYPE	Direct-acting	Direct-acting	Rotary	Does not apply.		Does not apply.	Cam and lever	Does not apply.
	CONTROL				Does not apply.		Does not apply.	7980677 7980607	Does not apply.
SPROCKET	PART NUMBER		D6547366	PCF-6319-1686				D7980550	
	TYPE	External tooth steel plate	External tooth steel plate	External tooth steel plate	External tooth steel plate			External tooth rubber-covered	
	NO. TEETH	12	13	11	10			12	
	PITCH DIAM. (in.)	22.920	25,038	14.00	10.270			7.639	
	ENGAGEMENT	Track shoe	End connector	Track shoe	Track shoe			Track shoe (dual)	
STATIC WHEEL LOADING (lb)		4275	6000	850	Does not apply.	646		1625	
SUSPENSION % VEHICLE WEIGHT				25.8					
SPRING RATE				767.6 in.-lb/deg					
CENTER OF GRAVITY (in.): VERTICAL HORIZONTAL					16.75 27.13 from c/l of sprocket				
NATURAL FREQUENCY (cpm): SUSPENSION VEHICLE									
TRACK ANGLE (deg): APPROACH DEPARTURE		30 28	32 30	22-1/2 21	46.3 41.7	53-1/4 Does not apply.		41 46	40 83
TRACK ENVELOPE (in.)		456	582	352	147			352	

TABLE A-10
SUSPENSION DATA

		Trailers		Special Purpose Vehicles						
VEHICLE		T81, Wanigan	Track-laying, 6-ton	M51, Heavy Recovery	M74 (T74)	T2, Tank Recovery, Armored	T4, Wrecker	T51, Heavy Tank Recovery	T88, Medium Recovery	T120, Wrecker, Armored
SUSPENSION TYPE		Quaird-track.	Twin idler tracks	Rear drive w/support rollers and front compensating idler.	Front drive w/support rollers and rear idler.	Front drive w/support rollers and rear idler.	Front drive w/support rollers and rear compensating idler.	Rear drive w/support rollers and front compensating idler.	Rear drive w/support rollers and front compensating idler.	Front drive and trailing idler. No support rollers.
INSTALLATION DRAWING NUMBER				K8686040			7755479	8347186	BML-6251E	PCF-6263-496
VEHICLE WEIGHT (lb): TARE GROSS		16,360 18,827	20,800	111,500 120,000	88,600 95,750	65,620	64,300 66,800	98,400 105,000	95,000	48,000
ROAD WHEELS	QUANTITY		4 dual	14 dual	12 dual	12 single	12 dual	14 dual	12 dual	10 dual
	TYPE		All steel	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber	Solid rubber
	SIZE (in.)		32	26 x 6	20 x 6-1/4	20 x 8	25-1/2 x 4-1/2	26 x 6	26 x 6	32 x 4
VEHICLE WIDTH ACROSS TRACKS (in.)			86	143	112-1/8	100-1/8	123-1/2	143	135	124
TRACK GROUND-CONTACT LENGTH (in.)			44	185-3/4	150-7/8	147	157-1/8	185-3/4	180	144
L/T RATIO			0.61	1.62	1.69	1.75	1.53	1.62	1.68	1.36
VEHICLE SPRINGING MEDIA			None	Torsion bar	Horizontal volute	Vertical volute	Torsion bar	Torsion bar	Torsion bar	Torsion bar
SHOCK ABSORBER	QUANTITY	None	None	8 (wheels 1,2,6,7)	6 (Between suspension arms)	None	8 (wheels 1,2,5,6)	8 (wheels 1,2,6,7)	6 (wheels 1,2,6)	10 (all wheels)
	TYPE	Does not apply.	Does not apply.	Direct-acting	Double-acting	Does not apply.	Direct-acting	Direct-acting	Direct-acting	Direct-acting w/lock out
	CONTROL	Does not apply.	Does not apply.	3100 rebound 2500 compression	6580067	Does not apply.	50 cpm, 2300 rebound & 3750 comp.; 100 cpm, 2750 rebound & 4450 comp.	D6580067	7953032	
SPROCKET	PART NUMBER		None	D8705893	D6547366	D6547366B	D7962691	K8705893	K8671597	PCF-F6263-446
	TYPE		Does not apply.	External tooth steel plate	External tooth steel plate	External tooth forging	External tooth steel plate	External tooth steel plate	External tooth steel plate	External tooth steel plate
	NO. TEETH		Does not apply.	11	13	13	12	11	11	11
	PITCH DIAM. (in.)		Does not apply.	25.00	25.038	25.038	23.422	25.000	25.504	21.350
	ENGAGEMENT			End connector	End connector	End connector	Track shoe	End connector	End connector	Track shoe
STATIC WHEEL LOADING (lb)			5070	8572	7979	5468	5734	8375	7910	4800
SUSPENSION % VEHICLE WEIGHT					17.3	23.6				23.5
SPRING RATE			Does not apply.	5950 in.-lb/deg			7350 in.-lb/deg	5950 in.-lb/deg		
CENTER OF GRAVITY (in): VERTICAL HORIZONTAL				53 114 from c/l cross drive						
NATURAL FREQUENCY (cpm): SUSPENSION VEHICLE								86		
TRACK ANGLE (deg): APPROACH DEPARTURE			Does not apply. Does not apply.	35 35	39 30	32 30	26 32	35 35		30 Does not apply.
TRACK ENVELOPE (in.)			189	574.594	474	474	522	574.594		L.-450 Rt.-456

TABLE A-11
TRACK DATA

	MODEL	T1	T2	T3	T4	T16	T16E1	T16E2	T16E3
	VEHICLE APPLICATION	Track Development Ch., T2	Allis Chalmers, Model 35	Caterpillar, Model 20	Medium Prime Mover, T12	Light Tank, M3, M3A1,M3A3,M5A1 Motor Carriage, M8, T64 Tractor,M5,M5A1 Cargo Carrier, M30 Recon. Vehicle T8, T8E1	Light Tank, M3, M3A1,M3A3	Light Tank, M3, M3A1, M3A3	Light Tank, M3, M3A1, M3A3
	DESCRIPTION	Cast Steel, w/Shoes Welded on Tread Plates	Cast Steel Shoes.	Cast Steel, w/Shoes Bolted on Tread Plates	Rubber and Plate Type, Fabricated.	Rubber, Reversible, Double-Pin, Block Thickness 2-15/16 in.	Rubber, Reversible, Double-Pin.	Rubber, Non-reversible, Double-Pin, Block Thickness 3 in.	Rubber, Non-reversible, Double-Pin, Block Thickness 3 in.
	ASSY. NO. (1 Shoe)	APG-2651AD	APG-2280AC	APG-L1150	APG-L1167	D34954	C63362	C5562024	D31862
	ASSY. NO. (8 Shoe-Strand or Section)					A7964955			
TRACK	WEIGHT (lb): SHOE (Pin Track) SECTION (Band Track)					21.5	23	22.6	22.6
TRACK	PITCH (in.)	5-1/8	6-1/2	6-3/4		5-1/2	5-1/2	5-1/2	5-1/2
TRACK	LENGTH OF SECTION (Band Track) (in.)	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.
TRACK	WIDTH (in.)	10-1/2	13	13		11-5/8	11-5/8	11-5/8	11-5/8
TRACK	GROUSER CONTACT AREA (sq in.)	Does not apply.	Does not apply.	Does not apply.	Does not apply.	26.2	26.2	26.2	26.2
TRACK	GROUSER HEIGHT (in.)					Flat	Flat	Flat	Flat
TRACK	PIN DIA. (in.)					1.000	1.000	1.000	1.000
TRACK	GUIDE TYPE					Removable outer	Removable outer	Removable outer	Removable outer
BUSHING	TYPE	Rubber	Rubber	Rubber		Rubber	Rubber	Rubber	Rubber
BUSHING	PROJECTED AREA (in.)					12.03	12.03	12.03	12.03
BUSHING	FLEXIBLE ANGLE (deg.) (At Assembly)					8	8	8	8
BUSHING	RUBBER ID (in.)					1.000	1.000	1.000	1.000
BUSHING	WALL THICKNESS (in.)					0.187	0.187	0.187	0.187
BUSHING	% STRAIN					9.6	13	13	13

TABLE A-12
TRACK DATA

	MODEL	T16E4	T16E5	T16E6	T16E7	T17	T17E1	T24	T24E1
	VEHICLE APPLICATION	Light Tank, M3, M3A1, M3A3	Combat Car, M1E2	Combat Car, M1	Combat Car, M1	Chevrolet, HTTIE1	Chevrolet, HTT1E1	Half Track Truck, T5	Light Tractor, TA30 Half Track Truck, T9, T9E1
	DESCRIPTION	Rubber, Reversible, Double-Pin.	Rubber, Non-reversible, Double-Pin, Block Thickness 2-7/8 in.	Rubber, Non-reversible, Double-Pin.	Rubber, Non-reversible, Double-Pin, Block Thickness 2-5/8 in.	Rubber Block Bases On Rubber Chain Incorporating Extended Pins.	Same as Model T17 Track except that the blocks are slightly heavier in weight	Band, Rubber, Continuous.	Band, Rubber, Continuous.
	ASSY. NO. (1 Shoe)	D31863	D31864	D31866	D31866	Does not apply.	Does not apply.	Does not apply.	Does not apply.
	ASSY. NO. (8 Shoe-Strand or Section)					APG-L1226	APG-L1226	Goodrich G-5376-B	D6527860
TRACK	WEIGHT (lb): SHOE (Pin Track) SECTION (Band Track)	23	21.5	23					
TRACK	PITCH (in.)	5-1/2	5-1/2	5-1/2	5-1/2	2	2	4	4
TRACK	LENGTH OF SECTION (Band Track) (in.)	Does not apply.	Does not apply.	Does not apply.	Does not apply.				
TRACK	WIDTH (in.)	11-5/8	11-5/8	11-5/8	11-5/8	7-1/2	7-1/2	10	10
TRACK	GROUSER CONTACT AREA (sq in.)	26.2	26.2	26.2	26.2				
TRACK	GROUSER HEIGHT (in.)	Flat	Flat	Flat	Flat			Flat	Flat
TRACK	PIN DIA. (in.)	1.000	1.000	1.000	1.00	Does not apply.	Does not apply.	Does not apply.	Does not apply.
TRACK	GUIDE TYPE	Removable outer	Removable outer	Removable outer	Removable outer			Center plate	Center plate
BUSHING	TYPE	Rubber	Rubber	Rubber	Rubber	Does not apply.	Does not apply.	Does not apply.	Does not apply.
BUSHING	PROJECTED AREA (in.)	12.03	12.03	12.03	12.03	Does not apply.	Does not apply.	Does not apply.	Does not apply.
BUSHING	FLEXIBLE ANGLE (deg.) (At Assembly)	8	8	8	8	Does not apply.	Does not apply.	Does not apply.	Does not apply.
BUSHING	RUBBER ID (in.)	1.000	1.000	1.000	1.000	Does not apply.	Does not apply.	Does not apply.	Does not apply.
BUSHING	WALL THICKNESS (in.)	0.187	0.187	0.187	0.187	Does not apply.	Does not apply.	Does not apply.	Does not apply.
BUSHING	% STRAIN	13	13	13	13	Does not apply.	Does not apply.	Does not apply.	Does not apply.

TABLE A-13
TRACK DATA

	MODEL	T36	T36E1	T36E2	T36E3	T36E4	T36E5	T36E6	T36E7
	VEHICLE APPLICATION	Light Tank, M3, M5	Light Tank, M3	Medium Tank, M3, M4 Series	Medium Tank, M3, M4 Series	Medium Tank, M3, M4 Series	Medium Tank, M3, M4 Series	Light Tank, M3 Tractor, M5, M5A1 Motor Carriage, M8, T64 Cargo Carrier, M30 Recon. Vehicle, T8, T8E1	Light Tank, M3, M5
	DESCRIPTION	Rubber, Double-Pin, Detachable Tread Plates.	Steel and Rubber, Double-Pin, Detachable Steel Grousers.	Steel, Rubber-Backed Surface, Double-Pin, Detachable Steel Grousers.	Steel, Rubber-Backed Surface, Double-Pin, Detachable Steel Grousers.	Steel, Rubber-Backed Surface, Double-Tapered Pin, Detachable Steel Grousers.	Steel and Rubber, Double-Pin, Closed End Connectors, Removable Steel Grousers.	Steel, Removable Interrupted Parallel Grousers. Double-Pin.	Steel, Rubber-Backed Surface, Interrupted Parallel Grousers.
	ASSY. NO. (1 Shoe)	D48011	D48011	D47934	D47896	D6547986		D78054	D40992
	ASSY. NO. (8 Shoe-Strand or Section)							D78054B (10 shoes)	D40993 (10 shoes)
TRACK	WEIGHT (lb): SHOE (Pin Track) SECTION (Band Track)		26.4				45	27.5	28
TRACK	PITCH (in.)	5-1/2	5-1/2	6	6	6	6	5-1/2	5-1/2
TRACK	LENGTH OF SECTION (Band Track) (in.)	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.
TRACK	WIDTH (in.)	11-5/8	11-5/8	16	16	16	16	11-5/8	11-5/8
TRACK	GROUSER CONTACT AREA (sq in.)	Does not apply.		Does not apply.	Does not apply.	Does not apply.		Does not apply.	Does not apply.
TRACK	GROUSER HEIGHT (in.)	1	1	1	1	1	1	1	1
TRACK	PIN DIA. (in.)	1.000	1.000	1.125	1.125	1.125	1.250	1.000	1.000
TRACK	GUIDE TYPE	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer
BUSHING	TYPE	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber
BUSHING	PROJECTED AREA (in.)	12.03	11.830	12.900	12.900	12.900	19.515	11.830	11.830
BUSHING	FLEXIBLE ANGLE (deg.) (At Assembly)	8	8	8	8	8	8	8	8
BUSHING	RUBBER ID (in.)	1.000	1.000	1.125	1.125	1.125	1.250	1.000	1.000
BUSHING	WALL THICKNESS (in.)	0.187	0.187	0.187	0.187	0.187	0.187	0.187	0.187
BUSHING	% STRAIN	13	9.6	16	16	16	20	9.6	9.6

TABLE A-14
TRACK DATA

	MODEL	T48	T48E1	T48E1B	T49	T51	T54	T54E1	T54E2
	VEHICLE APPLICATION	Medium Tank, M3, M4 Series, T23 Recovery Vehicle, M32 Series Tractor, M4 Series, T2 Motor Carriage, M7, M7B1, M10 M10A1, M12, M36, M36B1, T36, T52, T53E1 Cargo Carrier, M30, T22, T23 Wrecker, T2	Medium Tank, M3, M4 Series	Medium tank, M3, M4 Series	Medium Tank, M3, M4 Series Motor Carriage, M7, M7B1, M10, M10A1, M36, M36B1 Recovery Vehicle, M32 Series Tractor, M4 Series, T2 Wrecker, T2 Cargo Carrier, M30, T22, T23	Medium Tank, M3, M4 Series Wrecker, T5 Motor Carriage, M7, M7B1, M10, M10A1, M12, M36 M36B1	Medium Tank, M3, M4 Series (Never Built)	Medium Tank, M3, M4 Series, M7 Recovery Vehicle, M32 Series Tractor, M4 Series, T2 Wrecker, T2 Motor Carriage, M7, M7B1, M10 M10A1, M12, M36 Series Cargo Carrier, M30, T22, T23	Medium Tank, M4A4
	DESCRIPTION	Rubber, Double-Pin, Chevron Grouser.	Steel, Double-Pin, Chevron Grouser, Rolled and Welded Linkages.	Rubber, Non-reversible, Double-Pin, Smooth Tread Face.	Steel, Interrupted Parallel Grousers. Double-Pin.	Rubber, Non-reversible, Double-Pin, Smooth Tread Face.	Steel, Integral Parallel Grousers.	Steel, Double-Pin, Integral Chevron Grousers.	Steel, Double-Pin, Integral Chevron Grousers.
	ASSY. NO. (I Shoe)	D48067A	Goodrich G3019E	NR	D48026	D48076A	D58627	D58628	D6550172
	ASSY. NO. (8 Shoe-Strand or Section)	D48067B (10 shoes)			7964957	D48076B (10-shoe strand)		D58628B (10 shoes)	
TRACK	WEIGHT (lb): SHOE (Pin Track) SECTION (Band Track)	41			47.5	38		48	46.2
TRACK	PITCH (in.)	6	6	6	6	6	6	6	6
TRACK	LENGTH OF SECTION (Band Track) (in.)	Does not apply.		Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.
TRACK	WIDTH (in.)	16-9/16	16-9/16	16-9/16	16-9/16	16-9/16	16-9/16	16	16
TRACK	GROUSER CONTACT AREA (sq in.)	26.7		Does not apply.	31.5	60	Does not apply.	Does not apply.	Does not apply.
TRACK	GROUSER HEIGHT (in.)	1-1/2		Flat	1.1/4	Flat	1-1/4	1	1
TRACK	PIN DIA. (in.)	1.250			1.250	1.250	1.250	1.250	1.250
TRACK	GUIDE TYPE	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer
BUSHING	TYPE	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber
BUSHING	PROJECTED AREA (in.)	19.150			19.150	19.150	19.150	19.150	19.150
BUSHING	FLEXIBLE ANGLE (deg.) (At Assembly)	8			8	8	8	8	8
BUSHING	RUBBER ID (in.)	1.250			1.250	1.250	1.250	1.250	1.250
BUSHING	WALL THICKNESS (in.)	0.187			0.187	0.187	0.187	0.187	0.187
BUSHING	% STRAIN	20			20	20	20	22	20

TABLE A-15
TRACK DATA

	MODEL	T54E3	T55	T55E1	T55E2	T56	T56E1	T62	T68E1
	VEHICLE APPLICATION	Medium Tank, M4A1, M4A3	Light Tank, M3 (Never Built)	Light Tank, M3A1 Tractor, M5, M5A1 Motor Carriage, M8, T64 Cargo Carrier, M30 Recon. Vehicle, T8, T8E1	Light Tank, T7E2	Medium Tank, M3, M4 Series	Medium Tank, M3, M4 Series, Motor Carriage, M7, M7B1, M10, M10A1, M12 Wrecker, T2 Cargo Carrier, M30, T22, T23 Tractor, M4 Series, T2	Medium, Tank, M4 Series Motor Carriage, M7, M7B1, M10, M10A1, M12 Wrecker, T2 Crago Carrier, M30, T22, T23 Tractor, M4 Series, T2	Half Track, M2, M2A1, M3, M3A1, M3A2, M4, M4A1, M5, M5A1, M9A1, M13, M14, M15, M15A1, M16, M16A1, M16A2, M17, M21, T10E1, T19, T29, T30, T31, T48
	DESCRIPTION	Steel, Double-Pin Integral Chevron Grousers.	Steel, Rubber-Jointed, Parallel Interrupted Grousers.	Steel, Fabricated, Rubber-Jointed, Integral Chevron Grousers. Double-Pin.	Steel, Rubber-Jointed, Integral Chevron Grousers. Double-Pin.	Cast Steel, w/Bolt-on Type Interrupted Grousers. Double-Pin.	Cast Steel, w/Bolt-on Type Chevron Grousers. Double-Pin.	Steel, Rubber-Jointed, Riveted Chevron Grousers. Double-Pin.	Endless Band, Rubber, Integral Parallel Grousers.
	ASSY. NO. (1 Shoe)		D58630	D58632	Budd Mfg. 505938	D58641	D58642	D65978	C110400
	ASSY. NO. (8 Shoe-Strand or Section)			D58632B (10 shoes)		D58641B (10 shoes)	D58642B (10 shoes)	7007201 (10 shoes)	C5600400
TRACK	WEIGHT (lb): SHOE (Pin Track) SECTION (Band Track)			27.0		48.7	48.3	49.3	495 at 8.1#/pitch
TRACK	PITCH (in.)	6	5-1/2	5-1/2	5-1/2	6	6	6	4
TRACK	LENGTH OF SECTION (Band Track) (in.)	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply	Does not apply.	232
TRACK	WIDTH (in.)	17-1/2	11-11/16	11-11/16	14	16	16	16-9/16	13-1/4
TRACK	GROUSER CONTACT AREA (sq in.)	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	
TRACK	GROUSER HEIGHT (in.)	1	1	1	1	1	1	1	Does not apply.
TRACK	PIN DIA. (in.)	1.250	1.000	1.000	1.000	1.250	1.250	1.250	Does not apply.
TRACK	GUIDE TYPE	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer	Removable outer	Removable center
BUSHING	TYPE	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Does not apply.
BUSHING	PROJECTED AREA (in.)	19.515	11.171	11.171	12.033	19.150	19.150	19.515	Does not apply.
BUSHING	FLEXIBLE ANGLE (deg.) (At Assembly)	8	8	8	8	8	8	8	Does not apply.
BUSHING	RUBBER ID (in.)	1.250	1.000	1.000	1.000	1.250	1.250	1.250	Does not apply.
BUSHING	WALL THICKNESS (in.)	0.187	0.187	0.187	0.187	0.187	0.187	0.187	Does not apply.
BUSHING	% STRAIN	20	15	15	15	20	20	20	Does not apply.

TABLE A-16
TRACK DATA

	MODEL	T69	T72	T72E1	T80	T80E1	T80E2	T80E3	T80E4
	VEHICLE APPLICATION	Flame Thrower Tank, T65 Utility Vehicle, M39, T41, T41E1 Motor Carriage, M18, T70, T86, T86E1, T87	Light Tank, M24, T24 Motor Carriage, M19, M19A1, M37 M41, T16, T16E1, T64E1, T65, T76, T77, T96 Cargo Carrier, T22E1, T23E1 Self Propelled, T27, T29, T38 Wrecker, T5E1	Light Tank, M24, T24 Motor Carriage, M19, M19A1, M37, M41, T16, T16E1, T64E1, T65, T76, T77, T96 Cargo Carrier, T22E1, T23E1 Self-Propelled, Wrecker T5E1	Medium Tank, M4A1, M4A3 Series, T22 Motor Carriage, M40, M43 (T89), T95 Heavy Tank, T28 Recovery Vehicle M32 Series	Medium Tank, M26 Series M45, M46, M46A1, T26E4, T26E5, T40 Armored Engineer Vehicle, T39 Tractor, T94	High Speed Tractor, M6 (T23)	Heavy Tank, T29, T30, T32, T34 Motor Carriage, T92, T93	Medium Tank, M46, M46, M46A1, M47
	DESCRIPTION	Forged Steel, Single Pin, Parallel Grousers.	Cast Steel, Single-Pin, Parallel Grousers.	Same as Model T72 Track except that block has holes for attaching ice cleats or extending grousers.	Steel, Rubber-Backed Surface, Double-Pin, Welded Chevron Grousers.	Steel, Rubber-Backed Surface, Double-Pin, Welded Chevron Grousers.	Steel, Rubber-Backed Surface, Double-Pin, Welded Chevron Grousers.	Steel, Rubber-Backed Surface, Double-Pin, Outer Extended End Connector.	Steel, Rubber-Backed Surface, Double-Pin, Welded Chevron Grousers.
	ASSY. NO. (1 Shoe)	D67966A	D76086	D7038144	D7069557	D7044266	D7722602	D7321627 Rt. D7321628 Lt.	7725704
	ASSY. NO. (8 Shoe-Strand or Section)	D67966B (10 shoes)	5700194	7038145	7069575	5700190	7723119	Rt.-7321631 L.-7321632	Rt.-7524394 L.-7524395
TRACK	WEIGHT (lb): SHOE (Pin Track) SECTION (Band Track)	19.6	25	44.75	56.5	55.0	57.1	57.5	57.5
TRACK	PITCH (in.)	5-3/32	5-1/2	5-1/2	6	6	6	6	6
TRACK	LENGTH OF SECTION (Band Track) (in.)	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.
TRACK	WIDTH (in.)	14-3/8	16	16	23	23	23	23	23
TRACK	GROUSER CONTACT AREA (sq in.)	20.1	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.
TRACK	GROUSER HEIGHT (in.)	1	7/8	7/8	1	1	1-1/2	1	2
TRACK	PIN DIA. (in.)	1.1250	0.830	0.830	1.250	1.250	1.250	1.250	1.250
TRACK	GUIDE TYPE	Integral center	Integral center	Integral center	Removable center	Removable center	Removable center	Removable center	Removable center
BUSHING	TYPE	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber
BUSHING	PROJECTED AREA (in.)	13.130	11.643	11.643	26.125	26.125	26.125	26.125	26.125
BUSHING	FLEXIBLE ANGLE (deg.) (At Assembly)	7	7	14	8	8	8	8	8
BUSHING	RUBBER ID (in.)	1.1250	1	1	1.250	1.250	1.250	1.250	1.250
BUSHING	WALL THICKNESS (in.)	0.210	0.182	0.187	0.187	0.187	0.187	0.187	0.187
BUSHING	% STRAIN	21.9	8.5	43	20	20	20	20	20

TABLE A-17
TRACK DATA

	MODEL	T80E5	T80E6	T80E7	T81	T84	T84E1	T84E2	T84E3
	VEHICLE APPLICATION	Medium Tank, M4 Series, M43 Howitzer Motor Carriage, M43 Recovery Vehicle, M74 Tank, Flame Thrower, T33, T68	Medium Tank, M46, M46A1, M47, T40, T42 Self-Propelled, T97 (M53) Armored Eng. Vehicle, T39E1	Heavy Tank, T43 Cargo Tractor, T44	Medium Tank, T26 Medium Tank, M26, M26A1 Motor Carriage, T84 Cargo Carrier, T31	Medium Tank, M4 Series Gun Motor Carriage, M40 Howitzer Motor Carriage, M43 Recovery Vehicle, M32 Series Tank, Flame Thrower, T68	Medium Tank, M26 Series, M45, M46, M46A1, M47, T40 Self-Propelled, T97 (M53), T108 (M55) Tractor, T94 Armored Engineer Vehicle, T39E1 Tank, Flame Thrower, T66	High Speed Tractor, M6 (T23)	Motor Carriage, T88, T92, T93 Heavy Tank, T29, T30, T32, T34
	DESCRIPTION	Steel, Rubber-Backed Surface, Double-Pin, Welded or Integral Chevron Grousers.	Steel, Rubber-Backed Surface, Double-Pin, Iotegral or Welded Chevroo Grousers.	Steel, Rubber-Backed Surface, Double-Pin, Welded Chevron Grousers, Extended End Coo-oectors on Both Sides.	Cast Steel, Single-Pin, Par-allel Grousers.	Rubber, Double-Pin, Iotegral Chevron Grousers.	Rubber, Double-Pin, Iotegral Chevron Grousers.	Rubber, Double-Pin, Iotegral Chevroo Grousers.	Rubber, Double-Pin, Integral Chevron Grousers, Outer Extended End Cooneetors.
	ASSY. NO. (1 Shoe)	A7387206	D7386994	D7387435	Rt. 7014319 L. 7013904	D7048211	D7049657	D7049655	D7073127
	ASSY. NO. (8 Shoe-Straod or Sectioo)	7388421	7386996	7387434	D7058061 (82 shoes)	7320706	7072236	7072234	Rt. 7073337 (82 shoes) L. 7073338 (82 shoes)
TRACK	WEIGHT (lb): SHOE (Pin Track) SECTION (Baod Track)	59	57.5	60	43.5	51.7	50.6	53.6	53.1
TRACK	PITCH (in.)	6	6	6	6	6	6	6	6
TRACK	LENGTH OF SECTION (Band Track) (io.)	Does oot apply.	Does oot apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does oot apply.
TRACK	WIDTH (in.)	23	23	28, including extended cono.	24	23	23	23	28, including extended conn.
TRACK	GROUSER CONTACT AREA (sq in.)	Does not apply.	Does oot apply.	Does oot apply.	Does not apply.	31.2	39.8	39.8	39.8
TRACK	GROUSER HEIGHT (in.)	1-1/2	1-1/2	1-1/2	3/4	1	1-1/2	1-1/2	1-1/2
TRACK	PIN DIA. (in.)	1.250	1.250	1.250	1.000	1.250	1.250	1.250	1.250
TRACK	GUIDE TYPE	Removable ceoter	Removable center	Removable ceoter	Integral center	Removable center	Removable center	Removable center	Removable center
BUSHING	TYPE	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber
BUSHING	PROJECTED AREA (in.)	26.125	26.125	26.125	25.580	26.125	26.125	26.125	26.125
BUSHING	FLEXIBLE ANGLE (deg.) (At Assembly)	8	8	8	7	8	8	8	8
BUSHING	RUBBER ID (in.)	1.250	1.250	1.250	1.000	1.250	1.250	1.250	1.280
BUSHING	WALL THICKNESS (io.)	0.187	0.187	0.187	0.156	0.187	0.187	0.187	0.187
BUSHING	% STRAIN	20	20	20	16	20	20	20	20

TABLE A-18
TRACK DATA

	MODEL	T85	T85E1	T88	T88E1	T88E2	T91	T91E1	T91E2
	VEHICLE APPLICATION	Light Tank, M24	Light Tank, M24 Motor Carriage, M19, M19A1, M37, M41, M18, T86, T86E1, T87 Utility Vehicle, M39, T41, T41E1 Cargo Carrier, T22E1, T23E1	Cargo Tractor, M8, (T42) Armored Utility Vehicle, M44	Cargo Tractor, M8E1 Utility Armored Vehicle, M44E1 Light Tank, T41E1	Light Tank, M41E1, M41E2, T41E2	Light Tank, T37, T41 Self-Propelled, T98, T99 Motor Carriage, T100	Light Tank, T37	Light Tank, T41E1
	DESCRIPTION	Rubber, Double-Pin, Integral Chevron Grousers.	Rubber, Double-Pin, Integral Chevron Grousers.	Rubber, Double-Pin, Integral Chevron Grousers.	Rubber, Double-Pin, Integral Chevron Grousers.	Rubber, Double-Pin, Integral Chevron Grousers.	Forged Steel and Rubber, Detachable Rubber Pad, Single-Pin. Interrupted Chevron Grousers.	Forged Steel and Rubber Detachable Rubber Pad, Single-Pin.	Forged Steel and Rubber, Detachable Rubber Pad, Single-Pin.
	ASSY. NO. (1 Shoe)	D7073126	D7074450	C7038297	D7074680	F8376150	D7717559	D7728372	D7384181
	ASSY. NO. (8 Shoe-Straud or Section)	7073311	7074480		7074682 (83 shoes)	8376151	w/pads 7716509 w/o pads 7717129	w/pads 7752596 w/o pads 7752595	w/pads 7384179 w/o pads 7384180
TRACK	WEIGHT (lb); SHOE (Pin Track) SECTION (Band Track)	26.3	25.8	35.5	36.7	33.3	38 w/pad pad 4.5	40.3 w/pad pad 4.5	31.6 w/pad pad 5
	PITCH (in.)	5-1/2	5-1/2	6	6	6	6	6	6
	LENGTH OF SECTION (Band Track) (in.)	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.
	WIDTH (in.)	14	14	17	17	21	21	21	21
	GROUSER CONTACT AREA (sq in.)	16	22.2	22.6	22.6	26.5	pad 26	pad 26	pad 44.0
	GROUSER HEIGHT (in.)	1-1/4	1-1/4	1-1/2	1-1/2	1-1/2	1/2	1/2	1/2
	PIN DIA. (in.)	1.000	1.000	1.000	1.000	1.000	0.878 octagonal	0.878 octagonal	0.878 octagonal
	GUIDE TYPE	Integral center	Integral center	Removable center	Removable center	Removable center	Integral center	Integral center	Integral center
BUSHING	TYPE	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber
	PROJECTED AREA (in.)	12.600	12.600	12.00	12.00	16.00	14.43	14.43	14.43
	FLEXIBLE ANGLE (deg.) (At Assembly)	8	8	8	8	8	6	7-1/2	7-1/2
	RUBBER ID (in.)	1.000	1.000	1.000	1.000	1.000	1.125	1.130	1.130
	WALL THICKNESS (in.)	0.187	0.187	0.187	0.187	0.187	0.187	0.187	0.187
	% STRAIN	13	13	13	14	13	9.9	10.5	10.5

TABLE A-19
TRACK DATA

	MODEL	T91E3	T91E4	T94	T94E1	T96	T96E1	T96E2	T97
	VEHICLE APPLICATION	Light Tank, M41, M41E1, M41E2, T41E1, T49 Self-Propelled, T98E1, T99E1, T141, T162, M42, M44 Armored Infantry Vehicle, M59, M75, T18E1, T73 Cargo Tractor, M8E2, T43E1, T43E2, T85, T86, T93, T93E1 Wrecker, T4	Light Tank, T41E2	Cargo Tractor, M8 (T42) Armored Utility Vehicle, M44	Light Tank, M41E1, M41E2, T41E1, T41E2 Self-Propelled, T98E1, M44 (T-194) Cargo Tractor, M8E2	Heavy Tank, T43, T43E2 (M103), T54, T54E1, T57, T58, T77 Medium Tank, M48 Recovery Vehicle, T51 Tank, Flame Thrower, M87 (T87)	Heavy Tank, T43 Medium Tank, M48	Heavy Tank, T43, T43E2 (M103) Medium Tank, M48	Heavy Tank, T43 Series, T54, T54E1, T57, T58, T77 Medium Tank, M48 Recovery Vehicle, T51
	DESCRIPTION	Cast Steel and Rubber, Detachable Rubber Pad, Single-Pin.	Same as Model T91E3 Track except for steel-backed surface.	Rubber, Single-Pin, Integral Chevron Grousers.	Rubber, Single-Pin, Chevron Type Grousers.	Forged Steel and Rubber, Double-Pin Block, Integral Chevron Grousers.	Same as Model T96 Track except for wedge design changes.	Same as Model T96 Track except for end connector and wedge design changes.	Rubber, Double-Pin, Chevron Type Grousers.
	ASSY. NO. (1 Shoe)	D7952165	8339204	D7715756	K7360668	D7379087	D7985691		D7354773
	ASSY. NO (8 Shoe-Strand or Section)	w/pads 7952167 w/o pads 7952188	8339205	7737225 (Track Assy.)	7360701	7379874	7985692		7355050
TRACK	WEIGHT (lb): SHOE (Pin Track) SECTION (Band Track)	41 w/pad pad 5	44.6 w/pad pad 5	39.1	41	61.4	61.4	61.4	58.9
	PITCH (in.)	6	6	6	8	6-15/16	6-15/16	6-15/18	6-15/16
	LENGTH OF SECTION (Band Track) (in.)	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.	Does not apply.
	WIDTH (in.)	21	21	21	21	28	28	28	28
	GROUSER CONTACT AREA (sq in.)	pad 41.82	pad 41.8	35.25	48.3	Does not apply.	Does not apply.	Does not apply.	48.1
	GROUSER HEIGHT (in.)	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2
	PIN DIA. (in.)	0.878 octagonal	0.878 octagonal	0.878 octagonal	0.878 octagonal	1.250	1.250	1.250	1.250
	GUIDE TYPE	Integral center	Integral center	Integral center	Integral center	Removable center	Removable center	Removable center	Removable center
BUSHING	TYPE	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber	Rubber
	PROJECTED AREA (in.)	14.43	14.43	10.280	14.43	27.500	27.500	27.500	27.500
	FLEXIBLE ANGLE (deg.) (At Assembly)	7-1/2	7-1/2	8	7-1/2	7	7	7	7
	RUBBER ID (in.)	1.130	1.130	1.130	1.135	1.250	1.250	1.250	1.250
	WALL THICKNESS (in.)	0.187	0.187	0.187	0.187	0.187	0.187	0.187	0.187
	% STRAIN	14.0	10.5	9.9	10.5	14	14	14	14

TABLE A-20
TRACK DATA

	MODEL	T97E1	T97E2	T102	T102E1	T112
	VEHICLE APPLICATION	Heavy Tank, T43 Series, T54E2 Medium Tank, M48	Heavy Tank, T43 Medium Tank, M48A1, M48A2, M48E1 Recovery Vehicle, T88	Amphibious Cargo Carrier, T46, T46E1, M76 Trailer, T79	Amphibious, T46E1, M76 Trailer, T79	Amphibious Cargo Carrier, T60
	DESCRIPTION	Same as Model T97 Track except for wedge design change.	Same as Model T97 Track except for end connector and wedge design changes.	Siamese Track, Rubber and Steel Cable Band, Sectionalized, Steel Stamped Cross Bars.	Siamese Track, Rubber and Steel Cable Band, Steel Welded Cross Bars.	Spaced Linked Band Track, Steel and Rubber, Steel Grousers.
	ASSY. NO. (1 Shoe)	D7985689	8686333	Does not apply.	Does not apply.	Does not apply.
	ASSY. NO. (8 Shoe-Strand or Section)	7985690	8686334	K7965168	K7976863	SK-15599
TRACK	WEIGHT (lb): SHOE (Pin Track) SECTION (Band Track)	58.4	58.9	110		
TRACK	PITCH (in.)	6-15/16	6-15/16	4	4	5
TRACK	LENGTH OF SECTION (Band Track) (in.)	Does not apply.	Does not apply.	44	44	40
TRACK	WIDTH (in.)	28	28	30	30	30
TRACK	GROUSER CONTACT AREA (sq in.)	48.1	48.1	Does not apply.	Does not apply.	Does not apply.
TRACK	GROUSER HEIGHT (in.)	1-1/2	1-1/2	5/8	5/8	
TRACK	PIN DIA. (in.)	1.250	1.250	Does not apply.	Does not apply.	Does not apply.
TRACK	GUIDE TYPE	Removable center	Removable center	Outside fixed to cross bar.	Outside fixed to cross bar.	Integral outer
BUSHING	TYPE	Rubber	Rubber	Does not apply.	Does not apply.	Does not apply.
BUSHING	PROJECTED AREA (in.)	27.500	27.500	Does not apply.	Does not apply.	Does not apply.
BUSHING	FLEXIBLE ANGLE (deg.) (At Assembly)	8	7	Does not apply.	Does not apply.	Does not apply.
BUSHING	RUBBER ID (in.)	1.250	1.250	Does not apply.	Does not apply.	Does not apply.
BUSHING	WALL THICKNESS (in.)	0.187	0.187	Does not apply.	Does not apply.	Does not apply.
BUSHING	% STRAIN	20	14	Does not apply.	Does not apply.	Does not apply.

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ENGINEERING DESIGN HANDBOOK SERIES

Listed below are the Handbooks which have been published or are currently being printed. Handbooks with publication dates prior to 1 August 1962 were published as 20-series Ordnance Corps pamphlets. AMC Circular 310-38, 19 July 1963, redesignated those publications as 706-series AMC pamphlets (i.e., ORDP 20-138 was redesignated AMCP 706-138). All new, reprinted, or revised Handbooks are being published as 706-series AMC pamphlets.

General and Miscellaneous Subjects

No.	Title
106	Elements of Armament Engineering, Part One, Sources of Energy
107	Elements of Armament Engineering, Part Two, Ballistics
108	Elements of Armament Engineering, Part Three, Weapon Systems and Components
110	Experimental Statistics, Section 1, Basic Concepts and Analysis of Measurement Data
111	Experimental Statistics, Section 2, Analysis of Enumerative and Classificatory Data
112	Experimental Statistics, Section 3, Planning and Analysis of Comparative Experiments
113	Experimental Statistics, Section 4, Special Topics
114	Experimental Statistics, Section 5, Tables
121	Packaging and Pack Engineering
134	Maintainability Guide for Design
135	Inventions, Patents, and Related Matters (Revised)
136	Servomechanisms, Section 1, Theory
137	Servomechanisms, Section 2, Measurement and Signal Converters
138	Servomechanisms, Section 3, Amplification
139	Servomechanisms, Section 4, Power Elements and System Design
170(C)	Armor and Its Application to Vehicles (U)
270	Propellant Actuated Devices
290(C)	Warheads--General (U)
331	Compensating Elements (Fire Control Series)

Ammunition and Explosives Series

No.	Title
175	Solid Propellants, Part One
176(C)	Solid Propellants, Part Two (U)
177	Properties of Explosives of Military Interest, Section 1
178(C)	Properties of Explosives of Military Interest, Section 2 (U)
179	Explosive Trains
210	Fuzes, General and Mechanical
211(C)	Fuzes, Proximity, Electrical, Part One (U)
212(S)	Fuzes, Proximity, Electrical, Part Two (U)
213(S)	Fuzes, Proximity, Electrical, Part Three (U)
214(S)	Fuzes, Proximity, Electrical, Part Four (U)
215(C)	Fuzes, Proximity, Electrical, Part Five (U)
242	Design for Control of Projectile Flight Characteristics
244	Section 1, Artillery Ammunition--General, with Table of Contents, Glossary and Index for Series
245(C)	Section 2, Design for Terminal Effects (U)
246	Section 3, Design for Control of Flight Characteristics *(out of print)*
247	Section 4, Design for Projection
248	Section 5, Inspection Aspects of Artillery Ammunition Design
249	Section 6, Manufacture of Metallic Components of Artillery Ammunition

Automotive Series

No.	Title
355	The Automotive Assembly
356	Automotive Suspensions

Ballistic Missile Series

No.	Title
281(S-RD)	Weapon System Effectiveness (U)
282	Propulsion and Propellants

Ballistic Missile Series (continued)

No.	Title
283	Aerodynamics
284(C)	Trajectories (U)
286	Structures

Ballistics Series

No.	Title
140	Trajectories, Differential Effects, and Data for Projectiles
150	Interior Ballistics of Guns
160(S)	Elements of Terminal Ballistics, Part One, Introduction, Kill Mechanisms, and Vulnerability (U)
161(S)	Elements of Terminal Ballistics, Part Two, Collection and Analysis of Data Concerning Targets (U)
162(S-RD)	Elements of Terminal Ballistics, Part Three, Application to Missile and Space Targets (U)

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341	Cradles
342	Recoil Systems
343	Top Carriages
344	Bottom Carriages
345	Equilibrators
346	Elevating Mechanisms
347	Traversing Mechanisms

Guns Series

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252	Gun Tubes

Military Pyrotechnics Series

No.	Title
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187	Part Three, Properties of Materials Used in Pyrotechnic Compositions
189	Part Five, Bibliography

Surface-to-Air Missile Series

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292	Part Two, Weapon Control
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294(S)	Part Four, Missile Armament (U)
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