Handbook of Reliability Prediction Procedures for Mechanical Equipment

Logistics Technology Support

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Recognition of reliability and maintainability (R&M) as vital factors in the development, production, operation, and maintenance of today's complex systems has placed greater emphasis on the application of design evaluation techniques to logistics management. An analysis of a design for reliability and maintainability can identify critical failure modes and causes of unreliability and provide an effective tool for predicting equipment behavior and selecting appropriate logistics measures to assure satisfactory performance. Application of design evaluation techniques can provide a sound basis for determining spare parts requirements, required part improvement programs, needed redesign efforts, reallocation of resources and other logistics measures to assure that specified reliability and maintainability requirements will be met.

Many efforts have been applied toward duplicating the data bank approach or developing a new approach for mechanical equipment. The statistical analysis of equipment aging characteristics, regression techniques of equipment operating parameters related to failure rates, and analysis of field failure data have been studied in attempts to develop a methodology that can be used to evaluate a new mechanical design for R&M characteristics.

Many of the attempts to develop R&M prediction methodology have been at a system or subsystem level. The large number of variables at these levels and lack of detailed knowledge regarding operating environment have created a problem in applying the results to the design being evaluated. Attempts to collect failure rate data or develop an R&M prediction methodology at the system or subsystem level produce a wide dispersion of failure rates for apparently similar components because of the basic characteristics of mechanical components.

The Design Evaluation Techniques program was initiated by the Carderock Division of the Naval Surface Warfare Center (CDNSWC) and was sponsored by the Office of Naval Technology under the Logistics Exploratory Development Program, P.E. 62233N. The methodology for predicting R&M characteristics as part of this development effort does not rely solely on failure rate data. Instead, the design evaluation procedures consider the material properties, operating environment and critical failure modes at the component part level to evaluate a design for R&M. The purpose of this Handbook is to present the proposed methodology for predicting the reliability of mechanical equipment and solicit comments as to the potential utility of a standard reference for reliability predictions of mechanical equipment.

The development of this Handbook by the Logistics Technology Support Group (Code 2120) of CDNSWC was coordinated with the military, industry and academia. Sponsors of this effort included the U. S. Army Armament Research, Development &
Engineering Center (SMCAR-QAH-P), Picatinny Arsenal and the Robins AFB, WR-ALC/LVRS. These sponsors have provided valuable technical guidance in the development of the methodology and Handbook. Chapter 1 of the Handbook provides a summary of the testing program to validate the prediction methodology. Also, the Robins AFB supplied an MC-2A Air Compressor Unit for validation testing purposes. The procedures contained in this Handbook were used to predict the failure modes of the MC-2A and their frequency of occurrence. Reliability tests were then performed with a close correlation between predicted and actual reliability being achieved. Past sponsors and participants in the program include the Belvoir Research, Development, & Engineering Center; Wright-Patterson AFB; Naval Sea Systems Command; Naval Air Test Center and Louisiana Tech University.

Previous editions of this Handbook were distributed to interested engineering personnel in industry and DoD for comments as to the utility of the methodology in evaluating mechanical designs for reliability. The comments have been extremely useful in improving the prediction methodology and contents of the Handbook. The revised Handbook is available at no charge and can be downloaded by visiting the CDNSWC website (www.dt.navy.mil). Every effort has been made to validate the equations presented in this Handbook. However, limited funding has prevented the extensive testing and application of prediction procedures to the design/procurement process for full validation of the approach. Therefore, users are cautioned that this Handbook is the result of a research program and not an official DoD document.

Several companies have chosen to produce software packages containing the material in this Handbook, the attempt being to sell a software package whereby the reliability of mechanical components can be predicted in the same way as electronic components. The Navy has not been and is not now in any way connected with the commercial ventures to produce software packages. As described previously, it is important to understand the difference between the failure rate data used to evaluate electronic equipment and the procedures used to evaluate mechanical equipment. For a company to extract equations from the Handbook without regard to the application procedures is in violation of the intent of the Handbook, the result being a potentially dangerous situation for the user in logistically relying on inaccurate results. Another result is the damaging reputation to CDNSWC and the Navy in their attempts to improve the reliability of mechanical equipment through a greater understanding of mechanical system design. To extract equations from the Handbook without regard to the procedures and parameter limits defeats the purpose of the Handbook in helping the designer of mechanical systems gain a greater insight as to the reliability of his design.

CDNSWC has developed a software package that automates the use of procedures and equations in the Handbook that can be used to evaluate the methodology. This software program called MechRel can be downloaded free of charge by visiting the CDNSWC website. In summary, the Handbook and associated software package representing many years of research and development are already available at no charge. Commercial exploitation of this work by extracting material without the full
content of the evaluation procedures violates the purpose of the work being done by CDNSWC. Any product sold using material from the Handbook or referencing the Handbook must contain a statement that CDNSWC and the Navy have not participated in the development of or approve of their product.

Interested users of the technology presented in this Handbook are urged to contact CDNSWC to obtain the latest available information on mechanical reliability. Comments and recommended changes to the Handbook should be addressed to:

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CHAPTER 1
INTRODUCTION

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1.1 PREFACE

The "Handbook of Reliability Prediction Procedures for Mechanical Equipment" has been developed by the Logistics Technology Support Group, Carderock Division, Naval Surface Warfare Center (CDNSWC) in Bethesda, Maryland. The handbook presents a new approach for determining the reliability and maintainability (R&M) characteristics of mechanical equipment. It has been developed to help the user identify equipment failure modes and potential causes of unreliability in the early design phases of equipment development, and then to quantitatively evaluate the design for R&M and determine logistics support requirements.

A software program called "MechRel" has also been developed by the Logistics Technology Support Group to automate the Handbook procedures and equations. The Handbook and MechRel software program are available free of charge from the Carderock Division, Naval Surface Warfare Center. Contact information is contained in Section 1.6

1.2 CURRENT METHODS OF PREDICTING RELIABILITY

A reliability prediction is performed in the early stages of a development program to support the design process. Performing a reliability prediction provides for visibility of
equipment reliability requirements in the early development phase. A well done prediction also provides an awareness of potential equipment degradation during the equipment life cycle. As a result of performing a reliability prediction, equipment designs can be improved, costly over-designs prevented and development testing time optimized.

Performance of a reliability prediction for electronic equipment is well established by research and development. For example, MIL-HDBK-217 has been developed for predicting the reliability of electronic equipment. Development of this document was made possible because the standardization and mass production of electronic parts has permitted the creation of valid failure rate data banks for high population electronic devices. Such extensive sources of quality and reliability information can be used directly to predict operational reliability while the electronic design is still on the drawing board.

A commonly accepted method for predicting the reliability of mechanical equipment based on a data bank has not been possible because of the wide dispersion of failure rates which occur for apparently similar components. Inconsistencies in failure rates for mechanical equipment are the result of several basic characteristics of mechanical components:

a. Individual mechanical components such as valves and gearboxes often perform more than one function and failure data for specific applications of nonstandard components are seldom available. A hydraulic valve for example may contain a manual shut-off feature as well as an automatic control mechanism on the same valve structure.

b. Failure rates of mechanical components are not usually described by a constant failure rate distribution because of wear, fatigue and other stress-related failure mechanisms resulting in equipment degradation. Data gathering is complicated when the constant failure rate distribution can not be assumed and individual times to failure must be recorded in addition to total operating hours and total failures.

c. Mechanical equipment reliability is more sensitive to loading, operating mode and utilization rate than electronic equipment reliability. Failure rate data based on operating time alone are usually inadequate for a reliability prediction of mechanical equipment.

d. Definition of failure for mechanical equipment depends upon its application. For example, failure due to excessive noise or leakage can not be universally established. Leakage requirements for a water system are obviously different than those for a fuel system. Lack of such information in a failure rate data bank limits its usefulness.

The above deficiencies in a failure rate data base result in problems in applying published failure rates to an actual design analysis. The most commonly used tools for
determining the reliability characteristics of a mechanical design can result in a useful listing of component failure modes, system level effects, critical safety related issues, and projected maintenance actions. However, estimating the design life of mechanical equipment is a difficult task for the design engineer. Many life-limiting failure modes such as corrosion, erosion, creep, and fatigue operate on the component at the same time and have a synergistic effect on reliability. Also, the loading on the component may be static, cyclic, or dynamic at different points during the life cycle and the severity of loading may also be a variable. Material variability and the inability to establish an effective data base of historical operating conditions such as operating pressure, temperature, and vibration further complicate life estimates.

Although several analytical tools such as the Failure Modes, Effects and Criticality Analysis (FMECA) are available to the engineer, they have been developed primarily for electronic equipment evaluations, and their application to mechanical equipment has had limited success. The FMECA, for example, is a very powerful technique for identifying equipment failure modes, their causes, and the effect each failure mode will have on system performance. Results of the FMECA provide the engineer with a valuable insight as to how the equipment will fail; however, the problem in completing the FMECA for mechanical components is determining the probability of occurrence for each identified failure mode.

The above listed problems associated with acquiring failure rate data for mechanical components demonstrates the need for reliability prediction models that do not rely solely on existing failure rate data banks. Predicting the reliability of mechanical equipment requires the consideration of its exposure to the environment and subjection to a wide range of stress levels such as impact loading. The approach to predicting reliability of mechanical equipment presented in this Handbook considers the intended operating environment and determines the effect of that environment at the lowest part level where the material properties can also be considered. The combination of these factors permits the use of engineering design parameters to determine the design life of the equipment in its intended operating environment and the rate and pattern of failures during the design life.

1.3 DEVELOPMENT OF THE HANDBOOK

Useful models must provide the capability of predicting the reliability of all types of mechanical equipment by specific failure mode considering the operating environment, the effects of wear and other potential causes of degradation. The models developed for the Handbook are based upon identified failure modes and their causes. The first step in developing the models was the derivation of equations for each failure mode from design information and experimental data as contained in published technical reports and journals. These equations were simplified to retain those variables affecting reliability as indicated from field experience data. Modification factors were then compiled for each variable to reflect its quantitative impact on the failure rate of an individual component part. The total failure rate of the component is the sum of the
failure rates for the component parts for a particular time period in question. Failure rate equations for each component part, the methods used to generate the models in terms of failures per hour or failures per cycle and the limitations of the models are discussed in each chapter of the Handbook. The equations and procedures were validated to the extent possible with laboratory testing or engineering analysis.

The objective of the Handbook and MechRel software program is to provide procedures which can be used for the following elements of a reliability program:

- Evaluate designs for reliability in the early stages of development
- Provide management emphasis on reliability with standardized evaluation procedures
- Provide an early estimate of potential spare parts requirements
- Quantify critical failure modes for initiation of specific stress or design analyses
- Provide a relative indication of reliability for performing trade off studies, selecting an optimum design concept or evaluating a proposed design change
- Determine the degree of degradation with time for a particular component or potential failure mode
- Design accelerated testing procedures for verification of reliability performance

One of the problems any engineer can have in evaluating a design for reliability is attempting to predict performance at the system level. The problem of predicting the reliability of mechanical equipment is easier at the lower indenture levels where a clearer understanding of design details affecting reliability can be achieved. Predicting the life of a mechanical component, for example, can be accomplished by considering the specific wear, erosion, fatigue and other deteriorating failure mechanism, the lubrication being used, contaminants which may be present, loading between the surfaces in contact, sliding velocity, area of contact, hardness of the surfaces, and material properties. All of these variables would be difficult to record in a failure rate data bank; however, the derivation of such data can be achieved for individual designs and the potential operating environment can be brought down through the system level and the effects of the environmental conditions determined at the part level.

The development of design evaluation procedures for mechanical equipment includes mathematical equations to estimate the design life of mechanical components. These reliability equations consider the design parameters, environmental extremes, and operational stresses to predict the reliability parameters. The equations rely on a base failure rate derived from laboratory test data where the exact stress levels are known. Engineering equations are used to modify this failure rate to the appropriate stress/strength and environmental relationships for the equipment application. Figure 1.1 illustrates the method of considering the effects of the environment and the operating stresses at the lowest indenture level.
A component such as a valve assembly may consist of seals, springs, fittings, and the valve housing. The design life of the entire mechanical system is accomplished by evaluating the design at the component and part levels considering the material properties of each part. The operating environment of the system is included in the equations by determining its impact at the part level. Some of the component parts may not have a constant failure rate as a function of time and the total system failure rate of the system can be obtained by adding part failure rates for the time period in question.

Figure 1.1 Mechanical Components and Parts

Many of the parts are subject to wear and other deteriorating type failure mechanisms and the reliability equations must include the parameters which are readily accessible to the equipment designer. As part of this research project, Louisiana Tech University was tasked to establish an engineering model for mechanical wear which is correlated to the material strength and stress imposed on the part. This model for predicting wear considers the materials involved, the lubrication properties, the stress imposed on the part and other aspects of the wear process (Reference 72). The relationship between the material properties and the wear rate was used to establish generalized wear life equations for actuator assemblies and other components subject to surface wear.
In another research project, lubricated and unlubricated spline couplings were operated under controlled angular misalignment and loading conditions to provide empirical data to verify spline coupling life prediction models. This research effort was conducted at the Naval Air Warfare Center in Patuxent River, Maryland (Reference 71). A special rotating mechanical coupling test machine was developed for use in generating reliability data under controlled operating conditions. This high-speed closed loop testbed was used to establish the relationships between the type and volume of lubricating grease employed in the spline coupling and gear life. Additional tests determined the effects of material hardness, torque, rotational speed and angular misalignment on gear life.

Results of these wear research projects were used to develop and refine the reliability equations for those components subject to wear.

1.4 EXAMPLE DESIGN EVALUATION PROCEDURE

A hydraulic valve assembly will be used to illustrate the Handbook approach to predicting the reliability of mechanical equipment. An example diagram of a valve assembly is shown in Figure 1.2. Developing reliability equations for all the different types of hydraulic valves would be an impossible task since there are over one hundred different types of valve assemblies available. For example, some valves are named for the function they perform, e.g. check valve, regulator valve and unloader valve. Others are named for a distinguishing design feature, e.g. globe valve, needle valve, solenoid valve. However, from a reliability standpoint, dropping down one indenture level provides two basic types of valve assemblies: the poppet valve and the sliding action valve.

The example assembly chosen for analysis is a poppet valve which consists of a poppet assembly, spring, seals, guide and housing.

![Figure 1.2 Valve Assembly](image-url)
1.4.1 Poppet Assembly

The functions of the poppet valve would indicate the primary failure mode as incomplete closure of the valve resulting in leakage around the poppet seat. This failure mode can be caused by contaminants being wedged between the poppet and seat, wear of the poppet seat, and corrosion of the poppet/seat combination. External seal leakage, sticking valve stem, and damaged poppet return spring are other failure modes which must be considered in the design life of the valve.

A new poppet assembly may be expected to have a sufficiently smooth surface for the valve to meet internal leakage specifications. However, after some period of time contaminants will cause wear of the poppet assembly until leakage rate is beyond tolerance. This leakage rate, at which point the valve is considered to have failed, will depend on the application and to what extent leakage can be tolerated.

As derived in Chapter 6 of the Handbook, the following equation can be used to determine the failure rate of a poppet assembly:

\[
\lambda_p = \lambda_{p,B} \frac{2 \times 10^2 D_M f^3 \left(P_1^2 - P_2^2\right) K_1}{Q_f \nu_a L_W \left(S_S\right)^{1.5}}
\]

Where:
- \( \lambda_p \) = Failure rate of the poppet assembly, failures/million cycles
- \( \lambda_{p,B} \) = Base failure rate for poppet assembly, failures/million cycles
- \( D_M \) = Mean seat diameter, in
- \( f \) = Mean surface finish of opposing surfaces, in
- \( P_1 \) = Upstream pressure, lbs/in\(^2\)
- \( P_2 \) = Downstream pressure, lbs/in\(^2\)
- \( K_1 \) = Constant which considers the impact of contaminant size, hardness and quantity of particles
- \( Q_f \) = Leakage rate considered to be a valve failure, in\(^3\)/min
- \( \nu_a \) = Absolute fluid viscosity, lb-min/in\(^2\)
- \( L_W \) = Radial seat land width, in
- \( S_S \) = Apparent seat stress, lb/in\(^2\)
Values used to determine the failure rates for the parts used in this example are listed in Table 1-1. Throughout the Handbook, failure rate equations for each component and part are translated into a base failure rate with a series of multiplying factors to modify the base failure rate to the operating environment being considered. For example, as shown in Equation (6-6) of Chapter 6, the above equation can be rewritten as follows:

\[
\lambda_{PO} = \lambda_{PO,B} \cdot C_P \cdot C_Q \cdot C_F \cdot C_V \cdot C_N \cdot C_S \cdot C_{DT} \cdot C_{SW} \cdot C_W
\]

Where:

- \( \lambda_{PO} \) = Failure rate of poppet assembly in failures/million operations
- \( \lambda_{PO,B} \) = Base failure rate of poppet assembly, 1.40 failures/million operations
- \( C_P \) = Multiplying factor which considers the effect of fluid pressure on the base failure rate
- \( C_Q \) = Multiplying factor which considers the effect of allowable leakage on the base failure rate
- \( C_F \) = Multiplying factor which considers the effect of surface finish on the base failure rate
- \( C_V \) = Multiplying factor which considers the effect of fluid viscosity on the base failure rate
- \( C_N \) = Multiplying factor which considers the effect of contaminants on the base failure rate
- \( C_S \) = Multiplying factor which considers the effect of seat stress on the base failure rate
- \( C_{DT} \) = Multiplying factor which considers the effect of seat diameter on the base failure rate
- \( C_{SW} \) = Multiplying factor which considers the effect of seat land width on the base failure rate
- \( C_W \) = Multiplying factor which considers the effect of fluid flow rate on the base failure rate

The parameters in the failure rate equation can be located on an engineering drawing, by knowledge of design standards or by actual measurement. Other design parameters which have a minor effect on reliability are included in the base failure rate as determined from field performance data.
1.4.2 Spring Assembly

Depending on the application, a spring may be in a static, cyclic, or dynamic operating mode. In the current example of a valve assembly, the spring will be in a cyclic mode. The operating life of a mechanical spring arrangement is dependent upon the susceptibility of the materials to corrosion and stress levels (static, cyclic or dynamic). The most common failure modes for springs include fracture due to fatigue and excessive loss of load due to stress relaxation. Other failure mechanisms and causes may be identified for a specific application. Typical failure rate considerations include: level of loading, operating temperature, cycling rate and corrosiveness of the fluid environment. Other failure modes to be considered are listed in Chapter 4.

The failure rate of a compression spring depends upon the stress on the spring and the relaxation properties of the material. The load on the spring is equal to the spring rate multiplied by the deflection and calculated as explained in Chapter 4.

\[
P_L = R(L_1 - L_2) = \frac{G_M(D_w)^4(L_1 - L_2)}{8(D_C)^3 N_a}
\]

Where:
- \(P_L\) = Load, lbs
- \(R\) = Spring rate, lb/in
- \(L_1\) = Initial deflection of spring, in
- \(L_2\) = Final deflection of spring, in
- \(G_M\) = Modulus of rigidity, lb/in²
- \(D_w\) = Mean diameter of wire, in
- \(D_C\) = Mean diameter of spring, in
- \(N_a\) = Number of active coils

Stress in the spring will be proportional to loading according to the following relationship:

\[
S_G = \frac{8 P_L D_C K_w}{\pi D_w^3}
\]

Where:
- \(S_G\) = Actual stress, psi
$K_W = \text{Wahl stress correction factor}$

This equation permits determination of expected life of the spring by plotting the material S-N curve on a modified Goodman diagram. In the example valve application, the spring force and the failure rate remain constant. This projection is valid if the spring does not encounter temperature extremes. Corrosion is a critical factor in spring design because most springs are made of steel which is susceptible to a corrosive environment. In this example the fluid medium is assumed to be non-corrosive and the spring is always surrounded by the fluid, thus a corrosion factor need not be included in this analysis. If the valve were a safety device and subjected intermittently to a steam environment, then a corrosion factor would have to be applied consistent with any corrosion protection in the original spring design.

The failure rate of the compression spring can be estimated from the following equation:

$$\lambda_{SP} = \lambda_{SP,B} \left(\frac{S_G}{T_S}\right)^3 = \lambda_{SP,B} \left(\frac{8 P_L D_c K_W}{\pi T_S D_w^3}\right)^3$$

where:

$T_S = \text{Material tensile strength, lbs/in}^2$

Other multiplying factors based on field performance data are detailed in Chapter 4.

1.4.3 Seal Assembly

The primary failure mode of a seal is leakage, and the following equation as derived in Chapter 3 uses a similar approach as developed for evaluating a poppet design:

$$\lambda_{SE} = \lambda_{SE,B} \frac{K_i \left(\frac{P_1^2 - P_2^2}{Q_f V_a P_2}\right)}{r_o + r_i} \cdot \frac{r_o - r_i}{H^3}$$

Where:

$\lambda_{SE} = \text{Failure rate of seal, failures/million cycles}$

$\lambda_{SE,B} = \text{Base failure rate of seal, failures/million cycles}$

$K_i = \text{Constant} = 3.27 \times 10^{-4}$
\[ P_1 = \text{System pressure, lb/in}^2 \]
\[ P_2 = \text{Standard atmospheric pressure or downstream pressure, lb/in}^2 \]
\[ Q_f = \text{Allowable leakage rate under conditions of usage, in}^3/\text{min} \]
\[ \nu_a = \text{Absolute fluid viscosity, lb-min/in}^2 \]
\[ r_i = \text{Inside radius of circular interface, in} \]
\[ r_o = \text{Outside radius of circular interface, in} \]
\[ H = \text{Conductance parameter (Meyer hardness } M; \text{ contact pressure } C; \text{ surface finish } f), \text{ in} \]

The conductance parameter is a combination of Meyer hardness, contact pressure and surface finish per the following equation:

\[ H = 0.23 \left( \frac{M}{C} \right)^{1.5} \cdot f^{2/3} \]

Where:

\[ M = \text{Meyer hardness (or Young's modulus) for rubber and resilient materials, lbs/in}^2 \]
\[ C = \text{Contact stress, lbs/in}^2 \]
\[ f = \text{Surface finish, in} \]

In the case of an O-ring seal, the failure rate will increase as a function of time because of gradual hardening of the rubber material. A typical failure rate curve for an O-ring is shown in Figure 1.2. Multiplying factors considering such parameters as fluid temperature are detailed in Chapter 3.

1.4.4 Combination of Failure Rates

The addition of failure rates to determine the total valve failure rate depends on the life of the valve and the maintenance philosophy established. If the valve is to be discarded upon the first failure, a time-to-failure can be calculated for the particular operating environment. If, on the other hand, the valve will be repaired upon failure with the failed part(s) being replaced, then the failure rates must be combined for different time phases throughout the life expectancy until the wear-out phase has been reached. The effect of part replacement and overhaul is a tendency toward a constant failure rate at the system level and will have to be considered in the prediction for the total system.
The housing will exhibit an insignificant failure rate, usually verified by experience or by finite element analysis. Typical values as assumed for the example equations are listed in Table 1-1.

After the failure rates are determined for each component part, the rates are summed to determine the failure rate of the total valve assembly. Because some of the parameters in the failure rate equation are time dependent, i.e. the failure rate changes as a function of time, the total failure rate must be determined for particular intervals of time. In the example of the poppet assembly, nickel plating was assumed with an initial surface finish of 35 μ inches. The change in surface finish over a one year time period for non-acidic fluids such as water, mild sodium chloride solutions, and hydraulic fluids will be a deterioration to 55 μ inches. In the case of the O-ring seal, the hardness of the rubber material will change with age. The anticipated failure rate as a function of time for the component parts of the valve and the total valve assembly are shown in Figure 1.3.

### Table 1-1. Typical Values for Failure Rate Equations

<table>
<thead>
<tr>
<th></th>
<th>POPPET</th>
<th>SPRING</th>
<th>SEAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>PARAMETER</td>
<td>VALUE</td>
<td>PARAMETER</td>
<td>VALUE</td>
</tr>
<tr>
<td>λ_{P,B}</td>
<td>1.40</td>
<td>λ_{SP,B}</td>
<td>23.8</td>
</tr>
<tr>
<td>Q_r</td>
<td>0.06</td>
<td>L_1</td>
<td>3.35</td>
</tr>
<tr>
<td>D_M</td>
<td>1.69</td>
<td>L_2</td>
<td>2.28</td>
</tr>
<tr>
<td>F *</td>
<td>35 E-6</td>
<td>G_M</td>
<td>11.5 E 6</td>
</tr>
<tr>
<td>P_1</td>
<td>3000</td>
<td>D_C</td>
<td>0.58</td>
</tr>
<tr>
<td>P_2</td>
<td>15.0</td>
<td>D_W</td>
<td>0.085</td>
</tr>
<tr>
<td>ν_a</td>
<td>2 E-8</td>
<td>N_a</td>
<td>14</td>
</tr>
<tr>
<td>L_W</td>
<td>0.85</td>
<td>T_S</td>
<td>245 E3</td>
</tr>
<tr>
<td>S_s</td>
<td>4045</td>
<td>P_L</td>
<td>29.4</td>
</tr>
<tr>
<td>K_1</td>
<td>1.00</td>
<td>S_G</td>
<td>86.2 E 3</td>
</tr>
<tr>
<td>Ops/hour</td>
<td>0.5</td>
<td>K_W</td>
<td>1.219</td>
</tr>
<tr>
<td>TOTALS:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>λ_P</td>
<td>0.35</td>
<td>λ_{SP}</td>
<td>1.04</td>
</tr>
</tbody>
</table>

* Initial value = 35 μin; after 8,000 operating hours (4,000 operations) surface finish will equal 55 μin (Reference 5)
** Initial value = 0.55 (hardness, M = 500 psi; contact stress, C = 910 psi); after 1 year M estimated to be 575 psi (M/C = 0.63)
1.5 VALIDATION OF RELIABILITY PREDICTION EQUATIONS

A very limited budget during the development of this handbook prevented the procurement of a sufficiently large number of components to perform the necessary failure rate tests for all the possible combinations of loading roughness, operational environments, and design parameters to reach statistical conclusions as to the accuracy of the reliability equations. Instead, several test programs were conducted to verify the identity of failure modes and validate the engineering approach being taken to develop the reliability equations. For example, valve assemblies were procured and tested at the Belvoir Research, Development and Engineering Center in Ft. Belvoir, Virginia. The number of failures for each test was predicted using the equations presented in this handbook. Failure rate tests were performed for several combinations of stress levels and results compared to predictions. Typical results are shown in Table 1-2.
Table 1-2. Sample Test Data for Validation of Reliability Equations for Valve Assemblies

<table>
<thead>
<tr>
<th>TEST SERIES</th>
<th>VALVE NUMBER</th>
<th>TEST CYCLES TO FAILURE</th>
<th>ACTUAL FAILURES/10^6 CYCLES</th>
<th>AVERAGE FAILURES/10^6 CYCLES</th>
<th>PREDICTED FAILURES/10^6 CYCLES</th>
<th>FAILURE MODE #</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>11</td>
<td>68,322</td>
<td>14.64</td>
<td>14.64</td>
<td>18.02</td>
<td>3</td>
</tr>
<tr>
<td>24</td>
<td>8</td>
<td>257,827</td>
<td></td>
<td></td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>24</td>
<td>9</td>
<td>131,126</td>
<td>7.63</td>
<td>10.15</td>
<td>10.82</td>
<td>1</td>
</tr>
<tr>
<td>24</td>
<td>10</td>
<td>81,113</td>
<td>12.33</td>
<td></td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>24</td>
<td>11</td>
<td>104</td>
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<td>12</td>
<td>110,488</td>
<td>9.05</td>
<td></td>
<td></td>
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</tr>
<tr>
<td>24</td>
<td>13</td>
<td>86,285</td>
<td>11.59</td>
<td></td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>25</td>
<td>14</td>
<td>46,879</td>
<td>21.33</td>
<td>19.67</td>
<td>8.45</td>
<td>2</td>
</tr>
<tr>
<td>25</td>
<td>15</td>
<td>300</td>
<td></td>
<td></td>
<td></td>
<td>3</td>
</tr>
<tr>
<td>25</td>
<td>18</td>
<td>55,545</td>
<td>18.00</td>
<td></td>
<td></td>
<td>1</td>
</tr>
</tbody>
</table>

TEST PARAMETERS:
- SYSTEM PRESSURE: 3500 psi
- FLUID FLOW: 100% rated
- FLUID TEMPERATURE: 90 C
- FLUID: Hydraulic, MIL-H-83282

FAILURE MODE:
1 - Spring Fatigue
2 - No Apparent
3 - Accumulated Debris

Another example of reliability tests performed during development of the handbook is the testing of gearbox assemblies at the Naval Air Warfare Center in Patuxent River, Maryland (Reference 70). A spiral-bevel right angle reducer type gearbox with 3/8 inch steel shaft was selected for the test. Two models having different speed ratios were chosen, one gearbox rated at 12 in-lbs torque at 3600 rpm and the other gearbox rated at 9.5 in-lbs torque. Prior to testing the gearboxes, failure rate calculations were made using the reliability equations from this handbook. Test results were compared with failure rate calculations and conclusions made concerning the ability of the equations to be used in calculating failure rates.

Reliability tests were also performed on stock hydraulic actuators using a special-purpose actuator wear test apparatus (Reference 72). The actuators used in this validation project had a 2.50 inch bore, a 5.0 inch stroke, and a nominal operating...
pressure of 3000 psig. Various loads and lubricants were used to correlate test results with Handbook prediction procedures and equations. The effect of contamination of the oil was correlated by adding 10 micron abrasive particles to the lubricant in the actuators.

Additional reliability tests were performed during development of the handbook on air compressors for 4000 hours under six different environmental conditions to correlate the effect of the environment on mechanical reliability (Reference 73). The air compressors procured for the test were small reciprocating compressors with a maximum pressure of 35 psi and a ft³ rating of 0.35. The units were subjected to temperature extremes, blowing dust, and AC line voltage variations while operating at maximum output pressure. The data collected were used to verify the reliability equations for reciprocating compressors.

In another reliability test, a special environmentally controlled test chamber was constructed at the Naval Air Warfare Center in Patuxent River, Maryland to test gear pumps and centrifugal pumps (References 74 and 75). A series of bronze rotary gear pumps were operated for 8000 hours to collect data on operation under controlled hydraulic conditions. Tests were conducted under high temperature water, low temperature water, and water containing silicon dioxide abrasives. Data were collected on flow rates, and seal leakage while pump speed, output pressure, and fluid temperature were held constant. Similar tests were conducted on a series of centrifugal pumps.

To further evaluate wear mechanisms and their effect on mechanical reliability, fifteen impact wrenches were operated to failure with a drum brake providing frictional torque and inertial torque loading (Reference 76). The impact wrenches selected for testing were general purpose, 1/2 inch drive, pneumatic impact wrenches commonly found in Naval repair shops. This wrench is rated for 200 lb-ft of torque and uses 4 cfm at 90 psi of air. Results of these reliability tests were used to evaluate the utility of the related failure equations in the handbook.

Validation of the various reliability equations for brakes and clutches was accomplished with tests conducted at Louisiana Tech University by evaluating the wear process for the various elements used in disk and drum brakes and multiple-disk clutches (Reference 77). Two types of experimental tests were conducted in connection with development of the model: (a) abrasive wear tests and (b) measurements of the coefficient of friction. Brakes and clutches were tested while monitoring the rate of wear for various materials including asbestos-type composite, sintered resin composite, sintered bronze composite, carbon-carbon composite, cast iron, C1040 carbon steel, 17-4 PH stainless steel, and 9310 alloy steel. The number of passes required to initiate measurable wear for the various types of brakes and clutches were correlated to the models contained in this handbook.

Robins AFB, one of the sponsors of the project to develop this handbook, provided an MC-2A air compressor unit for validation testing of the handbook procedures. The
MC-2A is a diesel engine-driven, rotary vane compressor mounted in a housed mobile trailer. It is designed for general flight line activities such as operating air tools requiring air from 5 psig to 250 psig. Two objectives were established for the validation effort: (a) determine the utility of the handbook to effect significant improvements in the reliability of new mechanical designs, and (b) determine the reliability of the MC-2A in its intended operating environment and introduce any needed design modifications for reliability improvement (References 78 and 79).

An additional reliability test was performed at the Naval Air Warfare Center in Patuxent River, Maryland to verify the application of the handbook in identifying existing and impending faults in mechanical equipment. A commercial actuator assembly was purchased and its design life estimated using the equations in this handbook. The actuator was then placed on test under stress conditions and an inspection made at the minimum calculated design life taking into consideration the sensitive parameters in the reliability equations. Upon inspecting the actuator at this point in time a revised remaining life estimate of the actuator was made and the test continued until failure. Test results were then compared with estimated values. The purpose of this test was to demonstrate the use of the handbook equations to revise failure estimates based on actual operating conditions when they may be different than originally anticipated and to continually obtain a more accurate estimate of time before the next maintenance action will be required (Reference 80).

An application of the methodology included in this Handbook to a diagnostic/prognostic system was demonstrated at the Naval Surface Warfare Center in West Bethesda, Maryland. Sensors were placed on various components of a water purification system being designed and tested at the laboratory. Equations as contained in this Handbook were then loaded into a laptop computer so that a real time determination of the remaining life of critical components could be made. Results of the experiment demonstrated that the application of prognostics to cognitive-based maintenance systems achieves the goal of performing maintenance actions only when there is objective evidence that the equipment requires attention. The result is a minimally manned, low maintenance and self-sufficient platform.

1.6 SUMMARY

The procedures presented in this handbook should not be considered as the only methods for a design analysis. An engineer needs many evaluation tools in his toolbox and new methods of performing dynamic modeling, finite element analysis and other stress/strength evaluation methods must be used in combination to arrive at the best possible reliability prediction for mechanical equipment.

The examples included in this introduction are intended to illustrate the point that there are no simplistic approaches to predicting the reliability of mechanical equipment. Accurate predictions of reliability are best achieved by considering the effects of the
operating environment of the system at the part level. The failure rates derived from
equations as tailored to the individual application then permits an estimation of design
life for any mechanical system. It is important to realize that the failure rates estimated
using the equations in this handbook are time dependent and that failure rates for
mechanical components must be combined for the time period in question to achieve a
total equipment failure rate. Section 1.3 and specifically Figure 1.2 demonstrate this
requirement.

It will be noted upon review of the equations that some of the parameters are very
sensitive in terms of life expectancy. The equations and prediction procedures were
developed using all known data resources. Additional research is needed to obtain
needed information on some of these “cause and effect” relationships for use in
continual improvement to the Handbook. In the meantime, the value of the Handbook
lies in understanding these “cause and effect” relationships so that when a discrepancy
does occur between predicted and actual failure rate, the cause is immediately
recognized. It is hoped that users of the Handbook and the MechRel software program
will communicate observed discrepancies in the Handbook and suggestions for
improvement to the Naval Surface Warfare Center. Suggestions, comments and
questions should be directed to:

**Tyrone L. Jones**  
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**Telephone:** 301-227-4383  
**FAX:** 301-227-5991  
**E-mail:** Tyrone.L. Jones@.navy.mil

### 1.7 REFERENCES

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment".

70. "Validation of Gearbox Reliability Models from Test Data", Report No. 87-D-0075,  
    October, 1987, Eagle Technology, Inc.

71. Dennis N. Pratt, "Investigation of Spline Coupling Wear", Report No. SY-51R-87,  
    December 1987, Naval Air Warfare Center, Patuxent River, Maryland

72. Randall F. Barron, "Engineering Model for Mechanical Wear", Report No. CMLD-
    CR-09-88, June 1988, Louisiana Tech University
73. Dennis Pratt, "Results of Air Compressor Reliability Investigation", Report No. TM 88-38 SY, January 1989, Naval Air Warfare Center, Patuxent River, Maryland

74. D. Pratt, "Results of Gear Pump Reliability Investigation", Report No. TM 89-24 SY, February 1990, Naval Air Warfare Center, Patuxent River, Maryland

75. D. Pratt, "Results of Centrifugal Pump Reliability Investigation", Report No. TM 89-69 SY, February 1990, Naval Air Warfare Center, Patuxent River, Maryland

76. D. Pratt, "Results of Pneumatic Impact Wrench Reliability Investigation", Report No. TM 90-88 SY, December 1990, Naval Air Warfare Center, Patuxent River, Maryland


79. D. Pratt, "Results of Air Force MC-2A Air Compressor Unit Reliability Investigation", Report No. TM 92-89 SY, March 1993, Naval Air Warfare Center, Patuxent River, Maryland


81. TD 84-3 R&M Data for Industrial Plants, A.P. Harris and Associates, 1984
This chapter provides a definition of some basic engineering terms to help establish a reference for the engineering analysis of mechanical equipment.

**Abrasive Wear** - The removal of material from a surface by the sliding or rolling of hard particles across the surface under pressure.

**Adhesive Wear** - The removal of material from a surface by the welding together and subsequent shearing of two surface areas that slide across each other under pressure.

**Anodizing** - The forming of a conversion coating on a metal surface (usually aluminum) by anodic oxidation.

**Armature** - The portion of the magnetic structure of a DC or universal motor which rotates.

**Axial Thrust** - The force or loads that are applied to a shaft in a direction parallel to the axis of the shaft (such as from a fan or pump).

**B₁₀ Life** - The number of hours at a given load that 90 percent of a set of apparently identical bearings will complete or exceed.

**Base Failure Rate** - A failure rate for a component or part in failures per million hours or failures per million operations depending on the application and derived from a database where the exact design, operational, and environmental parameters are known. Multiplying factors are then used to adjust the base failure rate to the new operating environment.

**Bending Moment** - The algebraic sum of the moments of the external forces to the left or right of any section on a member subjected to bending by transverse forces.

**Boring** - A machining method using a single point tool on internal surfaces of revolution.

**Brake Lining** - A frictional material used for stopping or retarding the relative movement of two surfaces.
**Cavitation** - The formation and instantaneous collapse on innumerable tiny voids within a liquid subjected to rapid and intense pressure changes.

**Cavitation Damage** - Erosion of a solid surface through the formation and collapse of cavities in an adjacent liquid.

**Center Distance** - The distance between centers of two gears.

**Coefficient of Friction** - This relationship is the ratio between two measured forces. The denominator is the normal force pressing two surfaces together. The numerator is the frictional force resisting the motion of one surface over the other.

**Compressive Strength** - The maximum compressive stress that a material is capable of developing based on the original area of cross section.

**Contamination** - Foreign matter or particles in a fluid system that are transported during its operation and which may be detrimental to system performance or even cause failure of a component.

**Corrosion** - The slow deterioration of materials by chemical agents and/or electromechanical reactions.

**Corrosion Fatigue** – Cracking produced by the combined action of repeated or fluctuating stress and a corrosive environment.

**Creep** - Continuous increase in deformation under constant or decreasing stress.

**Dependent Failure** - Failure caused by failure of an associated item or by a common agent.

**Diaphragm** – A member made of rubber or similar material used to contain hydraulic fluid within the forming cavity and to transmit pressure to the part being formed.

**Dirt lock** - Complete impedance of movement caused by stray contaminant particles wedged between moving parts.

**Durometer** - A device used to measure the hardness of rubber compounds.

**Duty Cycle** - The ratio of “on-time” to “on time + off time”, usually expressed as a percentage.

**Elastic Limit** - The greatest stress at which a material is capable of withstanding without any permanent deformation after removal of the load.

**Endurance Limit** - The stress level value when plotted as a function of the number of stress cycles at which point a constant stress value is reached. This is the maximum...
stress below which it can be assumed the material can endure an indefinite number of stress cycles.

**External leakage** - Leakage resulting in loss of fluid to the external environment.

**Failure Mode** - The indicator or symptom by which a failure is evidenced.

**Failure Rate** - The probable number of times that a given component will fail during a given period of operation under specified operating conditions. Failure rate may be in terms of time, cycles, revolutions, miles, etc.

**Fatigue** - The cracking, fracture or breakage of mechanical material due to the application of repeated, fluctuating or reversed mechanical stress less than the tensile strength of the material.

**Fatigue Life** - The number of stress cycles that can be sustained prior to failure under stated conditions.

**Fatigue Limit** - The maximum stress that presumably leads to fatigue fracture in a specified number of stress cycles.

**Fatigue Strength** - The maximum stress that can be sustained for a specified number of cycles without failure.

**Fretting** (or **Fretting Corrosion**) – Surface pitting caused by contacting asperities on mating surfaces. Corrosion damage occurs at the asperities of contact surfaces. It is caused by the combination of corrosion and the abrasive effects of debris in equipment with moving parts.

**Friction Material** - A product manufactured to resist sliding contact between itself and another surface in a controlled manner.

**Gear** - The larger of two meshed gears. If both gears are the same size they are both referred to as “gears”. See pinion

**Hardness** - A measure of material resistance to permanent or plastic deformation equal to a given load divided by the resulting area of indentation.

**Hooke’s Law** - Stress is proportional to strain. The law holds only up to the proportional limit.

**Independent Failure** - A failure of a device which is not caused by or related to failure of another device.
**Inductance** - The characteristic of an electric circuit by which varying current in the circuit produces a varying magnetic field which causes voltages in the same circuit or in a nearby circuit.

**Internal Leakage** - Leakage resulting in loss of fluid in the direction of fluid flow past the valving unit.

**Joint Efficiency** - The strength of a welded joint expressed as a percentage of the strength of the unwelded base metal.

**Leakage** - The flow of fluid through the interconnecting voids formed when the surfaces of two materials are brought into contact.

**Line of Action** - The line along which the point of contact between gear teeth travels, between the first point of contact and the last.

**Lubricant** - A substance used to reduce friction between two surfaces in contact.

**Mean Cycles Between Failure** - The total number of functioning cycles of a population of parts divided by the total number of failures within the population during the same period of time. This definition is appropriate for the number of hours as well as for cycles.

**Mean Cycles to Failure** - The total number of functioning cycles divided by the total number of failures during the period of time. This definition is appropriate for the number of hours as well as for cycles.

**Mean Stress** - The algebraic mean of the maximum and minimum stress in one cycle.

**Mil** - One thousandth of an inch (0.001 in.)

**Mild Steel** - Carbon steel with a maximum of about 0.25% carbon.

**Modulus of Elasticity** – A measure of the rigidity of metal. The slope of the initial linear portion of the stress-strain diagram; the larger the value, the larger the stress required to produce a given strain. Also known as Young's Modulus.

**Modulus of Rigidity** - See Modulus of Elasticity. The rate of change of unit shear stress with respect to unit shear strain for the condition of pure shear within the proportional limit. Also called Shear Modulus of Elasticity.

**Pinion** - The smaller of two meshing gears.

**Poisson's Ratio** - Ratio of lateral strain to axial strain of a material when subjected to uniaxial loading.
**Pressure Angle** - The angle between the Line of Action in a gear tooth and a line perpendicular to the Line of Centers.

**Proportional Limit** - The maximum stress at which strain remains directly proportional to stress.

**Random Failures** - Failures that occur before wear out, are not predictable as to the exact time of similar and are not associated with any pattern of similar failures. However, the number of random failures for a given population over a period of time at a constant failure rate can be predicted.

**Reliability** - A quantitative measure of the ability of a product to fulfill its intended function for a specified period of time under stated operating conditions.

**Silting** - An accumulation and settling of particles during component inactivity.

**Smearing** - Surface damage resulting from unlubricated sliding contact within a bearing.

**S-N diagram** - A graph showing the relationship of stress (S) and the number of cycles (N) before fracture in fatigue testing.

**Spalling** - The cracking and flaking of particles out of a surface.

**Stiction** - A change in performance characteristics or complete impedance of poppet or spool movement caused by wedging of minute particles between a poppet stem and housing or between spool and sleeve.

**Strain** - A measure of the relative change in size or shape of a body, usually a reference to the linear strain in the direction of applied stress.

**Stress** - Used to indicate any agency that tends to induce “failure”. It is a measure of intensity of force acting on a definite plane passing through a given point, measured in force per unit area.

**Stress-corrosion Cracking** - Failure by cracking under combined action of corrosion and applied or residual stress

**Stress Raiser** - Change in contour or discontinuity in structure that causes a local increase in stress

**Surface Finish** - A measure of the roughness of a surface as a result of final treatment.
**Temperature Rise** - Some of the electrical energy losses inherent in motors and other components are converted to heat causing some of the component parts to heat up while running or activated. The heated parts are at a higher temperature than the surrounding air causing a rise above ambient temperature. Friction has the same effect on mechanical component parts such as actuators and shafts.

**Tensile Strength** - Value of nominal stress obtained when the maximum (or ultimate) load that the specimen supports is divided by the cross-sectional area of the specimen. See Ultimate Strength

**Thermal Fatigue** - Fracture resulting from the presence of thermal gradients producing cyclic stresses in a structure

**Thrust Bearing** - Special bearings used to handle higher than normal axial forces exerted on the shaft of the motor or gearmotor as is the case with some fan or pump blade mountings

**Torque** - Turning force delivered by a motor or gearmotor shaft usually expressed in ft-lbs derived by computing H.P. x 5250/RPM = full load torque

**Triaxial Stress** - A state of stress in which none of the three principal axis stresses is zero.

**Ultimate Strength** - The maximum stress (tensile, compressive or shear) the material will withstand. See Tensile Strength.

**Viscosity** - A measure of internal resistance of a fluid which tends to prevent it from flowing.

**Wear-out Failure** - A failure which occurs as a result of mechanical, chemical or electrical degradation.

**Yield Strength** - The stress that will produce a small amount of permanent deformation in a material, generally a strain equal to 0.1 or 0.2 percent of the length of the specimen.

**Young's Modulus** - See Modulus of Elasticity.
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3.1 INTRODUCTION

A seal is a device placed between two surfaces to prevent the flow of gas or fluid from one region to another. Seals are used for both static and dynamic applications. Static seals such as gaskets and sealants are used to prevent leakage through a mechanical joint when there is no relative motion of mating surfaces. Truly static seals are designed to provide a complete barrier to a potential leakage path. These seals are “zero leakage” seals (down to $10^{-11}$ scc/sec helium). Some static seals are designed to accommodate limited movement of the surfaces being sealed due to changes in
pressure, vibration or thermal cycling. These seals are sometimes referred to as semi-static seals. A mechanical seal is generally described as a seal used to contain fluids in machinery and thus a mechanical seal could be either a static or dynamic seal depending on the application. A typical mechanical seal is the O-ring which is used in both static and dynamic applications. However, the employment of O-rings as primary dynamic seals is normally limited to short strokes and moderate pressures.

A dynamic seal is a mechanical device used to control leakage of fluid from one region to another when there is rotating or reciprocating motion between the sealing interfaces. An example of static and dynamic seal applications is shown in Figure 3.1.

Specific seals designs include labyrinth seal, lap seal and . These specific seal designs are included in the appropriate section of this Chapter as static or dynamic seals.

The reliability of a seal design is determined by the ability of the seal to restrict the flow of fluid from one region to another for its intended life in a prescribed operating environment. The evaluation of a seal design for reliability must include a definition of the design characteristics and the operating environment in order to estimate its design life. Section 3.2 discusses the reliability of gaskets and other static seals. A discussion of dynamic seal reliability is contained in Section 3.3.

3.2 GASKETS AND STATIC SEALS

A gasket is used to develop and maintain a barrier between mating surfaces of mechanical assemblies when the surfaces do not move relative to each other. The
barrier is designed to retain internal pressures, prevent liquids and gases from escaping the assembly, and prevent contaminants from entering the assembly. Gaskets can be metallic or nonmetallic. Flange pressure compresses the gasket material and causes the material to conform to surface irregularities in the flange and is developed by tightening bolts that hold the assembly together.

There are many factors which interact and contribute to seal performance. Gasket reliability is affected by the type of liquid or gas to be sealed, internal pressure, temperature, external contaminants, types of surfaces to be joined, surface roughness, and flange pressure developed at the joint. To achieve the barrier to a potential leakage path the seal must be sufficiently resilient to conform to cavity irregularities and imperfections, while remaining rigid enough to provide the required contact force needed to ensure a tight seal. This contact force is a function of the seal cross section, as well as the compression of the seal between the mating cavity faces. While static seals in most cases are designed for “zero-leakage” semi-static seals for applications where there is limited movement are not designed or intended to be “zero-leakage” seals. Their contact or compression force is typically an order of magnitude lower than a static seal.

An O-ring is a mechanical gasket in the shape of a torus. It is a loop of elastomer with a disc-shaped cross-section, designed to be seated in a groove and compressed during assembly between two or more parts, creating a seal at the interface. The joint may be static or may have relative motion between the parts and the O-ring. Joints with motion usually require lubrication of the O-ring to reduce wear rate. This is usually accomplished with the fluid being sealed.

3.2.1 Static Seal Failure Modes

O-ring materials may be subjected to high and low temperatures, chemical attack, vibration, abrasion, and movement. The primary failure mode of a gasket or seal is leakage. The integrity of a seal depends upon the compatibility of the fluid and sealing components, conditions of the sealing environment, and the applied load during application. Table 3-1 contains a list of typical failure mechanisms and causes of seal leakage. Other failure mechanisms and causes should be identified for the specific product to assure that all considerations of reliability are included in any design evaluation.
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3.2.2 Failure Rate Model Considerations

A review of failure rate data suggests the following characteristics be included in the failure rate model for gaskets and seals:

- Material characteristics
- Amount of seal compression
- Surface irregularities
- Seal size
- Fluid pressure
- Extent of pressure pulses
- Temperature
- Fluid viscosity
- Contamination level
- Fluid/material compatibility
- Leakage requirements
- Assembly/quality control procedures

The failure rate of a static seal is a function of actual leakage and the allowable leakage under conditions of usage, failure occurring when the rate of leakage reaches a predetermined threshold. This rate, derived empirically, can be expressed as follows:

\[
\lambda_{SE} = \lambda_{SE,B} \left( \frac{Q_a}{Q_f} \right)
\]

(3-1)

Where:

- \( \lambda_{SE} \) = Failure rate of gasket or seal considering operating environment, failures per million hours
- \( \lambda_{SE,B} \) = Base failure rate of seal or gasket due to random cuts, installation errors, etc. based on field experience data, failures per million hours
- \( Q_a \) = Actual leakage rate, in\(^3/min\)
- \( Q_f \) = Allowable leakage rate under conditions of usage, in\(^3/min\)

The allowable leakage, \( Q_f \) is determined from design drawings, specifications, or knowledge of component applications. The actual leakage rate, \( Q_a \), for a seal is determined from the standard equation for laminar flow around two curved surfaces (Reference 5):
\[ Q_a = \left( \frac{\pi \left( P_1^2 - P_2^2 \right)}{25 \nu_a \rho} \right) \left( \frac{r_o + r_i}{r_o - r_i} \right) H^3 \]  

(3-2)

Where:

- \( P_1 \) = System or upstream pressure, lbs/in\(^2\)
- \( P_2 \) = Standard atmospheric pressure or downstream pressure, lbs/in\(^2\)
- \( \nu_a \) = Absolute fluid viscosity, lb-min/in\(^2\)
- \( r_i \) = Inside radius of circular interface, in
- \( r_o \) = Outside radius of circular interface, in
- \( H \) = Conductance parameter, in [See Equation (3-4)]

For flat seals or gaskets the leakage can be determined from the following equation:

\[ Q_a = \left( \frac{\pi L \left( P_1^2 - P_2^2 \right)}{12 \nu_a w P_2} \right) H^3 \]  

(3-3)

Where:

- \( w \) = Width of non-circular flat seals, in
- \( L \) = Contact length, in

The conductance parameter \( H \) is dependent upon contact stress of the two sealing surfaces, hardness of the softer material and surface finish of the harder material (Reference 5). First, the contact stress (load/area) is calculated and the ratio of contact stress to Meyer hardness of the softer interface material computed. The surface finish of the harder material is then determined. The conductance parameter is computed from the following empirically derived formula:

\[ H = 0.23 \left( \frac{M}{C} \right)^{1.5} \cdot f^{2/3} \]  

(3-4)

Where:

- \( M \) = Meyer hardness (or Young's modulus) for rubber and resilient materials, lbs/in\(^2\)
$C = \text{Contact stress, lbs/in}^2 \text{ [See Equation (3-9)]}$

$f = \text{Surface finish, in}$

The surface finish, $f$, will deteriorate as a function of time at a rate dependent upon several factors:

- Seal degradation
- Contaminant wear coefficient (in$^3$/particle)
- Number of contaminant particles per in$^3$
- Flow rate, in$^3$/min
- Ratio of time the seal is subjected to contaminants under pressure
- Temperature of operation, °F

The contaminant wear coefficient is an inherent sensitivity factor for the seal or gasket based upon performance requirements. The quantity of contaminants includes those produced by wear and ingestion in components upstream of the seal and after the filter. Combining and simplifying terms provides the following equations for the failure rate of a seal.

For circular seals:

$$
\lambda_{SE} = \lambda_{SE,B} \left[ \frac{K_1 \left( P_1^2 - P_2^2 \right) H^3}{Q_f \nu_a P_2} \right] \cdot \left[ \frac{r_o + r_i}{r_o - r_i} \right] \quad (3-5)
$$

and, for flat seals and gaskets:

$$
\lambda_{SE} = \lambda_{SE,B} \left[ \frac{K_2 \left( P_1^2 - P_2^2 \right) L H^3}{Q_f \nu_a w P_2} \right] \quad (3-6)
$$

Where $K_1$ and $K_2$ are empirically derived constants
3.2.3 Failure Rate Model for Gaskets and Static Seals

By normalizing the equation to those values for which historical failure rate data from the Navy Maintenance and Material Management (3-M) system are available, the following model can be derived:

\[
\lambda_{SE} = \lambda_{SE,B} \cdot C_P \cdot C_Q \cdot C_{DL} \cdot C_H \cdot C_F \cdot C_V \cdot C_T \cdot C_N
\]  

(3-7)

Where:

- \( \lambda_{SE} \) = Failure rate of a seal in failures/million hours
- \( \lambda_{SE,B} \) = Base failure rate of seal, 2.4 failures/million hours
- \( C_P \) = Multiplying factor which considers the effect of fluid pressure on the base failure rate (Figure 3.8)
- \( C_Q \) = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 3.9)
- \( C_{DL} \) = Multiplying factor which considers the effect of seal size on the base failure rate (See Figure 3.10 or Figure 3.11)
- \( C_H \) = Multiplying factor which considers the effect of contact stress and seal hardness on the base failure rate (See Figure 3.12)
- \( C_F \) = Multiplying factor which considers the effect of seat smoothness on the base failure rate (See Figure 3.13)
- \( C_V \) = Multiplying factor which considers the effect of fluid viscosity on the base failure rate (See Table 3-3)
- \( C_T \) = Multiplying factor which considers the effect of temperature on the base failure rate (See Figure 3.14)
- \( C_N \) = Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 3-4)

The base failure rate has been determined from performance data in failures/million hours. Although not normally required for static seals, the base failure rate can be converted to failures/million operations based on projected utilization rates to be compatible with other failure rate equations in the Handbook.

The parameters in the failure rate equation can be located on an engineering drawing, by knowledge of design standards or by actual measurement. Other design parameters which have a minor effect on reliability are included in the base failure rate.
as determined from field performance data. The following paragraphs provide background information on those parameters included in the model.

### 3.2.3.1 Fluid Pressure

**Figure 3.8** provides fluid pressure multiplying factors for use in the model. Fluid pressure on a seal will usually be the same as the system pressure. The fluid pressure at the sealing interface required to achieve good mating depends on the resiliency of the sealing materials and their surface finish. It is the resilience of the seal which insures that adequate sealing stress is maintained while the two surfaces move in relation to one another with thermal changes, vibration, shock and other changes in the operating environment. The reliability analysis should include verification that sufficient pressure will be applied to affect a good seal.

At least three checks should be made to assure the prevention of seal leakage:

1. One surface should remain relatively soft and compliant so that it will readily conform to the irregularities of the harder surface
2. Sufficient sealing load should be provided to elastically deform the softer of the two sealing surfaces
3. Sufficient smoothness of both surfaces is maintained so that proper mating can be achieved

### 3.2.3.2 Allowable Leakage

**Figure 3.9** provides an allowable leakage multiplying factor for use in Equation 3-7. Determination of the acceptable amount of leakage which can be tolerated at a seal interface can usually be obtained from component specifications. The allowable rate is a function of operational requirements and the rate may be different for an internal or external leakage path.

### 3.2.3.3 Seal Size

**Figure 3.5** shows a typical installation for a seal and the measurements for \( r_i \) and \( r_o \). For a gasket, the inside perimeter dimension \( w \) and the contact length \( L \) are used in the equation. Figures 3.10 and 3.11 show the effect of seal size on reliability. The inside diameter of the seal is used in **Figure 3.10** as a close approximation of the seal size.

### 3.2.3.4 Conductance Parameter

Three factors comprise the conductance parameter:
(1) Hardness of the softer material
(2) Contact stress of the seal interface
(3) Surface finish of the harder material

(1) **Hardness of the softer material**: In the case of rubber seals and O-rings, the hardness of rubber is measured either by durometer or international hardness methods. Both hardness test methods are based on the measurement of the penetration of a rigid ball into a rubber specimen. Throughout the seal/gasket industry, the Type A durometer is the standard instrument used to measure the hardness of rubber compounds. The durometer has a calibrated spring which forces an indentor point into the test specimen against the resistance of the rubber. The scale of hardness is from 0 degrees for elastic modulus of a liquid to 100 degrees for an infinite elastic modulus of a material, such as glass. Readings in International Rubber Hardness Degree (IRHD) are comparable to those given by a Type A durometer ([Reference18](#)) when testing standard specimens per the ASTM methods. The relationship between the rigid ball penetration and durometer reading is shown in **Figure 3.3**.

---

**Figure 3.3 Relation Between International Rubber Hardness Degree (IRHD) and Rigid Ball Penetration**
Well-vulcanized elastic isotropic materials, like rubber seals manufactured from natural rubbers and measured by IRHD methods, have a known relationship to Young’s modulus. The relation between a rigid ball penetration and Young’s modulus for a perfectly elastic isotropic material is (Reference 18):

\[
\frac{F_i}{M_p} = 1.90 R_P^2 \left( \frac{P_D}{R_P} \right)^{1.35}
\]

(3-8)

Where:
- \( F_i \) = Indenting force, lbf
- \( M_p \) = Young’s modulus, lbs/in\(^2\)
- \( R_P \) = Radius of ball, in
- \( P_D \) = Penetration, in

Standard IRHD testers have a ball radius of 0.047 inches with a total force on the ball of 1.243 lbf. Using these testing parameters, the relationship between seal hardness and Young's modulus is shown in Figure 3.4. Since Young's modulus is expressed in lbs/in\(^2\) and calculated in the same manner as Meyer's hardness for rigid material; then, for rubber materials, Young's modulus and Meyer's hardness can be considered equivalent.

2) Surface finish of the harder material: - The seal gland is the structure which retains the seal. The surface finish on the gland will usually be about 32 microinches for elastomer seals, 16 microinches for plastic seals and 8 microinches for metals. In addition to average surface finish, the allowable number and magnitude of flaws in the gland must be considered in projecting leakage characteristics. Flaws such as surface cracks, ridges or scratches will have a detrimental effect on seal leakage. When projecting seal and gasket failure rates for different time periods of the equipment life cycle, it is important to consider the exposure to contaminants and their effect on surface finish.

3) Contact stress of the seal interface: - Seals deform to mate with rigid surfaces by elastic deformation. Since the deformation of the seal is almost entirely elastic, the initially applied seating load must be maintained. Thus, a load margin must be applied to allow for strain relaxation during the life of the seal yet not to the extent that permanent deformation takes place. An evaluation of cold flow characteristics is required for determining potential seal leakage of soft plastic materials. Although dependent on surface finish, mating of metal-to-metal surfaces generally requires a seating stress of two to three times the yield strength of the softer material. Figure 3.5 shows a typical installation of a gasket seal.
If the seal is pressure energized, the force $F$ applied to the seal must be sufficient to balance the fluid pressure forces acting on the seal and thus, prevent separation of the interface surfaces. This requirement is determined by the maximum applied fluid pressure, geometry of the seal groove and pressure gradient at the interface due to leakage. Motion at the interface is prevented by the radial friction forces at the interface to counter the fluid pressure forces tending to radially deform the seal. Thus, the radial restraining force $F$ will be greater than the radial pressure deformation forces.

The contact stress, $C$, in lbs/in$^2$ can be calculated by:

$$C = \frac{F_C}{A_{SC}}$$

(3-9)

Where:

- $F_C$ = Force compressing seals, lb
- $A_{SC}$ = Area of seal contact, in$^2$
or:

\[
C = \frac{F - P_1 \pi r_i^2 - \left( P_1 - P_2 \right) \left( \frac{r_o + r_i}{2} \right) \left( r_o - r_i \right)}{\pi \left( r_o^2 - r_i^2 \right)}
\]

(3-10)

Where:

- \( F \) = Maximum allowable force, lb
- \( P_1 \) = System pressure, lbs/in\(^2\)
- \( P_2 \) = Standard atmospheric pressure or downstream pressure, lbs/in\(^2\)
- \( r_o \) = Outside seal radius, in
- \( r_i \) = Inside seal radius, in

**Figure 3.5 Typical Seal Installation**

For most seals, the maximum allowable force \( F \) is normally two and one-half times the Young's modulus for the material. If too soft a material is used, the seal material will have insufficient strength to withstand the forces induced by the fluid and will rapidly fail by seal blowout. If the seal is too hard it will not sufficiently deform in the gland and immediate leakage will occur.
3.2.3.5 Fluid Viscosity

Viscosity of a fluid is much more dependent on temperature than it is on pressure. For example, when air pressure is increased from 1 atmosphere to 50, its viscosity is only increased by about 10%. In contrast, Figure 3.6 shows the dependence of viscosity on temperature for some common fluids. The graph shows how viscosity of liquids decreases with temperature while that of gases increases with temperature. Multiplying factors for the effect of fluid viscosity on the base failure rate of seals and gaskets are provided in Table 3-3. Viscosities for other fluids at the operating temperature can be found in referenced sources and the corresponding multiplying factor determined using the equation following Table 3-3. If the value located is in terms of kinematic viscosity, multiply the value by the specific gravity (density) at the desired temperature to determine the dynamic viscosity.

![Figure 3.6 Dynamic Viscosities of Various Fluids](image-url)

**Figure 3.6 Dynamic Viscosities of Various Fluids**
3.2.3.6 Operating Temperature

The operating temperature has a definite effect on the aging process of elastomer and rubber seals. Elevated temperatures, those temperatures above the published acceptable temperature limits, tend to continue the vulcanization or curing process of the materials, thereby significantly changing the original characteristics of the seal or gasket. It can cause increased hardening, brittleness, loss of resilience, cracking, and excessive wear. Since a change in these characteristics has a definite effect on the failure rate of the component, a reliability adjustment must be made.

Manufacturers of rubber seals will specify the maximum temperature, $T_R$, for their products. Typical values of $T_R$ are given in Table 3-5. An operating temperature multiplying factor can be derived as follows (Reference 22):

$$C_T = \frac{1}{2^t} \quad (3-11)$$

Where:

$$t = \frac{T_R - T_O}{18} \quad \text{for} \quad (T_R - T_O) \leq 40 \, ^oF$$

$T_R$ = Maximum rated temperature of material, °F

$T_O$ = Operating temperature, °F

And:

$$C_T = 0.21 \quad \text{for} \quad (T_R - T_O) > 40 \, ^oF$$

3.2.3.7 Fluid Contaminants

The quantities of contaminants likely to be generated by upstream components are listed in Table 3-4. The number of contaminants depends upon the design, the enclosures surrounding the seal, its physical placement within the system, maintenance practices and quality control. The number of contaminants may have to be estimated from experience with similar system designs and operating conditions.

3.2.3.8 Other Design Analysis Considerations

Those failure rate considerations not specifically included in the model but rather included in the base failure rates are as follows:

- Proper selection of seal materials with appropriate coefficients of thermal expansion for the applicable fluid temperature and compatibility with fluid medium
• Potential corrosion from the gland, seal, fluid interface
• Possibility of the seal rolling in its groove when system surges are encountered
• If O-rings can not be installed or replaced easily they are subject to being cut by sharp gland edges
• Potential periods of dryness between applications of fluid

Other factors which need to be considered as a check list for reliability include:
• Chemical compatibility between fluid and seal material
• Thermal stability
• Appropriate thickness and width of the seal material
• Initial and final seating (clamping) force

3.3 DYNAMIC SEALS

In contrast to gaskets and static seals, dynamic seals are used to control the leakage of fluid in those applications where there is motion between the mating surfaces being sealed. O-rings, packings and other seal designs are used in dynamic applications. Refer to Section 3.2 for a discussion of seals in general, the basic failure modes of seals and the parameters used in the equations to estimate the failure rate of a seal. The following paragraphs discuss the specific failure modes and model parameters for dynamic seals. The dynamic seal is often referred to as a mechanical seal. Mechanical seals are normally referred to as non-rotating seals on a shaft.

There are several types of dynamic seals to be considered including the contacting types such as lip seals and noncontacting types such as labyrinth seals. Dynamic seals are further divided as follows:

• Reciprocating Seal: A seal where the rod or piston moves back and forth through or with the seal

• Rotary Seal: A seal where a shaft rotates with relation to the seal

• Oscillating Seal: A seal where a shaft turns and returns with relation to the seal (one that combines oscillating and reciprocating motion is classified as an oscillating-reciprocating seal)

A typical contacting type dynamic seal is shown in Figure 3.7. In this example, the sealing surfaces are perpendicular to the shaft, with contact between the primary and mating rings to achieve a dynamic seal. Each of the sealing surfaces is lapped flat to eliminate leakage. Wear occurs at the dynamic seal faces from sliding contact between
the primary and mating rings. The rate of wear is small, as a film of the liquid sealed is maintained between the sealing faces. Preload from a spring is required to produce an initial seal, the spring pressure holding the primary and mating rings together during shutdown or when there is a lack of liquid pressure.

Figure 3.7 Typical Dynamic Seal

The reliability of a dynamic seal depends to a very large extent on its ability to maintain a thin fluid film in the gap between the mating faces while simultaneously minimizing the duration and extent of mechanical contact between asperities on the rubbing areas of these faces. Too much contact may overheat the materials; not enough contact may cause high leakage rates. Seal faces are obviously the most vulnerable parts of a dynamic seal, but other parts such as the static O-rings and springs may affect the life of the seal when subjected to excessive movements or high temperatures and pressures.

A dynamic seal may be an unbalanced design or a balanced design. Unbalanced seals are seal arrangements in which the hydraulic pressure of the seal chamber acts on the entire seal face without any of the force being reduced through the seal design. Unbalanced seals usually have a lower pressure limitation than balanced seals. A balanced seal design reduces the hydraulic forces acting on the seal faces through mechanical seal design. As the seal faces rub together, the amount of heat generated is determined by the amount of pressure applied, the lubricating film between the faces, the rotational speed, and the seal ring materials. Balanced seals reduce the seal ring area on which the stuffing box pressure acts. With a reduction in area, the overall closing force is diminished. This results in better lubrication and reduced heat generation and face wear compared to unbalanced seals. Unbalanced and balanced seal designs are shown in Figure 3.8
3.3.1 Dynamic Seal Failure Modes

The dynamic seal may be used to seal many different liquids at various speeds, pressures, and temperatures. The sealing surfaces are perpendicular to the shaft with contact between the primary and mating rings to achieve a dynamic seal. Dynamic seals are made of natural and synthetic rubbers, polymers and elastomers, metallic compounds, and specialty materials.

The most common modes of seal failure are by fatigue-like surface embrittlement, abrasive removal of material, and corrosion. Wear and sealing efficiency of fluid system seals are related to the characteristics of the surrounding operating fluid. Abrasive particles present in the fluid during operation will have a strong influence on the wear resistance of seals, the wear rate of the seal increasing with the quantity of environmental contamination. A good understanding of the wear mechanism involved will help determine potential seal deterioration. For example, contaminants from the environment such as sand can enter the fluid system and become embedded in the elastomeric seals causing abrasive cutting and damage to shafts.

Dynamic seals typically operate with sliding contact. Elastomer wear is analogous to metal degradation. However, elastomers are more sensitive to thermal deterioration than to mechanical wear. Hard particles can become embedded in soft elastomeric and metal surfaces leading to abrasion of the harder mating surfaces forming the seal, resulting in leakage. Abrasive particles can contribute to seal wear by direct abrasion and by plugging screens and orifices creating a loss of lubricant to the seal.

Wear and sealing efficiency of fluid system seals are related to the characteristics of the surrounding operating fluid. Abrasive contaminant particles present in the fluid during operation will have a strong influence on the wear resistance of seals. Hard particles, for example, can become embedded in soft elastomeric and metal surfaces leading to abrasion of the harder mating surfaces forming the seal, resulting in leakage.
Wear often occurs between the primary ring and mating ring. This surface contact is maintained by a spring. There is a film of liquid maintained between the sealing surfaces to eliminate as much friction as possible. For most dynamic seals, the three common points of sealing contact occur between the following points:

1. Mating surfaces between primary and mating rings
2. Between the rotating component and shaft or sleeve
3. Between the stationary component and the gland plate

The various failure mechanisms and causes for mechanical seals are listed in Table 3-2. Wear and sealing efficiency of fluid system seals are related to the characteristics of the surrounding operating fluid. Abrasive particles present in the fluid during operation will have a strong influence on the wear resistance of seals. Seals typically operate with sliding contact. Elastomer wear is analogous to metal degradation. However, elastomers are more sensitive to thermal deterioration than to mechanical wear. Hard particles can become embedded in soft elastomeric and metal surfaces leading to abrasion of the harder mating surfaces forming the seal, resulting in leakage.

Compression set refers to the permanent deflection remaining in the seal after complete release of a squeezing load while exposed to a particular temperature level. Compression set reflects the partial loss of elastic memory due to the time effect. Operating over extreme temperatures can result in compression-type seals such as O-rings to leak fluid at low pressures because they have deformed permanently or taken a set after used for a period of time.

An additional important seal design consideration is seal balance. Seal balance refers to the difference between the pressure of the fluid being sealed and the contact pressure between the seal faces. It is the ratio of hydraulic closing area to seal face area (parameter $k$ in Equation (3-13)). A balanced seal is designed so that the effective contact pressure is always less than the fluid pressure, reducing friction at the seal faces. The result is less rubbing wear, less heat generated and higher fluid pressure capability. In an unbalanced seal, fluid pressure is not relieved by the face geometry, the seal faces withstand full system fluid pressure in addition to spring pressure and the face contact pressure is greater than or equal to fluid pressure.

Seal balance then is a performance characteristic that measures how effective the seal mating surfaces match. If not effectively matched, the seal load at the dynamic facing may be too high causing the liquid film to be squeezed out and vaporized, thus causing a high wear rate. The fluid pressure from one side of the primary ring causes a certain amount of force to impinge on the dynamic seal face. The dynamic facing pressure can be controlled by manipulating the hydraulic closing area with a shoulder on a sleeve or by seal hardware. By increasing the area, the sealing force is increased.
<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE MECHANISMS</th>
<th>FAILURE CAUSES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Excessive leakage</td>
<td>Wear</td>
<td>- Misalignment</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Shaft out-of-roundness</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Excessive shaft end play</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Excessive torque</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Surface finish</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Contaminants</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Inadequate lubrication</td>
</tr>
<tr>
<td>Dynamic instability</td>
<td></td>
<td>- Misalignment</td>
</tr>
<tr>
<td>Embrittlement</td>
<td></td>
<td>- Contaminants</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Fluid/seal incompatibility</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Thermal degradation</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Idle periods between use</td>
</tr>
<tr>
<td>Spring Failure</td>
<td></td>
<td>- See Chapter 4, Table 4-1</td>
</tr>
<tr>
<td>Fracture</td>
<td></td>
<td>- Stress-corrosion cracking</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Excessive PV value</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Excessive fluid pressure on seal</td>
</tr>
<tr>
<td>Edge chipping</td>
<td></td>
<td>- Excessive shaft deflection</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Seal faces out-of-square</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Excessive shaft whip</td>
</tr>
<tr>
<td>Axial shear</td>
<td></td>
<td>- Excessive pressure loading</td>
</tr>
<tr>
<td>Torsional shear</td>
<td></td>
<td>- Excessive torque due to improper lubrication</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Excessive fluid pressure</td>
</tr>
<tr>
<td>Compression set</td>
<td></td>
<td>- Extreme temperature operation</td>
</tr>
<tr>
<td>Fluid seepage</td>
<td></td>
<td>- Insufficient seal squeeze</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Foreign material on rubbing surface</td>
</tr>
<tr>
<td>Seal face distortion</td>
<td></td>
<td>- Excessive fluid pressure on seal</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Foreign material trapped between faces</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Excessive PV value of seal operation</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Insufficient seal lubrication</td>
</tr>
</tbody>
</table>
Table 3-2. (continued) Typical Failure Mechanisms and Causes
For Dynamic Seals (Also see Table 3-1)

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE MECHANISMS</th>
<th>FAILURE CAUSES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Slow mechanical</td>
<td>Excessive friction</td>
<td>- Excessive squeeze</td>
</tr>
<tr>
<td>response</td>
<td></td>
<td>- Excessive seal swell</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Seal extrusion</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Metal – to – metal contact (out of alignment)</td>
</tr>
</tbody>
</table>

3.3.2 Pressure Velocity

Of greatest importance with dynamic seals is a properly designed seal face. Proper mating surface materials must be matched so that excessive heat isn't generated from the dynamic motion of the seal faces. Too much heat can cause thermal distortions on the face of the seal and cause gaps which can increase the leakage rate. It can also cause material changes that can significantly increase the seal wear rate. Therefore, a careful review of the seal material should be made for each surface of the dynamic seal face. Equation (3-12) (Reference 26) includes such coefficients of friction and wear rate. Table 3-6 shows frictional values for various seal face materials.

An important factor in the design of rotary seals is the pressure velocity (PV) coefficient. The PV coefficient is defined as the product of the seal face or system pressure and the fluid velocity. This factor is useful in estimating seal reliability when compared with manufacturer’s limits. If the PV limit is exceeded, a seal may wear at a rate greater than desired.

$$Q_s = C_1 \cdot PV \cdot \mu \cdot a_o$$  \hspace{1cm} (3-12)

Where:

- $Q_s$ = Heat input from the seal, BTU/hour
- $C_1$ = Numerical constant, 0.077
- $PV$ = Pressure-velocity coefficient [See Equation (3-13)]
- $\mu$ = Coefficient of friction (See Table 3-6)
- $a_o$ = Seal face area, in$^2$

The following equation defines the "PV" factor.
$PV = \frac{\pi}{12} \cdot DP \cdot d \cdot V \cdot k$ \hspace{1cm} (3-13)

Where:

- $DP$ = Pressure differential across seal face, lbs/in$^2$
- $d$ = Diameter of face seal, inches
- $V$ = Operating speed, rpm
- $k$ = Degree of seal unbalance

The frictional aspects of materials are not only important from a reliability viewpoint. Performance must also be considered. The more resistance a system incurs, the more power is lost and also the lower the efficiency value for the component.

There should be special consideration for tradeoffs involved with each type of seal material. For example, solid silicon carbide has excellent abrasion resistance, good corrosion resistance, and moderate thermal shock resistance. This material has better qualities than a carbon-graphite base material but has a $PV$ value of 500,000 lb/in-min while carbon-graphite has a 50,000 lb/in-min $PV$ value. With all other values being the same, the heat generated would be five times greater for solid silicon carbide than for carbon-graphite materials. The required cooling flow to the solid silicon carbide seal would be larger to maintain the film thickness on the dynamic seal faces. If this cooling flow can't be maintained, then an increase in wear would occur due to higher surface temperatures. The analyst should perform tradeoff analysis for each candidate design to maximize reliability. Typical PV limits are shown in Table 3-9.

### 3.3.3 Failure Rate Model for Dynamic Seals

Most of the seal modifying factors will remain the same as the ones previously specified by Equation (3-7), the exceptions being surface finish (See Section 3.3.3.1) and the addition of the PV factor (See Section 3.3.3.2). The seal model is modified as shown in Equation (3-14).

$\lambda_{SE} = \lambda_{SE,b} \cdot C_Q \cdot C_F \cdot C_V \cdot C_T \cdot C_N \cdot C_{PV}$ \hspace{1cm} (3-14)

Where: $\lambda_{SE}$ = Failure rate of dynamic seal in failures/million hours
\[ \lambda_{SE,B} = \text{Base failure rate of dynamic seal, 22.8 failures/million hours} \]

\[ C_Q = \text{Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 3.9) and Section 3.2.3.2} \]

\[ C_F = \text{Multiplying factor which considers the effect of surface finish on the base failure rate (See Sections 3.3.3.1, 3.2.3.4 and Figure 3.15)} \]

\[ C_V = \text{Multiplying factor which considers the effect of fluid viscosity on the base failure rate (See Table 3-3 and Section 3.2.3.5)} \]

\[ C_T = \text{Multiplying factor which considers the effect of seal face temperature on the base failure rate (See Figure 3.14 and Section 3.2.3.6)} \]

\[ C_N = \text{Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 3-4 and Section 3.2.3.7)} \]

\[ C_{PV} = \text{Multiplying factor which considers the effect of the pressure-velocity coefficient on the base failure rate (See Sections 3.3.2 and 3.3.3.2)} \]

### 3.3.3.1 Surface Finish Multiplying Factor - Dynamic Seals

Surface irregularities of dynamic seals may be more pronounced than static seals. In dynamic seal applications where the seal mates with a shaft, shaft hardness, smoothness and material are factors which must be considered in the design evaluation process. Maximum seal efficiency and life are obtained with a finely finished gland surface, usually in the 10 to 20 microinch range. A metal surface finish of less than 8 microinches rms increases the total frictional drag of a compound moving against it. The degree to which the finish can be maintained in the operating range must be considered when determining the surface finish of the gland for use in the model. Figure 3.15 provides a value for the surface finish multiplying factor as a function of the surface finish.

### 3.3.3.2 Fluid Contaminant Multiplying Factor – Dynamic Seals

When a cylinder rod extends out into a dirty environment where it can pick up dirt, lint, metal chips and other contaminants, this foreign material can nullify the benefits of the lubricant and cause rapid abrasive wear of both the O-ring and the rod. Equipment exposed to such conditions should contain a wiper ring to prevent the foreign material from reaching the O-ring. A felt ring is usually installed between the wiper and the seal to maintain lubrication of the rod during its return stroke.
Table 3-8 provides multiplying factors for various operating environments.

### 3.3.3.3 PV Multiplying Factor

$C_{PV}$ is the multiplying factor that multiplies the base failure rate by the ratio of $PV$ value for actual seal operation to design $PV$ value. The values for $PV_{DS}$ and $PV_{OP}$ used in Equation (3-15) will use the $PV$ formulation in Equation (3-13).

$$C_{PV} = \frac{PV_{OP}}{PV_{DS}} \tag{3-15}$$

Where:

- $PV_{OP}$ = $PV$ factor for actual seal operation
- $PV_{DS}$ = $PV$ factor for the original design

### 3.4 FAILURE MODE, EFFECTS AND CRITICALITY ANALYSIS

A seal or gasket will normally be a component part of a valve, pump or other system component at a higher indenture level. Therefore, the failure mode at the higher component level may refer to the seal as damaged or in the case of a dynamic seal as worn. Or the failure mode at the higher indenture level may list the failure mode as internal or external leakage with the seal or gasket listed as the failure cause. In either case the failure modes at the seal or gasket level must be evaluated and entered on the FMECA worksheets. Sections 3.2.1 and 3.3.1 and Tables 3-1 and 3-2 provide information on possible failure modes for static and dynamic seals. The next section provides some recommended procedures for completing the worksheets.

#### 3.4.1 Failure Mode and Effects Analysis

The FMECA is normally performed by first identifying the failure modes and then estimating the probability of occurrence for each identified failure mode. This section provides some guidelines for performing the FMEA for seals and gaskets.

**Function:** The following guidelines assume that a hardware bottom-up analysis is being performed. The purpose of the FMECA is to identify all potential failure modes of the equipment. Since the seal or gasket is generally a part of a component at a higher indenture level, the seal must be evaluated for reliability considering the operating environment of the valve, pump, actuator or other component where the seal is located. It is suggested that the seal be identified as a functional description as shown below.
This process permits a technical review of the seal function within the equipment and identification of all failure modes applicable to the seal design.

<table>
<thead>
<tr>
<th>Function No.</th>
<th>Function</th>
<th>Mode ID No.</th>
<th>Failure Modes</th>
<th>Cause ID No.</th>
<th>Failure Causes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Functional description of the component at the next higher indenture level</td>
<td>1</td>
<td>Failure modes at the component level applicable to the seal or gasket such as internal leakage or external leakage</td>
<td>1</td>
<td>Description of all probable independent causes of this specific failure mode at the seal level such as damaged O-ring</td>
</tr>
<tr>
<td>2</td>
<td>Only one failure mode should be entered for each row because of individual failure effects, $\alpha$ and $\beta$ values and compensating provisions</td>
<td>1</td>
<td>2</td>
<td>3</td>
<td></td>
</tr>
</tbody>
</table>

**Failure Modes:** Sections 3.1 and 3.2 and Tables 3-1 and 3-2 provide some guidelines for listing the failure modes of seals on the FMECA worksheets. It is important that the identified failure mode be directed toward a specific failure mode at the component level such as leakage. Only one failure mode should be entered for each row because of individual failure effects, $\alpha$ and $\beta$ values and compensating provisions. Some failure modes will not be applicable on the FMEA worksheets. They can be either removed or a 0.0 probability of occurrence assigned.

**Failure Causes:** The different causes of the failure mode occurring should be entered one row at a time. Again, only one failure cause should be entered so that the probability of occurrence of each failure cause can be entered. Sections 3.2.1 and 3.3.1 provide some guidelines for identifying failure causes. The operating environment needs to be considered so that all possible causes are entered such as contaminants, fluid incompatibility, pressure pulsations and excessive PV coefficient.

**Failure Effects:** Local failure effects are the consequences of the failure mode on the equipment at the part level. This entry may be the same as the failure cause or the failure effect at the component level depending on the complexity of the total equipment. The effect of the failure mode occurrence at the next higher indenture level is then entered. The end failure effect is the consequence of the failure mode occurrence at the total system level. In some cases the failure effects for different failure causes will be the same as shown in the following table.
**Failure Detection Method:** The method by which occurrence of the failure cause is detected is entered on the worksheet which may be by many different means such as operator, warning device, sensing device or routine diagnostic maintenance.

<table>
<thead>
<tr>
<th>Mission Phase or Operational Mode</th>
<th>Local Failure Effect</th>
<th>Next Higher Failure Effect</th>
<th>End Failure Effect</th>
<th>Failure Detection Method</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Consequences of the failure cause on equipment operation at the part level</td>
<td>Impact of the failure at the next higher indenture level</td>
<td>Total effect of the failure mode on the operation, function or operational status</td>
<td>Method by which occurrence of the failure cause is detected</td>
</tr>
</tbody>
</table>

**Compensating Provisions:** Provisions in the design or operator actions are entered on the worksheet that circumvent or mitigate the effect of the failure such as equipment redundancy, alarm provisions, safety features or alternate modes of operation.

**Severity Class:** One of the following severity categories is recorded:

- Class 1 is catastrophic, a failure which may cause death, total property damage or loss of mission (fire).
- Class II is critical, a failure that may cause severe injury or extensive property damage (transmission failure).
- Class III is marginal, a failure that may cause minor injury, minor property damage or mission degradation (windshield breakage).
- Class IV is minor (a failure not serious enough to cause injury, property damage or mission degradation but may necessitate repairs at a later time (turn signal)).

**Remarks:** Remarks need to be entered that clarify or amplify other worksheet entries such as design features, safety provisions combination of entries, etc.
### 3.4.2 Criticality Analysis

A Criticality Analysis (CA) is performed following the FMEA to determine the probability of occurrence that an identified failure mode will actually result in the defined end effect. In most cases the CA worksheet will repeat the columns that identify the component function, failure modes, failure causes, and ID numbers so that the entries can be traced from one worksheet to another. This section provides some guidelines for determining failure rates and occurrence probabilities.

**Failure Rate Data Source:** The Chapter of the Handbook can be entered or other source of the part failure rate is entered such as RAC, OREDA, or NPRD.

**Base Failure Rate:** The failure rate of the part as provided by the data source is entered on the worksheet.

**Adjustment ($\pi$) factor:** If the failure rate is obtained from a source where the operational environment is different from the intended operating environment of the equipment being analyzed, an adjustment factor may need to be applied. The O-ring for example may be a static seal or part of a sliding-action dynamic seal. The seal may also be part of an actuator under continuous operation. If this Handbook is used the duty cycle is automatically part of the calculation. For other failure rate sources a duty cycle may need to be considered in determining an adjustment factor.

**Part Failure Rate:** This entry is simply the base failure rate multiplied by the $\pi$ factor.

**Failure Effect Probability:** The failure mode may directly result in the listed end effect or it may not always result in the listed end effect. It is an engineering judgment as to the probability of occurrence after reviewing the failure causes and the operating environment of the equipment.

**Failure Mode Ratio:** The part failure rate will be the total failure rate of the component containing the seal under normal operating conditions. This failure rate needs to be subdivided per the particular component failure rate. The total of all the $\alpha$ values for a particular assembly will be equal to 1.0.
<table>
<thead>
<tr>
<th>Failure Rate Data Source</th>
<th>Base Failure Rate</th>
<th>Adjustment Factors ($\pi$)</th>
<th>Part Failure Rate</th>
<th>Failure Effect Probability ($\beta$)</th>
<th>Failure Mode Ratio ($\alpha$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>OREDADA, RAC, NPRD, NSWC HDBK, etc.</td>
<td>Failure rate from source</td>
<td>Correction factors as required to convert acquired failure rate to application failure rate</td>
<td>Base failure rate multiplied by $\pi$ factor</td>
<td>Conditional probability that identified failure mode will result in end effect (0.0 to 1.0)</td>
<td>Ratio of part failure rate related to identified failure mode</td>
</tr>
</tbody>
</table>

**Failure Mode Criticality Number**: This number is a combination of the part failure rate, failure effect probability and failure mode ratio. This number establishes a reference that can be used to compare the occurrence probability of this particular failure mode with other failure mode probabilities.

**Part Criticality Number**: A criticality number for the part is sometimes valuable for evaluating the criticality of the particular valve being analyzed in relation to other components of the system. The part criticality number is simply the summation of all the failure mode criticality numbers for the component. The part criticality number can be used to compare the probability of part failure with other component failure probabilities.

**Severity Class**: The severity class from the FMEA worksheet is normally repeated for informational purposes.

**Remarks**: Remarks are included on the worksheet to clarify or amplify other entries on the worksheet such as explanations of failure rate sources, modification numbers, and calculations.

<table>
<thead>
<tr>
<th>Failure Mode Criticality Number, $C_N$ ($\lambda_p \times \beta \times \alpha$)</th>
<th>Part Criticality Number $\sum C_N$</th>
<th>Severity Class</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Criticality number establishes reference for comparison purposes</td>
<td>Criticality number establishes reference for comparison purposes</td>
<td>The severity of the failure mode copied from the FMEA worksheet</td>
<td>Remarks to clarify or amplify other worksheet entries such as failure rate sources and adjustment factors</td>
</tr>
</tbody>
</table>
For $P_S \leq 1500$ lbs/in², $C_p = 0.25$

For $P_S > 1500$ lbs/in², $C_p = \left(\frac{P_S}{3000}\right)^2$

Where $P_S = P_1 - P_2$

**Figure 3.8** Fluid Pressure Multiplying Factor, $C_p$
For Leakage $> 0.03$ in$^3$/min, $C_Q = \frac{0.055}{Q_f}$

For Leakage $\leq 0.03$ in$^3$/min, $C_Q = 4.2 - (79 \cdot Q_f)$

**Figure 3.9 Allowable Leakage Multiplying Factor, $C_Q$**
$C_{DL} = 1.1 \, D_{SL} + 0.32$

Where: $D_{SL} = \text{Inner diameter of seal}$

**Figure 3.10 Seal Diameter Multiplying Factor, $C_{DL}$**
\[ C_{DL} = 0.45 \left( \frac{L}{w} \right) + 0.32 \]

Where: \( L \) = Total linear length of gasket

\( w \) = Minimum width of gasket

Figure 3.11 Gasket Size Multiplying Factor, \( C_{DL} \)
\[ C_H = \left( \frac{M}{C} \right)^{4.3} \left( \frac{0.55}{M/C} \right) \]

Where:  
\( M \) = Meyer Hardness, lbs/in\(^2\)  
\( C \) = Contact Pressure, lbs/in\(^2\)

Figure 3.12  Material Hardness/Contact Pressure, \( C_H \)
For \( f \leq 15 \mu\text{in} \), \( C_f = 0.25 \)

For \( f > 15 \mu\text{in} \), \( C_f = \frac{f^{1.65}}{353} \)

Where: \( f = \) Surface Finish, \( \mu\text{in} \) RMS

**Figure 3.13** Surface Finish Multiplying Factor, \( C_f \) (for static seals)
\[ C_T = \frac{l}{2^t} \]

Where: \[ t = \frac{(T_R - T_O)}{18} \] for \((T_R - T_O) \leq 40 \, ^\circ F\)

and: \[ C_T = 0.2l \] for \((T_R - T_O) > 40 \, ^\circ F\)

\(T_R = \) Rated Temperature of Seal, \(^\circ F\) (See Table 3-6)

\(T_O = \) Operating Temperature of Seal, \(^\circ F\)

**Figure 3.14 Temperature Multiplying Factor, \(C_T\)**
For \( f \leq 10 \mu\text{in}, \ C_f = 1.00 \)

For \( f > 10 \mu\text{in}, \ C_f = \frac{1}{2^{((10-f)/38)}} \)

Where: \( f = \) Surface Finish, \( \mu\text{in} \) RMS

**Figure 3.15**  Surface Finish Multiplying Factor, \( C_f \)  
(for dynamic seals)
Table 3-3. Fluid Viscosity/Temperature Multiplying Factor, $C_V$
for Typical Fluids

<table>
<thead>
<tr>
<th>FLUID</th>
<th>$C_V$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Fluid Temperature, °F</td>
</tr>
<tr>
<td></td>
<td>-50</td>
</tr>
<tr>
<td>Air</td>
<td>554.0</td>
</tr>
<tr>
<td>Oxygen</td>
<td>504.6</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>580.0</td>
</tr>
<tr>
<td>Carbon Dioxide</td>
<td>---</td>
</tr>
<tr>
<td>Water</td>
<td>---</td>
</tr>
<tr>
<td>SAE 10 Oil</td>
<td>---</td>
</tr>
<tr>
<td>SAE 20 Oil</td>
<td>---</td>
</tr>
<tr>
<td>SAE 30 Oil</td>
<td>---</td>
</tr>
<tr>
<td>SAE 40 Oil</td>
<td>---</td>
</tr>
<tr>
<td>SAE 50 Oil</td>
<td>---</td>
</tr>
<tr>
<td>SAE 90 Oil</td>
<td>---</td>
</tr>
<tr>
<td>Diesel Fuel</td>
<td>0.1617</td>
</tr>
<tr>
<td>MIL-H-83282</td>
<td>0.0031</td>
</tr>
<tr>
<td>MIL-H-5606</td>
<td>0.0188</td>
</tr>
</tbody>
</table>

--- Data for these temperatures determined to be unreliable

$$C_V = \left( \frac{V_o}{V} \right)$$

Where: \( V_o = 2 \times 10^{-9} \) lbf-min/in²

\( V \) = Dynamic viscosity of fluid being used, lbf-min/in²
### Table 3-4. Contaminant Multiplying Factor, $C_N$

<table>
<thead>
<tr>
<th>HYDRAULIC COMPONENT PRODUCING PARTICLES</th>
<th>PARTICLE MATERIAL</th>
<th>NUMBER PARTICLES UNDER 10 MICRON PER HOUR ($N_{10}$)</th>
<th>PER GPM</th>
<th>PER LPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston Pump</td>
<td>steel</td>
<td></td>
<td>0.017</td>
<td>0.0045</td>
</tr>
<tr>
<td>Gear Pump</td>
<td>Steel</td>
<td></td>
<td>0.019</td>
<td>0.0050</td>
</tr>
<tr>
<td>Vane Pump</td>
<td>Steel</td>
<td></td>
<td>0.006</td>
<td>0.0016</td>
</tr>
<tr>
<td>Cylinder</td>
<td>Steel</td>
<td></td>
<td>0.008</td>
<td>0.0021</td>
</tr>
<tr>
<td>Sliding action valve</td>
<td>Steel</td>
<td></td>
<td>0.0004</td>
<td>0.00011</td>
</tr>
<tr>
<td>Hose</td>
<td>rubber</td>
<td></td>
<td>0.0013</td>
<td>0.00034</td>
</tr>
</tbody>
</table>

$$C_N = \left( \frac{C_o}{C_{10}} \right)^3 N_{10} GPM_R$$  \text{ or }  $$C_N = \left( \frac{C_o}{C_{10}} \right)^3 N_{10} LPM_R$$

Where:

- $C_o$ = System filter size in microns
- $C_{10}$ = Standard system filter size = 10 micron
- $GPM_R$ = Rated flow in gallons/min
- $LPM_R$ = Rated flow in liters/min
- $N_{10}$ = Particles/hour/rated GPM or particles/hour/rated LPM for gas applications
Table 3-5.  $T_R$ Values for Typical Seal Materials (Reference 27)

<table>
<thead>
<tr>
<th>SEAL MATERIAL</th>
<th>$T_R$ ($^\circ$F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural rubber</td>
<td>160</td>
</tr>
<tr>
<td>Ethylene propylene</td>
<td>250</td>
</tr>
<tr>
<td>Neoprene</td>
<td>250</td>
</tr>
<tr>
<td>Nitrile</td>
<td>250</td>
</tr>
<tr>
<td>Polyacrylate</td>
<td>300</td>
</tr>
<tr>
<td>Fluorosilicon</td>
<td>450</td>
</tr>
<tr>
<td>Fluorocarbon</td>
<td>475</td>
</tr>
<tr>
<td>Silicon rubbers</td>
<td>450</td>
</tr>
<tr>
<td>Butyl rubber</td>
<td>250</td>
</tr>
<tr>
<td>Urethane</td>
<td>210</td>
</tr>
<tr>
<td>Fluroelastomers</td>
<td>500</td>
</tr>
<tr>
<td>Fluroplastics</td>
<td>500</td>
</tr>
<tr>
<td>Leather</td>
<td>200</td>
</tr>
<tr>
<td>Impregnated poromeric material</td>
<td>250</td>
</tr>
</tbody>
</table>

Table 3-6. Coefficient of Friction for Various Seal Face Materials

<table>
<thead>
<tr>
<th>SLIDING MATERIALS</th>
<th>ROTATING (seal head)</th>
<th>STATIONARY (mating ring)</th>
<th>COEFFICIENT OF FRICTION ($\mu$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon-graphite</td>
<td>- Cast Iron</td>
<td></td>
<td>0.07</td>
</tr>
<tr>
<td>(resin filled)</td>
<td>- Ceramic</td>
<td></td>
<td>0.07</td>
</tr>
<tr>
<td></td>
<td>- Tungsten Carbide</td>
<td></td>
<td>0.07</td>
</tr>
<tr>
<td></td>
<td>- Silicon Carbide</td>
<td></td>
<td>0.02</td>
</tr>
<tr>
<td></td>
<td>- Silicon Carbide</td>
<td></td>
<td>0.02</td>
</tr>
<tr>
<td></td>
<td>- Silicon Carbide</td>
<td></td>
<td>0.08</td>
</tr>
<tr>
<td>Silicon carbide</td>
<td>- Tungsten Carbide</td>
<td></td>
<td>0.02</td>
</tr>
<tr>
<td></td>
<td>- Silicon Carbide</td>
<td></td>
<td>0.05</td>
</tr>
<tr>
<td></td>
<td>- Tungsten Carbide</td>
<td></td>
<td>0.02</td>
</tr>
</tbody>
</table>

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Table 3-7. Pressure Gradient for Various Solutions

<table>
<thead>
<tr>
<th>LIQUID SEALED</th>
<th>$k$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Light-specific-gravity fluids</td>
<td>0.3</td>
</tr>
<tr>
<td>Water-base solutions</td>
<td>0.5</td>
</tr>
<tr>
<td>Oil-base solutions</td>
<td>0.7</td>
</tr>
<tr>
<td>Hydraulic fluids</td>
<td>0.3</td>
</tr>
</tbody>
</table>

Table 3-8. Contaminant Multiplying Factor for Dynamic Seals, $C_N$

<table>
<thead>
<tr>
<th>Dynamic Seal Operating Environment</th>
<th>$C_N$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mild environment, internal operation</td>
<td>1.0</td>
</tr>
<tr>
<td>Harsh environment salt spray, sand, dust</td>
<td>4.0</td>
</tr>
</tbody>
</table>

Table 3-9. Typical Pressure Velocity (PV) Limits

<table>
<thead>
<tr>
<th>Face Materials</th>
<th>PV (lb/in$^2$ ft/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon vs hard faced stainless steel</td>
<td>543,000</td>
</tr>
<tr>
<td>Carbon vs ceramic</td>
<td>543,000</td>
</tr>
<tr>
<td>Carbon vs leaded bronze</td>
<td>992,000</td>
</tr>
<tr>
<td>Carbon vs nickel iron</td>
<td>1,142,000</td>
</tr>
<tr>
<td>Carbon vs tungsten carbide</td>
<td>2,570,000</td>
</tr>
</tbody>
</table>

3.4 REFERENCES


4.1 INTRODUCTION

Mechanical springs are used in machine designs to exert force, provide flexibility, and to store or absorb energy. Springs are manufactured for many different applications such as compression, extension, torsion, power, and constant force. Depending on the application, a spring may be in a static, cyclic or dynamic operating mode. A spring is usually considered to be static if a change in deflection or load...
occurs only a few times, such as less than 10,000 cycles during the expected life of the spring. A static spring may remain loaded for very long periods of time. Cyclic springs are flexed repeatedly and can be expected to exhibit a higher failure rate due to fatigue. Dynamic loading refers to those intermittent occurrences of a load surge such as a shock absorber inducing higher than normal stresses on the spring.

The reliability of a spring will depend not only on the material and design characteristics, but to a great extent on the operating environment. Most springs are made of steel and therefore corrosion protection has a significant impact on reliability. Material properties, the processes used in the manufacturing of the spring, operating temperature, and corrosive media must all be known before any estimate of spring reliability can be made.

### 4.2 FAILURE MODES

The operating life of a mechanical spring arrangement is dependent upon the susceptibility of the materials to corrosion and stress levels (static, cyclic or dynamic). The most common failure modes for springs are fracture due to fatigue and excessive loss of load due to stress relaxation. Table 4-1 is a list of failure mechanisms and causes of spring failure. Other failure mechanisms and causes may be identified for a specific application to assure that all considerations of reliability are included in the prediction. Typical failure rate considerations include: level of loading, operating temperature, cycling rate and corrosive environment.

**Table 4-1. Failure Modes for a Mechanical Spring**

<table>
<thead>
<tr>
<th>APPLICATION</th>
<th>FAILURE MODES</th>
<th>FAILURE CAUSES</th>
</tr>
</thead>
</table>
| - Static (constant deflection or constant load) | - Load loss  
- Creep  
- Set | - Parameter change  
- Hydrogen embrittlement |
| - Cyclic (unidirectional or reverse stress, 10,000 cycles or more during the life of the spring) | - Fracture | - Material flaws  
- Hydrogen embrittlement  
- Stress concentration due to tooling marks and rough finishes  
- Corrosion  
- Misalignment |
| - Dynamic | - Fracture | - Maximum load ratio exceeded |
In many applications, compression and extension springs are subjected to elevated temperatures at high stresses which can result in relaxation or loss of load. This condition is often referred to as "set". After the operating conditions are determined, set can be predicted and allowances made in the spring design. When no set is allowed in the application, the spring manufacturer may be able to preset the spring at temperatures and stresses higher than those to be encountered in the operating environment.

Most extension spring failures occur in the area of the spring end. For maximum reliability, the spring wire must be smooth with a gradual flow into the end without tool marks or other stress risers. The spring ends should be made as an integral part of the coil winding operation and the bend radius should be at least one and one-half times the wire diameter.

The $S_{10}$ value for a spring is the number of cycles that 90% of the springs operating at the published stress level can be expected to complete or exceed before exhibiting the first evidence of fatigue. If an $S_{10}$ value for the spring can be obtained, this value should be used in conjunction with the environmental multiplying factors contained in this Chapter. The procedure for estimating spring failure rates contained herein is intended to be used in the absence of specific $S_{10}$ data.

4.3 FAILURE RATE CONSIDERATIONS

The following paragraphs describe the terms and parameters used in developing failure rate equations for springs.

4.3.1 **Static Springs**

Static springs can be used in constant deflection or constant load applications. A constant deflection spring is cycled through a specified deflection range, the loads on the spring causing some set or relaxation which in turn lowers the applied stress. The spring may relax with time and reduce the applied load. Under constant load conditions, the load applied to the spring does not change during operation. Constant load springs may set or creep, but the applied stress is constant. The constant stress may result in fatigue lives shorter than those found in constant deflection applications.

4.3.2 **Cyclic Springs**

Cyclic springs can be classified as being unidirectional or reverse loaded. In one case, the stress is always applied in the same direction, while in the other, stress is applied first in one direction then in the opposite direction. Figure 4.18 shows the relationship between the cycle rate of a spring and its effect on failure rate.
4.3.3 **Modulus of Rigidity**

The modulus of rigidity \((G_M)\) is a material property defining the resistance to shearing stresses for the spring material, the ratio of shearing stress to shear strain. Typical values are provided in Table 4-2.

4.3.4 **Modulus of Elasticity**

The modulus of elasticity provides a measure of elasticity in tension for the spring material. Typical values are provided in Table 4-2.

4.3.5 **Spring Index**

Spring index \((r)\) is the ratio of mean coil diameter to wire diameter. A spring with a high index will tend to tangle or buckle.

4.3.6 **Spring Rate**

Spring rate \((R)\) is the change in load per unit deflection, a measure of spring relaxation.

4.3.7 **Shaped Springs**

If the spring has a variable diameter such as occurs for conical, barrel and hourglass springs, the spring can be divided analytically into smaller increments and the failure rate calculated for each. The failure rate for the total spring is computed by adding the rates for the increments.

4.3.8 **Number of Active Coils**

For compression springs with closed ends, either ground or not ground, the number of active coils is two less than the total number of coils. There is some activity in the end coils, but during deflection, some active material comes in contact with the end coils and becomes inactive. Therefore, the total number of coils minus two is a good approximation for the number of active coils. For extension springs, all coils are active.

4.3.9 **Tensile Strength**

The tensile strength provides a measure of spring material deformation or set as a function of stress. Values of tensile strength are included in Table 4-3.

4.3.10 **Corrosive Environment**

Corrosion will reduce the load-carrying capability of a spring and its life. The precise effect of a corrosive environment on spring performance is difficult to predict. The reliability of a spring in terms of fatigue life and load-carrying ability will be affected...
by corrosion, the quantitative effect being very hard to predict. Springs are almost always in contact with other metal parts. If a spring is to be subjected to a corrosive environment, the use of inert materials provides the best defense against corrosion. Protective coatings can also be applied. In special situations, shot peening can be used to prevent stress corrosion and cathodic protection systems can be used to prevent general corrosion. The spring material is normally more noble (chemically resistant to corrosion) than the structural components in contact with it because the lesser noble alloy will be attacked by the electrolyte. The effects of corrosion on spring reliability must be based on experience data considering the extent of a corrosive environment. If corrosive protection is known to be applied to the spring during the manufacturing process, a multiplying factor, $C_R$, of 1.0 is used in conjunction with the base failure rate. Values of $C_R$ greater than 1.0 are used based on the user's experience with the spring and the operating environment.

4.3.11 Manufacturing Processes

The following effects of manufacturing processes need to be considered in evaluating a design for reliability:

- Sharp corners and similar stress risers should be minimized.
- The hardness of the spring material can be sensitive to plating and baking operations. Quality control procedures for these operations should be reviewed. A multiplying factor, $C_M$, of 1.0 should be used in conjunction with the base failure rate for known acceptable quality control procedures; otherwise a higher value for the multiplying factor should be used based on previous experience with the manufacturer.

4.3.12 Other Reliability Considerations for Springs

The most common failure modes of springs include fracture due to fatigue and excessive loss of load. A reliability analysis should include a review of the following items to assure maximum possible life:

- When a spring is loaded or unloaded, a surge wave may transmit torsional stress to the point of restraint. The impact velocity should be determined to assure that the maximum load rating of the spring is not exceeded.
- Operating temperature should be determined. Both high and low temperature conditions may require consideration of specialized materials.
- Exposure to electrical fields may magnetize the spring material and cause fatigue failure.
4.4 FAILURE RATE MODELS

4.4.1 Compression Spring

The compression spring is the most commonly used spring in machine designs. An example of a compression spring is shown in Figure 4.1.

![Figure 4.1 Typical Helical Compression Spring](image)

The failure rate of a compression spring depends upon the stress on the spring and the relaxation provided by the material. This relaxation (change in load per unit deflection) is referred to as the spring rate, $R$. The spring rate for a compression spring is calculated using Equation (4-1).

$$R = \frac{G_M (D_W)^4}{8 (D_c)^3 N_a} = \frac{P_L}{L_1 - L_2}$$  \hspace{1cm} (4-1)

Where:
- $R$ = Spring rate, lbs/in
- $G_M$ = Modulus of rigidity, lbs/in$^2$
- $D_W$ = Wire diameter, in
- $D_c$ = Mean diameter of spring, in
- $N_a$ = Number of active coils (See Section 4.3.8)
- $P_L$ = Load, lbs
- $L_1$ = Initial length of spring, in
- $L_2$ = Final deflection of spring, in
The spring rate can be determined experimentally by deflecting the spring to 20% of available deflection and measuring the load \( P_1 \) and spring length \( L_1 \). Next, the spring is deflected to 80% of available deflection measuring the load \( P_2 \) and spring length \( L_2 \), being certain that no coils other than the closed ends are touching. The spring rate is then calculated as follows:

\[
R = \frac{P_2 - P_1}{L_1 - L_2} = \frac{P_L}{L_1 - L_2}
\]  

(4-2)

Stress in the spring is also proportional to the load, \( P_L \) according to the following relationship:

\[
S_G = \frac{8 P_L D_C}{\pi D_w^3} K_W
\]  

(4-3)

Where:
- \( S_G \) = Spring stress, lbs/in\(^2\)
- \( K_W \) = Spring concentration factor (See equation 4-4)
- \( D_C \) = Mean coil diameter, in
- \( D_W \) = Wire Diameter, in

The spring concentration factor, \( K_W \) is a function of the Spring index (ratio of the coil diameter to wire diameter).

\[
K_W = \frac{4r - 1}{4r - 4} + \frac{0.615}{r}
\]  

(4-4)

Where: \( r \) = Spring index = \( D_C / D_W \)

\( P_L \) in Equation (4-1) can be substituted into Equation (4-3) for a stress level equation, and the spring failure rate can be determined from a ratio of stress level to the material tensile strength according to the following empirically derived relationship (Reference 14):

\[
\lambda_{SP} = \lambda_{SP,B} \left( \frac{S_G}{T_S} \right)^3 = \lambda_{SP,B} \left( \frac{8 P_L D_C K_W}{\pi T_S D_w^3} \right)^3
\]  

(4-5)
Where:

\[ \lambda_{SP} = \text{Failure rate of spring, failures/million hours} \]
\[ \lambda_{SP,B} = \text{Base failure rate for spring, 23.8 failures/million hours} \]
\[ T_S = \text{Material tensile strength, lbs/in}^2 \]

A generalized equation that adjusts the base failure rate of a compression spring considering anticipated operating conditions can be established:

\[ \lambda_{SP} = \lambda_{SP,B} \cdot C_G \cdot C_{DW} \cdot C_{DC} \cdot C_N \cdot C_Y \cdot C_L \cdot C_K \cdot C_{CS} \cdot C_R \cdot C_M \]

(4-6)

Where:

\[ C_G = \text{Multiplying factor which considers the effect of the material rigidity modulus on the base failure rate (See Table 4-2)} \]
\[ C_{DW} = \text{Multiplying factor which considers the effect of the wire diameter on the base failure rate (See Figure 4.8)} \]
\[ C_{DC} = \text{Multiplying factor which considers the effect of coil diameter on the base failure rate (See Figure 4.9)} \]
\[ C_N = \text{Multiplying factor which considers the effect of the number of active coils on the base failure rate (See Figure 4.10)} \]
\[ C_Y = \text{Multiplying factor which considers the effect of material tensile strength, } T_S, \text{ on the base failure rate (See Table 4-3)} \]
\[ C_L = \text{Multiplying factor which considers the effect of spring deflection on the base failure rate (See Figure 4.11)} \]
\[ C_K = \text{Multiplying factor which includes the spring concentration factor on the base failure rate (See Figure 4.12)} \]
\[ C_{CS} = \text{Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.18)} \]
\[ C_R = \text{Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)} \]
\[ C_M = \text{Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)} \]

The parameters in the failure rate equation can be located on an engineering drawing by knowledge of design standards or by actual measurements. Other manufacturing, quality, and maintenance contributions to failure rate are included in the base failure rate as determined from field performance data.

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4.4.2 Extension Spring

Helical extension springs store energy in spring tensioning devices and are used to exert a pulling force. Most helical extension springs are coiled with initial tension, equal to the minimum force required to separate adjacent coils. Extension springs require a method of attachment to other parts of the assembly. For extension springs, all coils are active and $N_a$ will be equal to the number of coils. Otherwise, the failure rate equations for extension springs are similar to compression springs and the procedures in Section 4.4.1 should be used.

4.4.3 Torsion Spring

Helical torsion springs are used to apply a torque or store rotational energy, the most common application, the clothes pin. Torsion springs are stressed in bending as shown in Figure 4.2. A torsion spring should always be loaded in a direction that causes its body diameter to decrease because of increased stresses when the spring is loaded in a direction which increases body diameter.

![Figure 4.2 Typical Helical Torsion Spring](image)

The mean diameter of a helical torsion spring is equal to:

$$D_i = \frac{ID + OD}{2} \quad (4-7)$$

The spring diameter will change with deflection according to the following equation:
\[ D_c = \frac{D_I N_a}{N_a + \theta} \]  \hspace{1cm} (4-8)

Where:

- \( D_c \) = Mean diameter after deflection
- \( D_I \) = Initial mean diameter, in.
- \( \theta \) = Angular deflection from free position, revolutions
- \( N_a \) = Number of active coils

Most torsion springs are close-wound, with body length equal to wire diameter multiplied by the number of turns plus one. When the spring is deflected in a direction which reduces its coil diameter, body length increases to \( L_2 \) according to the following equation:

\[ L_2 = D_w (N_a + 1 + \theta) \]  \hspace{1cm} (4-9)

Where:

- \( D_w \) = Wire diameter, in

Stress in torsion springs is due to bending and for round wire is calculated with the following equation:

\[ S = \frac{3 E_M D_w \theta}{\pi D_I N_a} \]  \hspace{1cm} (4-10)

Where:

- \( S \) = Bending stress, lbs/in\(^2\)
- \( E_M \) = Modulus of Elasticity, lbs/in\(^2\)
- \( D_w \) = Wire diameter, in
- \( \theta \) = Angular deflection, revolutions
- \( D_I \) = Mean diameter of spring, in
- \( N_a \) = Number of active coils (See Section 4.3.8)

The equation to determine the failure rate of a torsion spring can be written as follows:
\[ \lambda_{SP} = \lambda_{SP,B} \left( \frac{S}{T_S} \right)^3 \cdot C_{CS} \cdot C_R \cdot C_M \] (4-11)

From this equation a generalized equation can be developed containing a base failure rate with applicable multiplying factors:

\[ \lambda_{SP} = \lambda_{SP,B} \cdot C_E \cdot C_{DW} \cdot C_N \cdot C_Y \cdot C_L \cdot C_{CS} \cdot C_{DC} \cdot C_R \cdot C_M \] (4-12)

Where:

- \( \lambda_{SP} \) = Failure rate of torsion spring, failures/million hours
- \( \lambda_{SP,B} \) = Base failure rate for torsion spring, 14.3 failures/million hours
- \( C_E \) = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)
- \( C_{DW} \) = Multiplying factor which considers the effect of the wire diameter on the base failure rate (See Figure 4.8)
- \( C_N \) = Multiplying factor which considers the effect of the number of active coils on the base failure rate (See Figure 4.10)
- \( C_Y \) = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)
- \( C_L \) = Multiplying factor which considers the effect of spring deflection on the base failure rate (See Figure 4.13)
- \( C_{CS} \) = Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.18)
- \( C_{DC} \) = Multiplying factor which considers the effect of coil diameter on the base failure rate (See Figure 4.19)
- \( C_R \) = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)
- \( C_M \) = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

4.4.4 Curved Washer

Curved washers are used to secure fasteners, distribute loads, absorb vibrations and axial end play, and other similar applications. A typical curved washer is shown in Figure 4.3. A special type of curved washer, the Belleville washer, is discussed in
Section 4.4.6. When a load is applied to a curved washer it tends to flatten causing radial and circumferential strains. This elastic deformation constitutes the spring action. Stress is not distributed uniformly in curved washers, the greatest stress occurring at the convex inner edge. Curved washers exert a relatively light thrust load. Bearing surfaces should be hard to prevent washer corners from scoring the shaft.

![Figure 4.3 Typical Curved Washer](image)

The stress on a curved washer is:

\[
S = \frac{6 E_M f t}{(OD)^2}
\]  

(4-13)  

Where:  

- \( S \) = Bending stress, lb/in\(^2\)  
- \( E_M \) = Modulus of Elasticity, lb/in\(^2\)  
- \( f \) = Washer deflection, in  
- \( t \) = Washer thickness, in  
- \( OD \) = Outside Diameter, in

The failure rate of a curved washer is determined using the following equation:
A generalized equation that adjusts the base failure rate of a curved washer considering anticipated operating conditions can be established:

\[
\lambda_{SP} = \lambda_{SP,B} \left( \frac{S}{T_s} \right)^3 \cdot C_{CS} \cdot C_{R} \cdot C_{M}
\]  \hspace{1cm} (4-11) \text{ ref}

Where:

- \( \lambda_{SP} \) = Failure rate of curved washer, failures/million hours
- \( \lambda_{SP,B} \) = Base failure rate for curved washer, 1.1 failures/million hours
- \( C_E \) = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)
- \( C_t \) = Multiplying factor which considers the effect of the material thickness on the base failure rate (See Figure 4.14)
- \( C_D \) = Multiplying factor which considers the effect of washer diameter on the base failure rate (See Figure 4.15)
- \( C_Y \) = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)
- \( C_f \) = Multiplying factor which considers the effect of washer deflection on the base failure rate (See Figure 4.16)
- \( C_{CS} \) = Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.18)
- \( C_R \) = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)
- \( C_M \) = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

### 4.4.5 Wave Washer

Wave washers are used to apply moderate thrust loads when radial space is limited. A typical wave washer is shown in Figure 4.4.

The stress on a wave washer is given by:
\[ S = \frac{0.3 \pi E_M f t N^2}{D^2} \]  \hspace{1cm} (4-15)

Where:
- \( S \) = Bending stress, lbs/in\(^2\)
- \( E_M \) = Modulus of Elasticity, lbs/in\(^2\)
- \( f \) = Deflection, in
- \( t \) = Material thickness, in
- \( N \) = Number of waves
- \( D \) = Mean diameter, in = (OD + ID)/2
- \( OD \) = Outside Diameter, in
- \( ID \) = Inside Diameter, in

![Figure 4.4 Typical Wave Washer](image)

The failure rate of a wave washer is determined using the following equation:
A generalized equation that adjusts the base failure rate of a wave washer considering anticipated operating conditions can be established:

\[ \lambda_{SP} = \lambda_{SP,B} \left( \frac{S}{T_S} \right)^3 \cdot C_{CS} \cdot C_R \cdot C_M \]  

(4-11) ref

A generalized equation that adjusts the base failure rate of a wave washer considering anticipated operating conditions can be established:

\[ \lambda_{SP} = \lambda_{SP,B} \cdot C_E \cdot C_t \cdot C_D \cdot C_Y \cdot C_f \cdot C_{NW} \cdot C_{CS} \cdot C_R \cdot C_M \]  

(4-16)

Where:

- \( \lambda_{SP} \) = Failure rate of wave washer, failures/million hours
- \( \lambda_{SP,B} \) = Base failure rate for wave washer, 1.9 failures/million hours
- \( C_E \) = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)
- \( C_t \) = Multiplying factor which considers the effect of the material thickness on the base failure rate (See Figure 4.14)
- \( C_D \) = Multiplying factor which considers the effect of washer diameter on the base failure rate (See Figure 4.15)
- \( C_Y \) = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)
- \( C_f \) = Multiplying factor which considers the effect of washer deflection on the base failure rate (See Figure 4.16)
- \( C_{NW} \) = Multiplying Factor which considers the number of waves on the base failure rate (See Table 4-4)
- \( C_{CS} \) = Multiplying factor which considers the effect of cycle rate on the base failure rate (See Figure 4.18)
- \( C_R \) = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)
- \( C_M \) = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

### 4.4.6 Belleville Washer

When a load is applied to a Belleville washer it tends to flatten causing radial and circumferential strains. This elastic deformation creates the spring action. A typical Belleville washer is shown in Figure 4.5. Belleville washers are capable of providing...
very high loads at small deflections. Stress is not distributed uniformly in Belleville washers. The highest stress occurs at the top inner edge and can be estimated with the following equation:

\[
S = \frac{E_M f R}{1 - \mu^2} \left(\frac{t}{a^2}\right)
\]

(4-17)

Where:
- \( S \) = Bending stress, lbs/in²
- \( E_M \) = Modulus of Elasticity, lbs/in²
- \( f \) = Deflection, in
- \( \mu \) = Poisson's Ratio
- \( R \) = Dimension factor (See Figure 4.17)
- \( t \) = Material thickness, in
- \( a \) = O.D./2, in

Figure 4.5 Typical Belleville Washer

The failure rate of a belleville washer is determined using the following equation:

\[
\lambda_{SP} = \lambda_{SP,B} \left(\frac{S}{T_S}\right)^3 \cdot C_{CS} \cdot C_R \cdot C_M
\]

(4-11) ref
A generalized equation that adjusts the base failure rate of a belleville washer considering anticipated operating conditions can be established:

\[
\lambda_{SP} = \lambda_{SP,B} \cdot C_E \cdot C_t \cdot C_D \cdot C_f \cdot C_Y \cdot C_S \cdot C_{CS} \cdot C_R \cdot C_M
\]  

(4-18)

Where:

- \( \lambda_{SP} \) = Failure rate of belleville washer, failures/million hours
- \( \lambda_{SP,B} \) = Base failure rate for belleville washer, 2.6 failures/million hours
- \( C_E \) = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)
- \( C_t \) = Multiplying factor which considers the effect of material thickness on the base failure rate (See Figure 4.14)
- \( C_D \) = Multiplying factor which considers the effect of washer size on the base failure rate (See Figure 4.15)
- \( C_f \) = Multiplying factor which considers the effect of washer deflection under load on the base failure rate (See Figure 4.16)
- \( C_Y \) = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)
- \( C_S \) = Multiplying factor for compressive stress (See Figure 4.17)
- \( C_{CS} \) = Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.18)
- \( C_R \) = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)
- \( C_M \) = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

4.4.7 Cantilever Spring

Cantilever springs are fabricated from flat strip material which stores and releases energy upon being deflected by an external load. A typical cantilever spring is shown in Figure 4.6. In complex designs, only a small part of the device may be functioning as a spring, and for analytical purposes, that portion which is active during operation may be considered as an independent device.

The bending stress for cantilever springs can be determined as follows:
\[
S = \frac{3E_M f t}{2 L^2} \tag{4-19}
\]

Where:
- \(S\) = Bending stress, lbs/in\(^2\)
- \(E_M\) = Modulus of elasticity, lbs/in\(^2\)
- \(f\) = deflection, in
- \(t\) = thickness, in
- \(L\) = length, in

Figure 4.6  Typical Cantilever Spring

The failure rate of a cantilever spring is determined using the following equation:

\[
\lambda_{SP} = \lambda_{SP,B} \left( \frac{S}{T_S} \right)^3 \cdot C_{CS} \cdot C_R \cdot C_M \tag{4-11} \text{ ref}
\]

A generalized equation that adjusts the base failure rate of a cantilever spring considering anticipated operating conditions can be established:

\[
\lambda_{SP} = \lambda_{SP,B} \cdot C_E \cdot C_t \cdot C_L \cdot C_f \cdot C_T \cdot C_{CS} \cdot C_R \cdot C_M \tag{4-20}
\]
Where:

\[ \lambda_{SP} = \text{Failure rate of cantilever spring, failures/million hours} \]
\[ \lambda_{SP,B} = \text{Base failure rate for cantilever spring, 1.1 failures/million hours} \]
\[ C_E = \text{Multiplying factor which considers the effect of the material} \]
\[ \text{elasticity modulus on the base failure rate (See Table 4-2)} \]
\[ C_t = \text{Multiplying factor which considers the effect of material} \]
\[ \text{thickness on the base failure rate (See Figure 4.14)} \]
\[ C_L = \text{Multiplying factor which considers the effect of spring length} \]
\[ \text{on the base failure rate (See Figure 4.20)} \]
\[ C_f = \text{Multiplying factor which considers the effect of spring deflection} \]
\[ \text{on the base failure rate (See Figure 4.16)} \]
\[ C_Y = \text{Multiplying factor which considers the effect of material} \]
\[ \text{tensile strength on the base failure rate (See Table 4-3)} \]
\[ C_{CS} = \text{Multiplying factor which considers the effect of spring cycle} \]
\[ \text{rate on the base failure rate (See Figure 4.18)} \]
\[ C_R = \text{Multiplying factor which considers the effect of a corrosive} \]
\[ \text{environment on the base failure rate (See Section 4.3.10)} \]
\[ C_M = \text{Multiplying factor which considers the effect of the} \]
\[ \text{manufacturing process on the base failure rate (See Section 4.3.11)} \]

### 4.4.8 Beam Spring

Beam springs are usually rectangular in shape and formed into an arc as shown in Figure 4.7. Assuming the ends are free to laterally expand, stress can be computed as follows:

\[
S = \frac{6 \ E_M \ f \ t}{L^2} \tag{4-21}
\]

Where:

\[ S = \text{Bending stress, lbs/in}^2 \]
\[ E_M = \text{Modulus of elasticity, lb/in}^2 \]
\[ f = \text{Spring deflection, in} \]
\[ t = \text{Material thickness, in} \]
\[ L = \text{Active spring length, in} \]
The failure rate of a beam spring is determined using the following equation:

\[
\lambda_{SP} = \lambda_{SP,B} \left( \frac{S}{T_s} \right)^3 \cdot C_{CS} \cdot C_R \cdot C_M
\]  

(4-11) ref

A generalized equation that adjusts the base failure rate of a beam spring considering anticipated operating conditions can be established:

\[
\lambda_{SP} = \lambda_{SP,B} \cdot C_E \cdot C_t \cdot C_L \cdot C_f \cdot C_Y \cdot C_{CS} \cdot C_R \cdot C_M
\]  

(4-22)

Where:

- \( \lambda_{SP} \) = Failure rate of beam spring, failures/million hours
- \( \lambda_{SP,B} \) = Base failure rate for beam spring, 4.4 failures/million hours
- \( C_E \) = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)
- \( C_t \) = Multiplying factor which considers the effect of material thickness on the base failure rate (See Figure 4.14)
- \( C_L \) = Multiplying factor which considers the effect of spring length on the base failure rate (See Figure 4.20)
$C_f = \text{Multiplying factor which considers the effect of spring deflection on the base failure rate (See Figure 4.16)}$

$C_Y = \text{Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)}$

$C_{CS} = \text{Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.18)}$

$C_R = \text{Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)}$

$C_M = \text{Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)}$
Wire Diameter Multiplying Factor, $C_{DW}$

$$C_{DW} = \left( \frac{D_W}{0.085} \right)^3$$

**Figure 4.8** Multiplying Factor for Wire Diameter
Figure 4.9 Multiplying Factor for Spring Coil Diameter

\[ C_{DC} = \left( \frac{0.58}{D_c} \right)^6 \]
$C_N = \left( \frac{14}{N_a} \right)^3$

**Figure 4.10 Multiplying Factor for Number of Coils in a Spring**
Spring Deflection, $L_1 - L_2$, inches

\[ C_L = \left( \frac{L_1 - L_2}{1.07} \right)^3 \]

**Figure 4.11** Multiplying Factor for Spring Deflection
Spring Index, \( r \)

Stress Concentration Multiplying Factor, \( C_K \)

\[
C_K = \left( \frac{K_w}{1.219} \right)^3
\]

Where:

\[
K_w = \frac{4r - 1}{4r - 4} + \frac{0.616}{r}
\]

and

\[
r = \frac{D_c}{D_w}
\]

\( D_c \) = Coil Diameter, inches

\( D_w \) = Wire Diameter, inches

Figure 4.12 Multiplying Factor for Stress Concentration Factor
Spring Deflection Multiplying Factor, $C_L$, is given by:

$$C_L = \left( \frac{\theta}{0.667} \right)^3$$

Where: $\theta = \text{Angular rotation, revolutions}$

Figure 4.13 Multiplying Factor for Deflection of a Torsion Spring
Figure 4.14  Multiplying Factor for Material Thickness

\[ C_t = \left( \frac{t}{0.025} \right)^3 \]
\[ C_D = \left( \frac{1.20}{OD} \right)^6 \]

Where: \( OD \) = Outside Diameter of Belleville, Curved or Wave Washer, inches

**Figure 4.15  Multiplying Factor for Washer Size**
Washer Deflection, $f$, inches

Washer Deflection Multiplying Factor, $C_f$

$C_f = \left( \frac{f}{0.055} \right)^3$

Figure 4.16 Multiplying Factor for Washer Deflection
$C_s = \left( \frac{6}{\pi \ln R} \right)^3 \left( \frac{R - I}{\ln R} - 1 \right) \left( \frac{R - I}{2} \right) \left( \frac{(R - I)^2}{R^2} \right)$

Where:  $R = \frac{\text{outside diameter}}{\text{inside diameter}}$

**Figure 4.17  Multiplying Factor for Belleville Washer Compressive Stress**
For CR ≤ 30 cycles/min, $C_{CS} = 0.100$

For 30 cycles/min < CR ≤ 300 cycles/min, $C_{CS} = \frac{CR}{300}$

For CR > 300 cycles/min, $C_{CS} = \left(\frac{CR}{300}\right)^3$

Where: CR = Spring cycle rate, cycles/min

Figure 4.18 Multiplying Factor for Spring Cycle Rate
Coil Diameter Multiplying Factor, \( C_{DC} \)

Where \( D_c \) = Coil diameter, inches

\[
C_{DC} = \left( \frac{0.58}{D_c} \right)^3
\]

Figure 4.19 Multiplying Factor for Spring Coil Diameter
(Torsion Springs)
Spring Length, $L$, inches

Multiplying Factor, $C_L$

$$C_L = \left(\frac{1.20}{L}\right)^6$$

Figure 4.20  Multiplying Factor for Spring Length
Table 4-2. Moduli of Rigidity and Elasticity for Typical Spring Materials

<table>
<thead>
<tr>
<th>MATERIAL</th>
<th>MODULUS OF RIGIDITY (G&lt;sub&gt;M&lt;/sub&gt;) lbs/in&lt;sup&gt;2&lt;/sup&gt; x 10&lt;sup&gt;6&lt;/sup&gt;</th>
<th>C&lt;sub&gt;G&lt;/sub&gt;</th>
<th>MODULUS OF ELASTICITY (E&lt;sub&gt;M&lt;/sub&gt;) lbs/in&lt;sup&gt;2&lt;/sup&gt; x 10&lt;sup&gt;6&lt;/sup&gt;</th>
<th>C&lt;sub&gt;E&lt;/sub&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ferrous:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Music Wire</td>
<td>11.8</td>
<td>1.08</td>
<td>29.0</td>
<td>1.05</td>
</tr>
<tr>
<td>Hard Drawn Steel</td>
<td>11.5</td>
<td>1.00</td>
<td>28.5</td>
<td>1.00</td>
</tr>
<tr>
<td>Chrome Steel</td>
<td>11.2</td>
<td>0.92</td>
<td>29.0</td>
<td>1.05</td>
</tr>
<tr>
<td>Silicon-Manganese</td>
<td>10.8</td>
<td>0.83</td>
<td>29.0</td>
<td>1.05</td>
</tr>
<tr>
<td>Stainless, 302, 304, 316</td>
<td>10.0</td>
<td>0.67</td>
<td>28.0</td>
<td>0.98</td>
</tr>
<tr>
<td>Stainless 17-7 PH</td>
<td>10.5</td>
<td>0.76</td>
<td>29.5</td>
<td>1.04</td>
</tr>
<tr>
<td>Stainless 420</td>
<td>11.0</td>
<td>0.88</td>
<td>29.0</td>
<td>1.05</td>
</tr>
<tr>
<td>Stainless 431</td>
<td>11.4</td>
<td>0.97</td>
<td>29.5</td>
<td>1.11</td>
</tr>
<tr>
<td>Non-Ferrous:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Spring Brass</td>
<td>5.0</td>
<td>0.08</td>
<td>15.0</td>
<td>0.15</td>
</tr>
<tr>
<td>Phosphor Bronze</td>
<td>6.0</td>
<td>0.14</td>
<td>15.0</td>
<td>0.15</td>
</tr>
<tr>
<td>Beryllium Copper</td>
<td>7.0</td>
<td>0.23</td>
<td>17.0</td>
<td>0.21</td>
</tr>
<tr>
<td>Inconel</td>
<td>10.5</td>
<td>0.76</td>
<td>31.0</td>
<td>1.09</td>
</tr>
<tr>
<td>Monel</td>
<td>9.5</td>
<td>0.56</td>
<td>26.0</td>
<td>0.76</td>
</tr>
</tbody>
</table>

NOTE: Modulus G<sub>M</sub> is used for compression and extension springs; modulus E<sub>M</sub> is used for torsion springs, flat springs and spring washers.

\[
C_G = \left( \frac{G_M}{11.5 \times 10^6} \right)^3 \quad C_E = \left( \frac{E_M}{28.5 \times 10^6} \right)^3
\]

where:  
\( G_M \) = Modulus of Rigidity (lbs/in<sup>2</sup>)  
\( E_M \) = Modulus of Elasticity (lbs/in<sup>2</sup>)
Table 4-3. Material Tensile Strength Multiplying Factor, \( C_Y \)

<table>
<thead>
<tr>
<th>MATERIAL</th>
<th>TENSILE STRENGTH, ( T_S ) lbs/in(^2) ( \times 10^3 )</th>
<th>( C_Y )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brass</td>
<td>110</td>
<td>5.15</td>
</tr>
<tr>
<td>Phosphor Bronze</td>
<td>125</td>
<td>3.51</td>
</tr>
<tr>
<td>Monel 400</td>
<td>145</td>
<td>2.25</td>
</tr>
<tr>
<td>Inconel 600</td>
<td>158</td>
<td>1.74</td>
</tr>
<tr>
<td>Monel K500</td>
<td>175</td>
<td>1.28</td>
</tr>
<tr>
<td>Copper-Beryllium</td>
<td>190</td>
<td>1.00</td>
</tr>
<tr>
<td>17-7 PH, RH 950</td>
<td>210</td>
<td>0.74</td>
</tr>
<tr>
<td>Hard Drawn Steel</td>
<td>216</td>
<td>0.68</td>
</tr>
<tr>
<td>Stainless Steel 302, 18-8</td>
<td>227</td>
<td>0.59</td>
</tr>
<tr>
<td>Spring Temper Steel</td>
<td>245</td>
<td>0.47</td>
</tr>
<tr>
<td>Chrome Silicon</td>
<td>268</td>
<td>0.36</td>
</tr>
<tr>
<td>Music Wire</td>
<td>295</td>
<td>0.27</td>
</tr>
</tbody>
</table>

NOTE: These are typical values based on a wire diameter of 0.1 inch. Actual values of tensile strength will vary with wire diameter.

\[
C_Y = \left( \frac{190 \times 10^3}{T_S} \right)^3
\]

where \( T_S \) = Tensile Strength, lbs/in\(^2\)

Table 4-4. Wave Washer Multiplying Factor, \( C_{NW} \)

<table>
<thead>
<tr>
<th>NUMBER OF WAVES</th>
<th>( C_{NW} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>2.78</td>
</tr>
<tr>
<td>4</td>
<td>1.56</td>
</tr>
<tr>
<td>5</td>
<td>1.00</td>
</tr>
<tr>
<td>6</td>
<td>0.69</td>
</tr>
<tr>
<td>7</td>
<td>0.51</td>
</tr>
<tr>
<td>8</td>
<td>0.39</td>
</tr>
</tbody>
</table>

\[
C_{NW} = \left( \frac{5}{NW} \right)^2
\]
4.5 REFERENCES


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5.1 INTRODUCTION

Solenoids are electromechanical devices that convert electrical energy into mechanical motion. Generally this motion is used to move a load a specified distance or rotational angle within a specified time. Linear magnetic solenoids usually produce motion by pulling a plunger into the coil when energized. They can also be equipped with a push rod mounted to one end of the plunger providing a pushing motion when energized. The plunger of the solenoid assembly, also known as the armature, is made of ferrous material to increase magnetism or permeability. Rotary solenoids convert axial motion into a rotary stroke.

Component parts of a solenoid include a coil to carry current and generate ampere turns, an iron shell to provide a magnetic circuit and a movable plunger to act as the working element. Component parts of example linear and rotary solenoids are shown in Figure 1. Electrical current is supplied to the solenoid coil that is wound tightly enough to limit the current drain and sized large enough to provide for adequate heat dissipation. The resulting magnetic field draws the plunger from its unpowered, extended position to a seated position against a backstop or pole piece. Because the linear force on the plunger is nonlinear with position, the force is relatively high immediately adjacent to the seated position and declines rapidly with increased distance from the seated position. Return motion of the solenoid upon deenergizing the coil is provided by the load itself or a return spring. Rotary solenoids normally consist of a plunger and small ball bearings that ride on an inclined plane.
The stroke in linear solenoids or rotation in rotary solenoids is the total plunger travel or rotary shaft motion when electrical power is applied to the solenoid. The force in linear solenoids or torque in rotary solenoids is the load that the solenoid is capable of pulling, pushing, holding or rotating at the start of a specified motion. This load capability will depend on applied voltage, ambient temperature and the solenoid operating duty cycle. The solenoid force or torque will decrease as the solenoid coil temperature increases depending on coil wire resistance. Continuous duty solenoids are considerably larger than intermittent duty devices and provide more pull-and-hold strength. Solenoids that are capable of both intermittent and continuous duty will exhibit force ratings that vary widely with duty cycle.

![Rotary Solenoid](image)

**Figure 1. Component Parts of a Solenoid**

The solenoid may be used to control the motion of various components such as a valve or contactor. Clearance between the coil assembly and the armature assembly must be maintained for proper operation and the design should be evaluated for reliability considering the operating environment. For small solenoids, the armature has minimal effect on the part failure rate. For large solenoid operated assemblies, the procedures in Chapter 20 can be used to evaluate the wear rate of the solenoid plunger. And, if the solenoid is used to control an actuator, the procedures of Chapter 9 can be used to determine the failure rate of the actuator. Chapter 6 contains procedures for evaluating a valve if the solenoid is part of a valve assembly. If a return spring is being used in the solenoid, the procedures in Chapter 4 can be used to determine the reliability of the spring in its operating environment. Chapter 7 contains procedures for evaluating the rotating bearing if a rotary solenoid is being evaluated. Section 5.6 contains the procedures for evaluating the contactor if the solenoid is part of a relay configuration.

The reliability of a typical solenoid assembly depends on the construction of the coil assembly, length of the stroke, and the environment in which it operates. The failure rate of the solenoid assembly is dependent upon manufacturing quality associated with the assembly of the coil in relation to the armature as well as upon operating...
environment including shock, vibration and corrosive environment. The failure rate of
the solenoid assembly can be determined using the procedures in Section 5.3. The
maximum specified cycling rate of the solenoid should be determined and compared
with the potential operating rate to make sure the coil will not be overheated in its
operating environment. When a solenoid is energized by the voltage source, heat is
generated which increases the temperature of the coil. This temperature rise may have
some undesired effects, since resistance of the coil winding increases with temperature,
which in turn, reduces electrical current. This reduction in current reduces the force
output. An extreme increase in temperature can result in damage to the winding.
Usually the limiting factor for operating temperature is the rated temperature of the
solenoid insulating material (see Section 5.4).

5.2 FAILURE MODES

The primary failure modes of a solenoid inductor (coil) assembly include one or
more winding shorts or an open coil usually caused by overheating. Heat in a solenoid
is a function of power and the time during which time is applied. Heat can be dissipated
by controlling the air flow, by mounting the solenoid on a surface large enough to
dissipate the energy (heat sink), or introduction of alternate cooling methods. Table 5-1
provides some typical failure modes of the solenoid assembly and contactor, if
applicable. Solenoids are enclosed within a steel housing to form part of the magnetic
circuit and to provide structural integrity. It is important to review the mounting of the
solenoid assembly to assure an adequate heat sink for high temperature operations.
Usually a mounting surface area of ten times the area of the solenoid should be
maintained.

Solenoids have a maximum ON time for a given duty cycle, wattage and power
input so it is important to consider the duty cycle and wattage when the solenoid is
pulsed repeatedly. If the ON time of the solenoid is more than 5 minutes during a single
pulse, it should be considered a continuous duty (100% duty cycle) solenoid. It is
important that the solenoid reaches a sufficient cool-down time before being
reenergized.

5.3 FAILURE RATE OF SOLENOID ASSEMBLY

The failure rate of the solenoid can be estimated from the following equation:

\[ \lambda_{SO} = \lambda_{SO,B} C_T C_K \] (5-1)

Where: \[ \lambda_{SO} = \text{Failure rate of a solenoid in failures/million hours} \]
\[ \lambda_{SO,B} = \text{Base failure rate of solenoid, 2.77 failures/million operations} \]

\[ C_T = \text{Temperature multiplying factor, See Figure 5.2} \]

\[ C_K = \text{Application multiplying factor, See Table 5-2} \]

### Table 5-1. Typical Failure Modes of a Solenoid and Contactor

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE MECHANISM</th>
<th>FAILURE CAUSE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coil burnout</td>
<td>Inrush current causes coil overheating and burnout</td>
<td>Mechanical jamming of plunger</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Insufficient heat sink area for solenoid</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Supply voltage interruption resulting in inductive surge</td>
</tr>
<tr>
<td></td>
<td>Heat builds up faster than it can be dissipated</td>
<td>Excessive cycling rate</td>
</tr>
<tr>
<td>Failure to operate</td>
<td>Increase in coil resistance preventing solenoid closure</td>
<td>Excessive ambient temperature</td>
</tr>
<tr>
<td></td>
<td>Shorted coil at lead wires</td>
<td>Excessive moisture</td>
</tr>
<tr>
<td>Open inductor winding</td>
<td>Open lead at termination</td>
<td>Coil voltage overload, vibration</td>
</tr>
<tr>
<td>Damaged contactor</td>
<td>Contactor arcing</td>
<td>Excessive load voltage</td>
</tr>
<tr>
<td>Armature (plunger) failure</td>
<td>Mismatch of solenoid force and load</td>
<td>Excessive plunger force creating hammering</td>
</tr>
<tr>
<td>Poor response time (pull-in time)</td>
<td>Insufficient solenoid force with respect to load</td>
<td>Jammed return spring</td>
</tr>
<tr>
<td>Poor release time (drop-out time)</td>
<td>Insufficient load or spring force to release plunger</td>
<td>Damaged / jammed spring or loss of load force</td>
</tr>
</tbody>
</table>

Since an inductor (coil) consists of a number of turns of wire it will have some small amount of direct current resistance. This copper loss of the inductor can be calculated by multiplying the square of the current in the inductor by the resistance of the winding (I^2R). In addition to copper loss, an iron core coil will have hysteresis and eddy-current losses.
losses. Hysteresis loss is due to power that is consumed in reversing the magnetic field of the inductor core each time the direction of current in the inductor changes. Eddy-current loss is due to heating of the core by circulating currents that are induced in the core by the magnetic field around the turns of the coil. All these losses dissipate power in the form of heat. Inductor manufactures publish these power losses in their product specification sheets.

Manufacturers also rate their inductors in terms of insulation rating. Common ratings include:

- Class A rated 105 °C
- Class B rated 130 °C
- Class F rated 155 °C
- Class H rated 180 °C
- Class C rated 220 °C

5.4 TEMPERATURE MULTIPLYING FACTOR, $C_T$

For a specific solenoid stroke or rotation, the solenoid force or torque will decrease as the solenoid coil temperature increases. A solenoid coil will tend to achieve a stabilized coil temperature when operated at the manufacturer’s rated power and duty cycle. When operated under standard operating conditions, the coil insulation will not be exceeded and the base failure rate can be assumed. It is important to determine the power dissipation and surface area of the solenoid to assure that the insulation temperature rating is not exceeded. Heat in a solenoid is a function of power and the time during which power is applied. Heat can be dissipated by controlling the air flow, by mounting the solenoid on a surface (heat sink) large enough to dissipate the energy or other cooling method.

Wear of moving parts and solenoid electrical contacts can change mechanical tolerances, resulting in increased electrical current through the solenoid coil when the solenoid is energized and possible premature failure. Energizing the solenoid at nominal design voltage does not normally impart any significant stress on the solenoid. However, after degradation from electrical stress, energizing the solenoid at nominal design voltages may result in solenoid failure. A temperature rise of the solenoid coil must be carefully considered during the analysis of design reliability. Elevated ambient temperatures must be projected; for example, relay cabinets and the air flow design. Duty cycle and maximum on time have a significant effect on coil temperature. Coil temperature in turn affects the solenoid force or torque capability and its failure rate. For a specific solenoid stroke or rotation, the solenoid force or torque will decrease as the solenoid coil temperature increases.
The following equation can be used to estimate the temperature multiplying factor:

\[
C_T = \left( \frac{1}{1.5 I} \right)^3
\]  

(5-2)

where:

\[
t = \frac{(T_R - T_D) - 20}{10}
\]

The effect of temperature on the temperature multiplying factor is shown in Figure 5.2. As current flows through the coil, the wire becomes warmer and its resistance increases. The gauge of the wire in the coil determines how much self-heating occurs. The thermal mass of the solenoid helps to dissipate the heat. The following equation predicts the coil’s self-heating

\[
T = T_A + 10^{(33 S / (I^2 A) + 234) + (234 + T_A)}
\]  

(5-3)

Where

- \( T \) = Temperature of the wire after \( S \) seconds,
- \( T_A \) = Ambient temperature
- \( S \) = Application time
- \( A \) = Conductor area, mils
- \( I \) = Application current, amps

When a solenoid is actuated, current is applied to the coil. A by-product of this is temperature rise in the coil. When a solenoid is energized continuously, heating of the coil increases until a saturation level is reached which is equal to the ambient heat radiation. When a solenoid is operated in an intermittent (on-off) manner, the on-time becomes critical if higher voltages (thus currents) are applied to the coil. These higher currents cause higher coil temperatures which can exceed the cooling capacity of the solenoid through ambient radiation. This heat rise can thermally destroy the solenoid’s coil. To avoid such a catastrophic failure, there is a maximum “on” time which is the longest time a solenoid can be energized without thermal damage.

\[
Duty \ Cycle(\%) = \frac{ON \ Time}{ON \ Time + Off \ Time} \times 100
\]

Solenoids have a maximum ON time for a given duty cycle, wattage and power input. For example, if a solenoid is energized for one second out of four (25% duty cycle), its ON time of one second will cause no damage. However, if the solenoid is
energized for 10 minutes out of every 40 minutes (25% duty cycle), the coil could be damaged if not rated for continuous duty.

### 5.5 APPLICATION MULTIPLYING FACTOR, \( C_k \)

The application multiplying factor is dependent on the length of stroke or degree of rotation and the side loading or torque. Table 5-2 provides the recommended value for this multiplying factor.

#### Table 5-2. Solenoid Application Multiplying Factor, \( C_k \)

<table>
<thead>
<tr>
<th>Operating Environment</th>
<th>Normal load or torque, minimum stroke or rotation</th>
<th>Normal load or torque, extended stroke or rotation</th>
<th>High load or torque, minimum stroke or rotation</th>
<th>High load or torque, extended stroke or rotation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uniform</td>
<td>1.1</td>
<td>1.3</td>
<td>1.2</td>
<td>1.4</td>
</tr>
<tr>
<td>Light shock</td>
<td>1.2</td>
<td>1.4</td>
<td>1.3</td>
<td>1.5</td>
</tr>
<tr>
<td>Medium shock</td>
<td>1.3</td>
<td>1.5</td>
<td>1.4</td>
<td>1.6</td>
</tr>
<tr>
<td>Heavy shock</td>
<td>1.4</td>
<td>1.6</td>
<td>1.5</td>
<td>1.7</td>
</tr>
</tbody>
</table>

### 5.6 FAILURE RATE OF CONTACTOR ASSEMBLY

Contactor life is usually limited by the contacts depending on physical, chemical and electrical phenomenon. Failure of an electrical contact can usually be determined by an increase in contact resistance to approximately twice the initial value. The failure rate of the contactor can be written as (Reference 28):

\[
\lambda_c = \lambda_{c,B} \cdot V^m \cdot I^n
\]  

(5-3)

Where:

- \( \lambda_c \) = Failure rate of contactor assembly, failures/million operations
- \( \lambda_{c,B} \) = Base failure rate of contactor assembly, failures/million operations
- \( V \) = Voltage across contactor assembly, volts
- \( I \) = Current, amperes
\[ m = \text{Voltage constant} \]
\[ n = \text{Current constant} \]

A more general equation can be written for AC resistive loads (Reference 28):

\[ \lambda_C = \lambda_{C,B} \cdot C_V \cdot C_I \]  \hspace{1cm} (5-7)

Where:
\[ \lambda_C = \text{Failure rate of contactor assembly, failures/million operations} \]
\[ \lambda_{C,B} = \text{Base Failure of contactor assembly, resistive load, 1.10 failures/million operations} \]
\[ C_V = \text{Multiplying factor considering contactor voltage (See Figure 5.3)} \]
\[ C_I = \text{Multiplying factor considering contactor current (See Figure 5.4)} \]

For AC inductive loads, the power factor must be considered, modifying Equation (5-7) as follows (Reference 28):

\[ \lambda_C = \lambda_{C,B} \cdot C_V \cdot C_I \cdot C_{PF} \]  \hspace{1cm} (5-8)

Where:
\[ \lambda_{C,B} = \text{Base failure rate of contactor assembly, inductive load, 3.60 failures/million operations} \]
\[ C_V = \text{Multiplying factor considering contactor voltage (See Figure 5.3)} \]
\[ C_I = \text{Multiplying factor considering contactor current (See Figure 5.4)} \]
\[ C_{PF} = \text{Multiplying factor considering the power factor (See Figure 5.5)} \]

DC loads generate greater arcing across the contacts than do AC loads. The failure rate equation for a contactor with DC loads is written as follows (Reference 28):

\[ \lambda_C = \lambda_{C,B} \cdot C_V \cdot C_I \]  \hspace{1cm} (5-9)

Where:
\[ \lambda_C = \text{Failure rate of contactor assembly, failures/million operations} \]
\[ \lambda_{C,B} = \text{Base Failure of contactor assembly, DC load, 2.5 failures/million operations} \]
\[ C_V = \text{Multiplying factor considering contactor voltage (See Figure 5.6)} \]
\[ C_I = \text{Multiplying factor considering contactor current (See Figure 5.7)} \]


\[ C_T = \left( \frac{1}{1.5^t} \right)^3 \]

where:

\[ t = \frac{(T_R - T_o) - 20}{10} \]

* \( T_o \) = Operating temperature, °C
* \( T_R \) = Rated temperature of the solenoid, °C

**Figure 5.2 Multiplying Factor for Coil Temperature**
\[ C_V = \left( \frac{V_o}{V_r} \right)^{0.75} \]

Where:  
\( V_o \) = Operating voltage, volts  
\( V_r \) = Rated voltage, volts

**Figure 5.3 Multiplying Factor for AC Contactor Voltage**
AC Current Multiplying Factor, $C_I$

Where:

\[ I_o = \text{Operating current, amperes} \]
\[ I_r = \text{Rated current, amperes} \]

\[ C_I = 3.50 \left( \frac{I_o}{I_r} \right)^{1.14} \]

Figure 5.4 Multiplying Factor for AC Contactor Current
$$C_{PF} = 4.0 \left( PF \right)^2$$

Figure 5.5 Multiplying Factor for Power Factor
$C_r = \left( \frac{V_o}{V_r} \right)^{1.33}$

Where:  
$V_o = \text{Operating voltage, volts}$
$V_r = \text{Rated voltage, volts}$

**Figure 5.6 Multiplying Factor for DC Contactor Voltage**
$C_I = 4.20 \left( \frac{I_o}{I_r} \right)^{1.30}$

Where:

- $I_o$ = Operating current, amperes
- $I_r$ = Rated current, amperes

**Figure 5.7 Multiplying Factor for DC Contactor Current**
5.5 REFERENCES

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment".


81. TD 84-3 R&M Data for Industrial Plants, A.P. Harris and Associates, 1984

6.1 INTRODUCTION

This chapter contains failure rate models for fluid valve assemblies which can be used to support the development of mechanical equipment and provide a reliability estimate for a new design or a proposed design modification. The models are intended to focus attention on further design analyses which should be accomplished to assure the allocated reliability of the valve in its intended operating environment.
A typical valve assembly is shown in Figure 6.1. After the failure rates are determined for each component part, the results are summed to determine the failure rate of the total valve assembly:

\[ \lambda_{VA} = \lambda_{PO} + \lambda_{SE} + \lambda_{SP} + \lambda_{SO} + \lambda_{HO} \]  \hspace{1cm} (6-1)

for a poppet type valve, or

\[ \lambda_{VA} = \lambda_{SV} + \lambda_{SE} + \lambda_{SP} + \lambda_{SO} + \lambda_{HO} \]  \hspace{1cm} (6-2)

for a sliding-action valve.

Where:

- \( \lambda_{VA} \) = Failure rate of total valve assembly in failures/million operations
- \( \lambda_{PO} \) = Failure rate of poppet assembly in failures/million operations as derived from Section 6.3
- \( \lambda_{SV} \) = Failure rate of sliding action valve assembly in failures/million operations as derived from Section 6.4
- \( \lambda_{SE} \) = Failure rate of the seals in failures/million operations as derived
6.2 FAILURE MODES OF VALVE ASSEMBLIES

Failure modes of a valve assembly are identified from the analyst’s knowledge of the valve’s design characteristics and the intended operating environment. Table 6-1 provides a summary of valve failure modes to be considered. Many valve assemblies are uniquely designed for special applications and a more detailed analysis is often required for those failure modes identified as critical or where results of the analysis indicate that an additional investigation is warranted. Failure rate models in this Handbook are based on the identification of failure modes with multiplying factors modifying a base failure rate considering the particular design characteristics and intended operating environment. This approach permits the estimate of an occurrence probability for individual failure modes.

The primary failure mode of the valve seating surface is wear caused by the impact of contaminants. In a gas valve contaminants include liquids under pressure. A valve seat will wear where the sealing element contacts the seat. The local effect is insufficient seating and internal leakage. However, assuming the correct seat material for the application and minimal contaminants, the failure rate of the valve due to seating wear should be minimal. The sealing element may be a poppet, plate or ring. Possible failure modes for the sealing element include fatigue failure caused by repeated impact on the valve seat, sudden changes in differential pressure across the valve, and contaminants causing corrosion.

The failure rate of a compression spring is usually very low depending on the application. However, it is important to consider the operating environment of the valve when estimating the failure rate of the return spring. The spring material in a gas valve may not be resistant to all gas compositions. Likewise, in any valve it is important to consider the compatibility of the fluid and potential contaminants with the spring material. Failure modes of the stop plate are the same as the valve seat. The wear area of the stop plate is where the spring contacts the plate. This wear is minimal assuming correct hardness of the plate material.

For a sliding action valve the main failure modes to be considered include internal leakage caused by a worn spool and slow response of the valve caused by a sticking piston. As with a poppet valve the chief cause of a worn spool or slow response is the
presence of contaminants. Assistance with the valve parts such as O-rings and solenoids can be found in the appropriate chapters of this Handbook.

After the basic parts of the valve are evaluated for reliability, the dynamics of the valve must be considered. Again, the spring is probably most critical to estimating the failure rate of the valve. For example, a sticking sealing element caused by a misaligned spring resulting in poor dynamic response is a potential failure mode that needs to be considered. Pressure differential, operating temperature, viscosity and use rate are variables that determine the failure rate of the valve. Surface finish of the poppet, allowable leakage and a contaminant factor considering flow rate are used to estimate the failure rate of the poppet due to wear from contaminants. The failure rates of valve seals, the return spring and the solenoid driver are determined separately. See Chapters 3, 4 and 5.

Typical failure modes for a valve assembly are listed in Table 6-1. It should be noted that the failure modes, failure causes and failure effects may be interchanged depending upon the type of analysis being performed. For example, a functional analysis will tend to identify those entries in Table 6-1 under local effects as the failure mode while a very detailed hardware analysis would result in the identification of those entries under failure cause as the failure mode.

When performing a Failure Mode, Effects and Criticality Analysis (FMECA), probabilities of occurrence for each listed failure mode on the FMECA worksheet are derived from the multiplying factors. See Section 6.6

6.3 FAILURE RATE MODEL FOR POPPET ASSEMBLY

The term poppet refers to those valves in which the valve element travels perpendicular to a plane through the seating surface. The poppet valve element is used in flow control, pressure control and directional control valves. In a poppet valve, a relatively large flow area is provided with short travel of the poppet. This characteristic simplifies the actuator requirements and permits the use of solenoids and diaphragms, which are characteristically short stroke devices.

Figure 6.2 illustrates the operation of a simple poppet valve. The valve consists primarily of a movable poppet which closes against a valve seat. The valve may be actuated manually or by electrical, mechanical or pneumatic means. This section of the manual discusses the valve mechanism itself. Refer to Chapter 5 for procedures to evaluate the reliability of the solenoid and to other appropriate sections of the manual to determine the reliability of other components of the actuation mechanism. In the closed position, fluid pressure on the inlet side tends to hold the valve tightly closed. A force applied to the top of the valve stem opens the poppet and allows fluid to flow through the valve.
Table 6-1. Failure Modes for a Valve Assembly

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>LOCAL EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seal leakage</td>
<td>- Embrittlement</td>
<td>- Internal or external valve leakage</td>
</tr>
<tr>
<td></td>
<td>- Installation damage</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Wear</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Surface damage</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Distortion</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Dynamic instability</td>
<td></td>
</tr>
<tr>
<td>Worn or damaged poppet seat</td>
<td>- Wear of poppet/seat Assembly</td>
<td>- Poppet not seating properly causing internal leakage and low/erratic pressure drop</td>
</tr>
<tr>
<td></td>
<td>- Contaminants</td>
<td></td>
</tr>
<tr>
<td>Damaged valve stem</td>
<td>- Vibration, shock</td>
<td>- Poor valve response</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Failure to open and/or close</td>
</tr>
<tr>
<td>Worn or damaged spool</td>
<td>- Contaminants</td>
<td>- Internal leakage</td>
</tr>
<tr>
<td></td>
<td>- Misalignment</td>
<td></td>
</tr>
<tr>
<td>Sticking valve piston in main valve body</td>
<td>- Contaminants</td>
<td>- Low/erratic pressure drop</td>
</tr>
<tr>
<td></td>
<td>- Loss of lubrication</td>
<td>- Slow operating response</td>
</tr>
<tr>
<td></td>
<td>- Air entrapment</td>
<td>- Valve immobile</td>
</tr>
<tr>
<td></td>
<td>- Excessively high temperature</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Structural interference</td>
<td></td>
</tr>
<tr>
<td>Broken spring or damaged spring ends</td>
<td>- Fatigue</td>
<td>- Unable to adjust or maintain pressure</td>
</tr>
<tr>
<td>Inoperative solenoid assembly</td>
<td>- Open coil winding</td>
<td>- Valve fails to open or close</td>
</tr>
<tr>
<td></td>
<td>- Misalignment of solenoid with respect to spool or poppet stem</td>
<td></td>
</tr>
<tr>
<td>External leakage</td>
<td>- Contaminants</td>
<td>- Poppet Stem Wear</td>
</tr>
<tr>
<td>Cracked fitting/housing</td>
<td>- Fatigue</td>
<td>- External leakage</td>
</tr>
<tr>
<td></td>
<td>- External shock</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Vibration</td>
<td></td>
</tr>
</tbody>
</table>
Figure 6.2 Poppet Valve Assembly

The poppet fits into the center bore of the seat. The seating surfaces of the poppet and the seat are lapped or closely machined so that the center bore will be sealed when the poppet is seated. An O-ring is usually installed on the stem of the poppet to prevent leakage past this portion of the poppet assembly.

Table 6-2 is a list of typical failure modes, mechanisms and causes for a poppet assembly. A review of failure rate data suggests the following characteristics be included in the failure rate model for poppet assemblies:

- Leakage requirement
- Material hardness
- Surface irregularities
- Fluid viscosity
- Fluid/material compatibility
- Fluid pressure
- Physical size of poppet/seat
- Q.C./manufacturing processes
- Contamination level
- Utilization rate

Table 6-2. Failure Rate Considerations for a Poppet Assembly

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE MECHANISMS</th>
<th>FAILURE CAUSES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal Leakage</td>
<td>Worn poppet/seat</td>
<td>- Contaminants</td>
</tr>
<tr>
<td>Poor Response</td>
<td>Sticking/jammed poppet assembly</td>
<td>- Side Loading</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Incorrect spring pressure</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Contaminants</td>
</tr>
<tr>
<td>External Leakage</td>
<td>Wear of poppet stem</td>
<td>- Contaminants</td>
</tr>
</tbody>
</table>
A new poppet assembly (or fairly new if some initial deformation exists) may be expected to have a sufficiently smooth surface finish for the valve to meet internal leakage specifications. However, after some period of time contaminants will cause wear of the poppet/seat assembly until the leakage rate is beyond tolerance. This leakage rate, at which point the valve is considered to have failed, will depend upon the application.

A failure rate equation for a poppet assembly is dependent upon the ratio of actual leakage rate to that allowable under conditions of usage. This rate, based on Navy Maintenance and Material Management (3-M) data can be expressed as follows:

\[
\lambda_{PO} = \lambda_{PO,B} \left( \frac{Q_a}{Q_f} \right)
\]  

(6-3)

Where:

- \( \lambda_{PO} \) = Failure rate of the poppet assembly, failures/million operations
- \( \lambda_{PO,B} \) = Base failure rate for poppet assembly, failure/million operations
- \( Q_a \) = Leakage rate, in \( ^3/\text{min} \)
- \( Q_f \) = Leakage rate considered to be valve failure, in \( ^3/\text{min} \)

The allowable leakage, \( Q_f \), is determined from design drawings, specifications or knowledge of component applications. The actual leakage rate, \( Q_a \), is determined from the following equation (Reference 22):

\[
Q_a = \frac{2 \times 10^2 D_M f^3 (P_1^2 - P_2^2) K_1}{V_a L_w (S_s)^{1.5}}
\]  

(6-4)

Where:

- \( Q_a \) = Actual fluid leakage, in \( ^3/\text{min} \)
- \( D_M \) = Mean seat diameter, in
- \( f \) = Mean surface finish of opposing surfaces, in
- \( P_1 \) = Upstream pressure, lb/in\(^2 \)
\[ P_2 = \text{Downstream pressure, lb/in}^2 \]
\[ \nu_a = \text{Absolute fluid viscosity, lbf-min/in}^2 \]
\[ L_W = \text{Radial seat land width, in} \]
\[ S_S = \text{Seat stress, lb/in}^2 \]
\[ K_I = \text{Constant which considers the impact of contaminant size, hardness and quantity of particles} \]

Failure rate of the poppet assembly will be dependent upon leakage rate and those factors which influence the deterioration of surface finish such as rate of cycling, material properties and contaminants. Deterioration of the poppet and seat by contaminants is dependent upon material properties and the number of contaminants, and that part of the time the poppet is open and subject to contaminants under fluid pressure.

A contamination factor can be derived from the following equation:

\[
Z = \left( \text{function of} \right) \left[ \alpha, n, Q, d, T \right]
\]  

(6-5)

Where:

- \( Z \) = Poppet/seat degradation
- \( \alpha \) = Contaminant wear coefficient, in\(^3\)/particle
- \( n \) = Number of contaminant particles/in\(^3\)
- \( Q \) = Flow rate, in\(^3\)/min
- \( d \) = Ratio of time the poppet is open to total operating time
- \( T \) = Temperature of operation, °F

Table 6-5 provides typical quantities of contaminants for use in establishing a multiplying factor. By normalizing the equation to those values for which historical failure rate data are available the following model can be derived:

\[
\lambda_{PO} = \lambda_{PO,B} \cdot C_P \cdot C_Q \cdot C_F \cdot C_v \cdot C_N \cdot C_S \cdot C_{DT} \cdot C_{SW} \cdot C_W
\]  

(6-6)

Where:

- \( \lambda_{PO} \) = Failure rate of poppet assembly in failures/million operations
- \( \lambda_{PO,B} \) = Base failure rate of poppet assembly, 1.40 failures/million operations
\[ C_P = \text{Multiplying factor which considers the effect of fluid pressure on the base failure rate (See Figure 6.6)} \]
\[ C_Q = \text{Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 6.7 or Figure 6.8)} \]
\[ C_F = \text{Multiplying factor which considers the effect of surface finish on the base failure rate (See Figure 6.9)} \]
\[ C_V = \text{Multiplying factor which considers the effect of fluid viscosity and temperature on the base failure rate (See Table 6-6 and Figure 6.15)} \]
\[ C_N = \text{Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 6-5)} \]
\[ C_S = \text{Multiplying factor which considers the effect of the seat stress on the base failure rate (See Figure 6.9)} \]
\[ C_{DT} = \text{Multiplying factor which considers the effect of the seat diameter on the base failure rate (See Figure 6.10)} \]
\[ C_{SW} = \text{Multiplying factor which considers the effect of the seat land width on the base failure rate (See Figure 6.11)} \]
\[ C_W = \text{Multiplying factor which considers the effect of flow rate on the base failure rate (See Figure 6.14)} \]

The following paragraphs provide background information on those parameters included in the model.

### 6.3.1 Fluid Pressure

Figure 6.6 contains the fluid pressure multiplying factors for use in the model. Valves having high response characteristics and consequently a high poppet velocity will incur large impact loading which tends to reduce the life expectancy of the valve. As with any piece of mechanical equipment, the higher the structural loads the shorter the life. Pressure forces arise from any net pressure unbalance acting on the valve element. Depending upon the functional design of the valve, the pressure force may increase, decrease, or virtually have no effect on the actuation force. In an unbalanced valve design such as a conventional poppet, upstream pressure normally acts in a direction to seat the valve so that an increasing upstream pressure will tend to force the valve element tighter against its seat. The use of pressure unbalance to aid in sealing requires a higher actuation force to open the valve. When the size of the valve and/or magnitude of pressure demand excessively large actuation forces, a balanced design and/or piloting are often utilized. In most cases the pressure on the poppet can be assumed to be the system upstream pressure, \( P_1 \), minus the downstream pressure, \( P_2 \).
6.3.2 **Allowable Leakage**

Figure 6.7 shows the allowable leakage multiplying factor for use in equation 6-6. Allowable internal leakage of a poppet design can be obtained from valve specifications. Leakage requirements vary from molecular flow for certain shut-off valves at one extreme to several cubic feet per minute in some inexpensive valves which control water or other inexpensive fluid. Allowable leakage must be evaluated with respect to total mission and operational requirements.

6.3.3 **Surface Finish**

Evaluation of surface finish involves both poppet and seat assemblies. Surface finishes will usually be specified on assembly drawings in terms of microinches or by a manufacturing process. Typical surface finishes for manufacturing processes are provided in Table 6-4. These values are for a finish as initially manufactured and a new valve can be expected to have a sufficiently smooth surface finish to meet internal leakage specifications. However, after some period of time contaminants will cause wear of the poppet/seat assembly until leakage rate is beyond tolerance. This deterioration of surface finish will be influenced by operating temperature and pressure, rate of cycling, loads and material properties.

6.3.4 **Fluid Viscosity**

Viscosity of a fluid is much more dependent on temperature then it is on pressure. For example, when air pressure is increased from 1 atmosphere to 50, its viscosity is only increased by about 10%. In contrast, Figure 6.3 shows the dependence of viscosity on temperature for some common fluids. The graph shows how viscosity of liquids decreases with temperature while that of gases increases with temperature. Multiplying factors for viscosities of typical fluids are provided in Table 6-6. Multiplying factors for other fluids are determined from the table by a knowledge of viscosity at the applicable fluid temperature. Viscosity for a specific fluid is obtainable from many reference sources. If the value located is in terms of kinematic viscosity, multiply the value by the specific gravity (density) at the desired temperature to determine the dynamic viscosity.

6.3.5 **Contamination Sensitivity**

Cleanliness of the system and of the fluid medium has a direct effect upon the operation and life of a poppet valve. Contaminants can clog or jam the poppet and cause excessive leakage in metal-to-metal seated valves. Particulate matter in gaseous media, especially in the lighter gases such as helium, can be extremely destructive to internal parts, particularly seats, because of the very high velocity that can be attained under sonic conditions.
The analysis of particle sizes includes the determination of upstream filter size, the filter maintenance schedule, the number of upstream components between the valve and filter, and the number of particles likely to be encountered at the poppet/seat assembly. **Table 6-5** lists typical quantities of contaminants for use in determining the multiplying factor.

![Dynamic Viscosities of Various Fluids](image)

**Figure 6.3 Dynamic Viscosities of Various Fluids**

### 6.3.6 Contact Pressure

The force applied by the poppet against the seat is found by actual measurement or design specifications. In the typical valve seat example of **Figure 6.4**, the seat area $A_{ST}$ can be computed as follows:
\[ A_{ST} = \pi \left( D_1^2 - D_2^2 \right) / 4 \]  \quad (6-7)

Where: 
- \( A_{ST} \) = Seat Area, in\(^2\)
- \( D_1 \) = Outside diameter of poppet, in
- \( D_2 \) = Diameter of poppet shaft, in

And the force of the poppet against the seat is:

\[ F_s = A_{ST} \left( P_1 - P_2 \right) \]  \quad (6-8)

Where: 
- \( F_s \) = Force on seat, lb
- \( P_1 \) = Upstream fluid pressure, psi
- \( P_2 \) = Downstream fluid pressure, psi

The seat land area acting as a seal when the valve is closed, is calculated by:

\[ A_{SL} = \pi \left( D_1^2 - D_3^2 \right) / 4 \]  \quad (6-9)

Where: 
- \( A_{SL} \) = Seat land area, in\(^2\)
- \( D_3 \) = Inside diameter of valve outlet

Then the expression for contact pressure, \( S_S \), is the force applied to the seat divided by the seat land area:

\[ S_S = \frac{F_s}{A_{SL}} \]  \quad (6-10)

Therefore:
The minimum contact pressure to prevent leakage for most materials is approximately three times the fluid pressure. In Equation (6-4), leakage varies inversely with the seat stress raised to the 1.5 power. Therefore, a multiplying factor for the effect of contact pressure on the valve base failure rate can be derived as follows:

\[ C_S = 0.26 \left( \frac{9000}{S_S} \right)^{1.5} \quad (6-12) \]

Figure 6.9 provides the multiplying factors for different values of contact pressure.

Figure 6.4 Typical Poppet Valve Seat Configuration
6.3.7 Physical Dimensions

The poppet diameter, seat diameter, and seat land width are shown in Figure 6.4. Figures 6.10 and 6.11 provide multiplying factors for seat diameter and land width.

6.3.8 Operating Temperature

The duty cycle of a poppet valve can vary from several on-off cycles to many hundreds of cycles per hour. Multiple cycling under high pressure or operating temperature decreases the life of the valve. The rate of cycling may be important if the temperature rise, as a result of the operation, becomes significant. The effects of fluid temperature on failure rate are included in the fluid viscosity multiplying factor, $C_\nu$.

6.3.9 Other Considerations

Several failure rate considerations are not specifically included in the model but rather included in the base failure rate. The base failure rate is an average value which reflects field performance data. The following items can be used as a check list to assure that the potential failure mechanisms have been considered:

- Fluid medium considerations which are important in valve designs include the physical properties of the fluid and the compatibility of the fluid with poppet/seat materials. Corrosive fluids will rapidly change the surface finish. The state and physical properties of the fluid become particularly important in determining pressure drop and flow capacity.

- In considering maintenance, requirements for special tasks must be identified. Valve seats should be accessible and easily replaced, preferably without removing the valve from its circuit. When it is necessary to service a valve in the field, care must be exercised to insure that contamination from the work area is not introduced into the valve or system. Requirements for lubrication and adjustments should be minimized to provide high reliability in service use.

- While critical design features are usually based upon one primary fluid, consideration must also be given to secondary fluids with which the valving unit will be required to operate during cleaning and testing operations.

- The location of the valve in the circuit must be considered when considering system failure modes and failure rates. For example, in some circuits a backup control valve is used to permit continued operation in event one valve becomes stuck in the open or closed position.
6.4 FAILURE RATE MODEL FOR SLIDING ACTION VALVES

Sliding action valves consist of a movable spool (a piston with more than one land) within a cylinder. Sliding action valves are usually designed such that the spool slides longitudinally to block and uncover ports in the housing. A rotary spool is sometimes used. Fluid under pressure which enters the inlet port acts equally on both piston areas regardless of the position of the spool. Sealing is accomplished by a very closely machined fit between the spool and the valve body. In sleeve valves the solid piston or spool is replaced by a hollow cylinder with either the inner or outer cylinder serving as the valve element. A typical sliding action valve is shown in Figure 6.5.

The great majority of sliding action valves utilizes axial motion of the valving element, although some designs for special applications use rotating pistons or sleeves. A primary advantage of sliding action valves is the feasibility of obtaining a pressure-balanced design, especially with sleeve or spool configurations. An inherent disadvantage of sliding action valves is leakage, a problem which can only be controlled by close machining or reliable dynamic sealing techniques. Spool valves, for example, are widely used in fluid power applications where perfect internal sealing is not required.

Diametrical spool clearances of approximately 50 microinches are common and surface finishes of 4 to 6 microinches are standard requirements for spools and sleeves. Therefore, contamination tolerance and dirt sensitivity are critical factors in the design and use of sliding action valves, and reliability will be directly affected by dirt particles. Force balances, flow rate and general mechanical operation can be influenced by the presence of contaminants within the valve. Contamination problems include wear of the spool and sleeve until the leakage rate is beyond tolerance. The steps to investigating internal leakage are the same as for the poppet type valve. Table 6-3 is a list of failure modes, mechanisms and causes for spool assemblies. Other failure modes should be identified for the specific application and evaluated to determine the applicability of the failure rate model to the analysis being performed.

Figure 6.5 Sliding Action Valve Assembly
An equation similar to that for poppet valves can be used to predict the reliability of a sliding action valve:

\[
\lambda_{SV} = \lambda_{SV,B} \frac{B^2 D_{SP} \left( P_1^2 - P_2^2 \right)^{1/2}}{Q_f V_a} \mu \alpha \eta
\]

Where:
- \( \lambda_{SV} \) = Failure rate of sliding action valve assembly in failures/million operations
- \( \lambda_{SV,B} \) = Base failure rate = 1.25 failures/million operations
- \( B \) = Spool clearance, in
- \( D_{SP} \) = Spool diameter, in
- \( P_1 \) = Upstream pressure, lb/in\(^2\)
- \( P_2 \) = Downstream pressure, lb/in\(^2\)
- \( V_a \) = Absolute fluid viscosity, lb-min/in\(^2\)
- \( Q_f \) = Leakage rate considered to be device failure, in\(^3\)/min
- \( \mu \) = Friction coefficient
- \( \alpha \) = Contaminant wear coefficient, in\(^3\)/particle
- \( \eta \) = Number of contaminant particles/in\(^3\)

Table 6-3. Failure Rate Considerations for Sliding Action Valve

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE MECHANISMS</th>
<th>FAILURE CAUSES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal leakage</td>
<td>Worn spool/sleeve</td>
<td>- Contaminants</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Side loading</td>
</tr>
<tr>
<td>Poor response</td>
<td>Sticking sleeve assembly</td>
<td>- Side Loading</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Incorrect spring pressure</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Contaminants</td>
</tr>
<tr>
<td>External leakage</td>
<td>Worn gasket/seal</td>
<td>- Contaminants</td>
</tr>
<tr>
<td>Valve port fails to open</td>
<td>Jammed sleeve assembly</td>
<td>- Excessive side loading</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Contaminants</td>
</tr>
</tbody>
</table>
By normalizing the characteristic equation to those values for which historical failure rate data are available, the following model can be derived:

\[
\lambda_{SV} = \lambda_{SV,B} \cdot C_P \cdot C_Q \cdot C_V \cdot C_B \cdot C_{DS} \cdot C_\mu \cdot C_W
\]  

(6-14)

Where:
- \(\lambda_{SV,B}\) = Base failure rate, 1.25 failures/million operations
- \(C_P\) = Multiplying factor which considers the effect of fluid pressure on the base failure rate (See Figure 6.6)
- \(C_Q\) = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 6.7)
- \(C_V\) = Multiplying factor which considers the effect of fluid viscosity/temperature on the base failure rate (See Table 6-6 and Figure 6.15)
- \(C_N\) = Multiplying factor which considers the effect of fluid contaminants on the base failure rate (See Table 6-5)
- \(C_B\) = Multiplying factor which considers the effect of spool clearance on the base failure rate (See Figure 6.12)
- \(C_{DS}\) = Multiplying factor which considers the effect of spool diameter on the base failure rate (See Figure 6.13)
- \(C_\mu\) = Multiplying factor which considers the effect of friction coefficient on the base failure rate (See Table 6-7)
- \(C_W\) = Multiplying factor which considers the effect of flow rate on the base failure rate (See Figure 6.14)

6.4.1 Fluid Pressure

In most sliding action valves the applied fluid pressure is the upstream pressure minus the downstream pressure. Figure 6.6 provides the multiplying factors for fluid pressure. Other factors in evaluating the effects of fluid pressure on valve reliability include the following:

Size - Structural strength becomes an increasingly important consideration with increasing valve size because pressure loads are a function of the square of the valve size.
**Balance** - If the valve is inherently pressure-balanced, the influence of pressure upon such parameters as size and actuation forces will be far less than in the case of an inherently unbalanced unit.

**Pressure Induced Strain** - Binding of certain close-tolerance sliding action valves can result with excessive pressure load on a port.

**Conditions of Pressure** - Circumstances under which the valve unit is subjected to high pressure must be considered. A drain valve, for example, may be required to seal against high pressure, but never be required to open until after the pressure has been relieved.

### 6.4.2 Allowable Leakage

Allowable internal leakage of the sliding action valve can be obtained from valve specifications usually in terms of quiescent flow or leakage flow. Quiescent flow is the internal valve flow or leakage from supply-to-return with no flow in the load ports. Allowable leakage will vary considerably according to the operational requirements. Figure 6.7 provides multiplying factors for allowable leakage.

### 6.4.3 Contamination Sensitivity

Cleanliness of the fluid medium and surrounding medium has a direct effect upon the occurrence of stiction, weldment and general operation of sliding valve assemblies. No fluid system is completely free of particulate contamination and sensitivity of a valve to contamination is an important consideration in reliability.

In sliding action valves there is a tradeoff between contamination sensitivity and leakage based on clearances between the spool and sleeve. If leakage is minimized by reducing the clearance between the valving element and its housing, a larger number of contaminant particles can become lodged, causing valve failure. The clearance values should be checked at both of the temperature extremes to which the valve will be subjected, in order to ensure adequate design for the largest size of contamination particle anticipated.

The analysis of particle sizes includes the determination of upstream filter sizes, the filter maintenance schedule, the number of upstream components between the valve and filter, and the number of particles likely to be encountered at the spool assembly. Table 6-5 provides typical quantities of contaminants for use in the failure rate equation.
6.4.4 **Fluid Viscosity**

Viscosity of a fluid is much more dependent on temperature than it is on pressure. For example, when air pressure is increased from 1 atmosphere to 50, its viscosity is only increased by about 10%. In contrast, the following graph shows the dependence of viscosity on temperature for some common fluids. The graph shows how viscosity of liquids decreases with temperature while that of gases increases with temperature. Multiplying factors for viscosities of typical fluids are provided in Table 6-6. Multiplying factors for other fluids are determined from the table by a knowledge of viscosity at the applicable fluid temperature. Viscosity for a specific fluid is obtainable from many reference sources. If the value located is in terms of kinematic viscosity, divide the value by the specific gravity (density) at the desired temperature to determine the dynamic specific gravity.

6.4.5 **Spool-to-Sleeve Clearance**

Highly polished and uniform surface finishes of 4-6 microinches can usually be assumed for a valve spool. The model assumes that the spool is environmentally protected. If this is not the case, a separate analysis will be required to determine the effects of aging and deterioration of the surfaces on the spool to sleeve clearance. A diametrical spool clearance of 50 microinches is typical for sliding action valves. The exact value is taken from assembly drawings. Figures 6.12 and 6.13 provide multiplying factors for the spool-to-sleeve clearance and spool diameter.

6.4.6 **Friction Coefficient**

A sticking valve spool is usually caused by contaminants. Particles can accumulate between the spool and sleeve as part of the silting process until the build-up is sufficient to cause stiction. Results include valve hunting, erratic regulation and eventual locking. The silting process can be aggravated by inactivity of the valve. Another failure mechanism to be considered is reduced clearance between the spool and sleeve caused by soft metal particles being wedged and burnished on the surfaces. The actual friction coefficient is used in the model.

6.5 **FAILURE RATE ESTIMATE FOR HOUSING ASSEMBLY**

There are many factors which could be considered in determining the potential rate of fatigue failure of a valve housing including connectors. For critical safety related applications, a review of the stress analysis is warranted. Normally, the probability of a cracked housing is minimal and the failure rate is best determined from field experience data.
\[ \lambda_{HO} = \lambda_{HO,B} \]  \hspace{1cm} (6-15)

Where: \( \lambda_{HO} \) = Failure rate of valve housing, failures/million operating hours
\( \lambda_{HO,B} \) = Base failure rate of housing, 0.01 failures/million hours

### 6.6 FAILURE MODE, EFFECTS AND CRITICALITY ANALYSIS

The guidelines in this section should be reviewed if the procedures in this Chapter are being used in the generation of a Failure Mode, Effects and Criticality Analysis (FMECA). See Chapter 25 for specific instructions on generating a FMECA for mechanical equipment. Since the valve will be comprised of seals, springs, fittings and other parts that will be evaluated for reliability, the lowest indenture level for listing individual failure modes needs to be determined. It is recommended that these components be entered on the same worksheet with the valve.

#### 6.6.1 Failure Mode and Effects Analysis

The FMECA is normally performed by first identifying the failure modes and then estimating the probability of occurrence for each identified failure mode. This section provides some guidelines for performing the FMEA.

**Function:** The following guidelines assume that a hardware bottom-up analysis is being performed. The purpose of the FMEA is to identify all potential failure modes of the equipment. It is suggested that the valve itself be identified as a functional description as shown below. This process permits a technical review of the valve function within the equipment and identification of all failure modes applicable to the valve design.

<table>
<thead>
<tr>
<th>Function No.</th>
<th>Function</th>
<th>Mode ID No.</th>
<th>Failure Modes</th>
<th>Cause ID No.</th>
<th>Failure Causes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Functional description of the valve</td>
<td>1</td>
<td>Only one failure mode should be entered for each row because of individual failure effects, ( \alpha ) and ( \beta ) values and compensating provisions</td>
<td>1</td>
<td>Description of all probable independent causes of this specific failure mode</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>2</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td></td>
<td>1</td>
<td>1</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>2</td>
<td>2</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
**Failure Modes:** Section 6.2 and Tables 6-1, 6-2 and 6-3 provide some guidelines for listing the failure modes of valves on the FMEA worksheets. It is important that the identified failure mode be directed toward a specific failure mode at the valve level such as poor response, failure to open, etc. Only one failure mode should be entered for each row because of individual failure effects, $\alpha$ and $\beta$ values and compensating provisions. Some failure modes will not be applicable on the FMEA worksheets. They can be either removed or a 0.0 probability of occurrence assigned.

**Failure Causes:** The different causes of the failure mode occurring should be entered one row at a time. Again, only one failure cause should be entered so that the probability of occurrence of each failure cause can be entered. Section 6.2 provides some guidelines for identifying failure causes. The operating environment needs to be considered so that all possible causes are entered such as contaminants, shock and vibration.

**Failure Effects:** Local failure effects are the consequences of the failure mode on the equipment at the part level. Often this entry will be the same as the failure cause. The effect of the failure mode occurrence at the next higher indenture level is then entered. The end failure effect is the consequence of the failure mode occurrence at the total system level. In some cases the failure effects for different failure causes will be the same as shown in the following table.

**Failure Detection Method:** The method by which occurrence of the failure cause is detected is entered on the worksheet which may be by many different means such as operator, warning device, sensing device or routine diagnostic maintenance.

<table>
<thead>
<tr>
<th>Mission Phase or Operational Mode</th>
<th>Local Failure Effect</th>
<th>Next Higher Failure Effect</th>
<th>End Failure Effect</th>
<th>Failure Detection Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>The mode in which the equipment is operating when the valve fails to operate</td>
<td>Consequences of the failure cause on equipment operation at the part level</td>
<td>Impact of the failure at the next higher indenture level</td>
<td>Total effect of the failure mode on the operation, function or operational status</td>
<td>Method by which occurrence of the failure cause is detected</td>
</tr>
</tbody>
</table>

**Compensating Provisions:** Provisions in the design or operator actions are entered on the worksheet that circumvent or mitigate the effect of the failure such as equipment redundancy, alarm provisions, safety features or alternate modes of operation.
Severity Class: One of the following severity categories is recorded:

Class 1 is catastrophic, a failure which may cause death, total property damage or loss of mission (fire).

Class II is critical, a failure that may cause severe injury or extensive property damage (transmission failure).

Class III is marginal, a failure that may cause minor injury, minor property damage or mission degradation (windshield breakage).

Class IV is minor (a failure not serious enough to cause injury, property damage or mission degradation but may necessitate repairs at a later time (turn signal).

Remarks: Remarks need to be entered that clarify or amplify other worksheet entries such as design features, safety provisions combination of entries, etc.

<table>
<thead>
<tr>
<th>Compensating Provisions</th>
<th>Severity Class</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Provisions in the equipment design which alter the effect of failure mode occurrence</td>
<td>Severity Class of I, II, III, or IV</td>
<td>Remarks that clarify or amplify worksheet entries</td>
</tr>
</tbody>
</table>

6.6.2 Criticality Analysis

A Criticality Analysis (CA) is performed following the FMEA to determine the probability of occurrence that an identified failure mode will actually result in the defined end effect. In most cases the CA worksheet will repeat the columns that identify the component function, failure modes, failure causes, and ID numbers so that the entries can be traced from one worksheet to another. This section provides some guidelines for determining failure rates and occurrence probabilities.

Failure Rate Data Source: The Chapter of the Handbook can be entered or other source of the part failure rate is entered such as RAC, OREDA, or NPRD.

Base Failure Rate: The failure rate of the part as provided by the data source is entered on the worksheet.
**Adjustment (π) factor:** If the failure rate is obtained from a source where the operational environment is different from the intended operating environment of the equipment being analyzed, an adjustment factor may need to be applied. Some valves are designed to open and close continuously or the valve may be a shut-off valve. The valve may also be a sliding action regulator valve under continuous operation. If this Handbook is used the duty cycle is entered into the calculation. For other failure rate sources a duty cycle may need to be considered on determining an adjustment factor.

**Part Failure Rate:** This entry is simply the base failure rate multiplied by the π factor.

**Failure Effect Probability:** The failure mode may directly result in the listed end effect or it may not always result in the listed end effect. It is an engineering judgment as to the probability of occurrence after reviewing the failure causes and the operating environment of the equipment.

**Failure Mode Ratio:** The part failure rate will be the total failure rate of the valve under normal operating conditions. This failure rate needs to be subdivided per the particular valve failure rate. The total of all the α values for a particular valve assembly will be equal to 1.0.

<table>
<thead>
<tr>
<th>Failure Rate Data Source</th>
<th>Base Failure Rate</th>
<th>Adjustment Factors (π)</th>
<th>Part Failure Rate</th>
<th>Failure Effect Probability (β)</th>
<th>Failure Mode Ratio (α)</th>
</tr>
</thead>
<tbody>
<tr>
<td>OREDA, RAC, NPRD, NSWC HDBK, etc.</td>
<td>Failure rate from source</td>
<td>Correction factors as required to convert acquired failure rate to application failure rate</td>
<td>Base failure rate multiplied by π factor</td>
<td>Conditional probability that identified failure mode will result in end effect (0.0 to 1.0)</td>
<td>Ratio of part failure rate related to identified failure mode</td>
</tr>
</tbody>
</table>

**Failure Mode Criticality Number:** This number is a combination of the part failure rate, failure effect probability and failure mode ratio. This number establishes a reference that can be used to compare the occurrence probability of this particular failure mode with other failure mode probabilities.

**Part Criticality Number:** A criticality number for the part is sometimes valuable for evaluating the criticality of the particular valve being analyzed in relation to other components of the system. The part criticality number is simply the summation of all the failure mode criticality numbers for the valve. The part criticality number can be used to compare the probability of part failure with other component failure probabilities.
**Severity Class:** The severity class from the FMEA worksheet is normally repeated for informational purposes.

**Remarks:** Remarks are included on the worksheet to clarify or amplify other entries on the worksheet such as explanations of failure rate sources, modification numbers, and calculations.

<table>
<thead>
<tr>
<th>Failure Mode Criticality Number, $C_N$ ($\lambda p \beta x \alpha$)</th>
<th>Part Criticality Number $\sum C_N$</th>
<th>Severity Class</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Criticality number establishes reference for comparison purposes</td>
<td>Criticality number establishes reference for comparison purposes</td>
<td>The severity of the failure mode copied from the FMEA worksheet</td>
<td>Remarks to clarify or amplify other worksheet entries such as failure rate sources and adjustment factors</td>
</tr>
</tbody>
</table>
For $P_1 - P_2 > 50$ psi:

$$C_P = \left( \frac{P_1 - P_2}{3000} \right)^2$$

**Figure 6.6 Fluid Pressure Multiplying Factor**
For leakage \( > 0.03 \text{ in}^3/\text{min} \), \( C_\varnothing = \frac{0.055}{Q_f} \)

For leakage \( \leq 0.03 \text{ in}^3/\text{min} \), \( C_\varnothing = 4.2 - (79Q_f) \)

**Figure 6.7 Allowable Leakage Multiplying Factor**
Table 6-4 provides typical surface finishes

\[ C_F = \frac{f^{1.65}}{353} \]

**Figure 6.8 Surface Finish Multiplying Factor**
$C_S = 0.26 \left( \frac{9000}{S_S} \right)^{1.5}$

where: $S_S = \text{poppet contact pressure, lbs/in}^2$

**Figure 6.9 Contact Pressure Multiplying Factor**
$C_{DT} = 1.1D_S + 0.32$

Where $D_S = $ Seat Diameter, inches

**Figure 6.10  Seat Diameter Multiplying Factor**
For $L_w \leq 0.34$, \[ C_{SW} = 3.55 - 24.52L_w + 72.99L_w^2 - 85.75L_w^3 \]

For $L_w > 0.34$, \[ C_{SW} = 0.25 \]
For $B < 500 \mu \text{in}$, $C_B = 0.42$

For $B > 500 \mu \text{in}$, $C_B = \frac{B^2}{6 \times 10^5}$

**Figure 6.12 Spool Clearance Multiplying Factor**
$C_{DS} = 0.615 \, D_{SP}$

Figure 6.13  Spool Diameter Multiplying Factor
Figure 6.14 Flow Rate Multiplying Factor

\[ C_W = 1.0 + F_L^2 \]
See Table 6-6 for values of typical fluid viscosities

Figure 6.15 Fluid Viscosity Multiplying Factor
### Table 6-4. Typical Surface Finishes for Manufacturing Processes

<table>
<thead>
<tr>
<th>PROCESS</th>
<th>SURFACE FINISH, ( \mu \text{in} )</th>
<th>PROCESS</th>
<th>SURFACE FINISH, ( \mu \text{in} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lapping</td>
<td>2 - 16</td>
<td>Boring, turning</td>
<td>16 - 200</td>
</tr>
<tr>
<td>Polishing</td>
<td>4 - 16</td>
<td>Electron beam</td>
<td>32 - 250</td>
</tr>
<tr>
<td>Honing</td>
<td>4 - 32</td>
<td>Reaming</td>
<td>32 - 125</td>
</tr>
<tr>
<td>Grinding</td>
<td>4 - 64</td>
<td>Milling</td>
<td>32 - 250</td>
</tr>
<tr>
<td>Burnishing</td>
<td>8 - 16</td>
<td>Drilling</td>
<td>64 - 250</td>
</tr>
</tbody>
</table>

### Table 6-5. Contaminant Multiplying Factor, \( C_N \)

<table>
<thead>
<tr>
<th>HYDRAULIC COMPONENT PRODUCING PARTICLES</th>
<th>PARTICLE MATERIAL</th>
<th>NUMBER PARTICLES UNDER 10 MICRON PER HOUR ( (N_{10}) )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>PER GPM</td>
</tr>
<tr>
<td>Piston Pump</td>
<td>steel</td>
<td>0.017</td>
</tr>
<tr>
<td>Gear Pump</td>
<td>steel</td>
<td>0.019</td>
</tr>
<tr>
<td>Vane Pump</td>
<td>steel</td>
<td>0.006</td>
</tr>
<tr>
<td>Cylinder</td>
<td>steel</td>
<td>0.008</td>
</tr>
<tr>
<td>Sliding action valve</td>
<td>steel</td>
<td>0.0004</td>
</tr>
<tr>
<td>Hose</td>
<td>rubber</td>
<td>0.0013</td>
</tr>
</tbody>
</table>

\[
C_N = \left( \frac{C_o}{C_{10}} \right)^3 N_{10} GPM_R \quad \text{or} \quad C_N = \left( \frac{C_o}{C_{10}} \right)^3 N_{10} LPM_R
\]

Where:
- \( C_o \) = System filter size in microns
- \( C_{10} \) = Standard system filter size = 10 micron
- \( GPM_R \) = Rated flow in gallons/min
- \( LPM_R \) = Rated flow in liters/min
- \( N_{10} \) = Particles/hour/rated GPM or particles/hour/rated LPM for gas valve applications
Table 6-6. Fluid Viscosity/Temperature Multiplying Factor, $C_V$
for Typical Fluids

<table>
<thead>
<tr>
<th>FLUID</th>
<th>$C_V$</th>
<th>Fluid Temperature, °F</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>-50</td>
<td>0</td>
</tr>
<tr>
<td>Air</td>
<td>554.0</td>
<td>503.4</td>
</tr>
<tr>
<td>Oxygen</td>
<td>504.6</td>
<td>457.8</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>580.0</td>
<td>528.0</td>
</tr>
<tr>
<td>Carbon Dioxide</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Water</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>SAE 10 Oil</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>SAE 20 Oil</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>SAE 30 Oil</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>SAE 40 Oil</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>SAE 50 Oil</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>SAE 90 Oil</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Diesel Fuel</td>
<td>0.1617</td>
<td>0.7492</td>
</tr>
<tr>
<td>MIL-H-83282</td>
<td>0.0031</td>
<td>0.0432</td>
</tr>
<tr>
<td>MIL-H-5606</td>
<td>0.0188</td>
<td>0.0951</td>
</tr>
</tbody>
</table>

--- Data for these temperatures determined to be unreliable

\[ C_V = \left( \frac{\nu_o}{\nu} \right) \]

Where: \( \nu_o = 2 \times 10^{-8} \) lbf-min/in²
\( \nu = \) Dynamic viscosity of fluid being used, lbf-min/in²
Table 6-7. Friction Coefficient of Typical Materials used in Valve Designs

<table>
<thead>
<tr>
<th>Material</th>
<th>Friction Coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$C_μ$ Dry</td>
</tr>
<tr>
<td>Steel on steel</td>
<td>0.8</td>
</tr>
<tr>
<td>Aluminum on steel</td>
<td>0.6</td>
</tr>
<tr>
<td>Copper on steel</td>
<td>0.5</td>
</tr>
<tr>
<td>Brass on steel</td>
<td>0.5</td>
</tr>
<tr>
<td>Cast iron on steel</td>
<td>0.4</td>
</tr>
<tr>
<td>Brass on nylon</td>
<td>0.3</td>
</tr>
<tr>
<td>Steel on nylon</td>
<td>0.3</td>
</tr>
<tr>
<td>Teflon on Teflon</td>
<td>0.05</td>
</tr>
<tr>
<td>Hard carbon on carbon</td>
<td>0.2</td>
</tr>
<tr>
<td>Copper on copper</td>
<td>1.3</td>
</tr>
<tr>
<td>Aluminum on aluminum</td>
<td>1.1</td>
</tr>
<tr>
<td>Nickel on nickel</td>
<td>0.7</td>
</tr>
<tr>
<td>Brass on brass</td>
<td>0.9</td>
</tr>
</tbody>
</table>
6.7 REFERENCES


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7.1 INTRODUCTION

Bearings are among the few components that are designed for a finite life because of the fatigue properties of the materials used. Most bearings can be assigned a $L_{10}$ life, which is the number of hours at a given load that 90 percent of a set of apparently identical bearings will complete or exceed before failure. There are a number of other factors that can be applied to the $L_{10}$ life so that it more accurately correlates with the intended operating environment. These factors include actual lubrication film thickness, misalignment, velocity, load stresses and subjection to contaminants.

There are many different types of bearings in use making it extremely difficult to establish base failure rates for bearings based on field performance data. Bearing analysis is also extremely difficult due to the large number of engineering parameters related to bearing design. The most common failure mode of a bearing is wear. The
fundamental problem is that the bearing surfaces are neither perfectly flat nor smooth; and when two surfaces such as a ball and raceway come into contact, only a small percentage of the apparent surface area is actually supporting the load. The result is high contact stresses, which can lead to excessive friction and wear. The procedures for estimating bearing reliability presented in this chapter utilize the manufacturer's published $L_{10}$ life with multiplying factors to relate the $L_{10}$ value to intended operating conditions.

7.2 BEARING TYPES

7.2.1 Rotary Motion Bearings

The rotary motion bearing is used in those applications in which the main load is transferred through elements in rolling contact. These bearings are manufactured to take pure thrust loads, pure radial loads, or a combination of the two loads. Rolling contact is provided by a rolling element, ball or roller to carry a load with minimal wear and friction. Because of the greatly reduced starting friction when compared to the conventional journal bearing, rotary motion bearings have acquired the common designation of "anti-friction" bearings. The most common rotary motion bearing application is that of a ball bearing used to support a shaft with radial and thrust loads in rotating equipment. Load, speed, and the operating viscosity of the lubricant affect the frictional characteristics of a rotary motion bearing.

Rolling element bearings have a life which is limited by the fatigue life of the material from which they are made and as modified by the lubricant used. In rolling contact fatigue, precise relationships between life, load, and design characteristics are difficult to predict and, therefore, the statistical $L_{10}$ life based on a probability of survival is used with multiplying factors to adjust the $L_{10}$ life to the actual conditions being projected.

7.2.1.1 Ball Bearings

Ball bearings are generally used where there is likely to be excessive misalignment or shaft deflection. Most ball bearing designs originate from three basic types:

1. Single-row radial - the most widely used ball bearing, a symmetrical unit capable of absorbing combined radial and thrust loads. It is not intended for pure thrust loads. Because this type of ball bearing is not self-aligning, accurate alignment between the shaft and housing bore is required.

2. Single-row angular contact - designed for combined radial and thrust loads where the thrust component may be large and axial deflection must be confined. A high shoulder on one side of the outer ring is provided to take the thrust, and the shoulder on the other side is sufficiently high to make the bearing non-separable.
(3) Double-row angular contact - two single-row angular contact bearings built as a unit with the internal fit between balls and raceway fixed during assembly. These ball bearings have a known amount of internal preload built in for maximum resistance to deflection. They are very effective for radial loads where bearing deflection must be minimized.

7.2.1.2 Roller Bearings

Cylindrical roller bearings are used to support pure radial loads. They are often used at one end of a highly loaded gear shaft with either tapered roller bearings or multiple-row matched ball bearings at the other end. Roller bearing life is drastically reduced by excessive misalignment or deflection; hence, when using roller bearings, the stack-up of tolerances contributing to misalignment and the shaft or housing deflections should be carefully considered. To compensate for some degree of misalignment or deflection and to carry heavy radial loads, roller bearings are crowned to prevent the phenomenon known as end loading. End loading invariably leads to a drastic reduction in bearing life. The crowning process distributes the load away from the roller ends and prevents excessive stress that could cause fatigue at the roller bearing ends.

Tapered roller bearings are being used increasingly in modern drive systems, since they can react to both thrust and radial loads and can offer the greatest load-carrying capacity in the smallest possible envelope. Although early tapered roller bearings were speed limited, these restrictions have been removed by utilizing bearings with special lubrication features. However, on very high-speed shafts, the use of tapered roller bearings may be precluded due to their inability to operate for required time intervals under survivability (oil-off) conditions. Tapered roller bearings, unlike single-row ball and cylindrical roller bearings require spacers or shims to give these bearings the proper amount of preload or end play for proper operation. Usually it is desirable to have a light preload although a small amount of end play is often acceptable. As with internal clearance, extremes in end play or preload should be avoided.

Needle bearings are characterized by their relatively small size rollers, usually not ranging above 1/4 inch in diameter, and a relatively high ratio of length to diameter, usually ranging about 8 to 1. Another common characteristic of needle bearings is the absence of a cage or separator for retaining the individual rollers.

7.2.2 Linear Motion Bearings

Linear motion bearings provide sliding contact between mating surfaces. The more common types of sliding contact bearings include:

(1) Radial bearings designed to support rotating shafts or journals
(2) Thrust bearings designed to support axial loads on rotating members
Guide bearings designed to guide moving parts in a straight line

The relative motion between the parts of linear motion bearings may take place as a sliding contact without the benefit of a lubricating medium such as with the dry operation of Teflon. Sliding action may also occur with hydrodynamic lubrication in which a film build-up of lubrication medium is produced, with either whole or partial separation of the bearing surfaces. Hydrostatic lubrication may also be used in which a lubricating medium is introduced under pressure between the mating surfaces causing a force opposite to the applied load.

Although linear motion bearings are relatively inexpensive, they can cause costly equipment shutdowns if not properly integrated into the design. Short bearing life can be caused by misalignment, a bent shaft, a rotating part rubbing on a stationary part, a rotor out of balance causing vibration, excessive thrust caused by mechanical failure of other parts, excessive temperature caused by lack of lubrication, dirt or other contaminant and corrosion from water in the bearing housing.

The reliability analysis procedures in this chapter focus mainly on rotary motion bearings. Linear motion and sliding bearings are covered in greater detail in Chapter 18.

7.3 DESIGN CONSIDERATIONS

7.3.1 Internal Clearance

Internal clearance, the clearance between the inner race and the shaft, is an important consideration in the design of ball and roller bearings, since improper internal clearance can drastically shorten the life of a bearing. A small internal clearance may limit the amount of misalignment that can be tolerated and can lead to heavily preloaded bearings. Excessive internal clearance will cause the load to be carried by too few rolling elements. The best practice is to ensure that under all conditions there will be a small positive internal clearance. Usually, the most significant factors to consider when determining mounted internal clearance of the bearing are the reduction of internal clearance due to shaft or housing fits and the effect of temperature on the housing/outer race interface diameters.

7.3.2 Bearing Race Creep

The creeping or spinning of bearing inner races on gear shafts is a fairly common, although not usually serious, problem in most drive systems. Lundberg and Palmgren developed fairly simple parametric calculations for the minimum fit to prevent creep with solid shafts, but there has been little if anything published on minimum press fits for hollow shafts, as are used in helicopter drive systems. Since an accurate mathematical solution to such a problem would be extremely difficult, the best approach seems to be
a reliance on past experience. Sometimes it may not be possible to achieve the necessary press fit to prevent creep without introducing excessively high hoop stress in the bearing race. A common practice in this case is to use separate anti-rotation devices with a slotted bearing race. Although this practice is fairly effective with stationary races, it is seldom effective with rotating races.

7.3.3 Bearing Material

Because the wear rate of a material is proportional to the load applied to it, and inversely proportional to its hardness, one obvious way of reducing wear on bearing components is to increase the hardness at their surface. This is commonly accomplished by using hard coatings, such as electro-less nickel, hard anodised aluminum and thin dense chrome. In addition, other hard coatings, such as titanium carbide, carburising, and both carbo- and plasma nitriding are also widely used. Another advance in bearing technology has been the development of extremely clean bearing steels resulting from vacuum-melt processing. Vacuum-melt bearings have significantly increased the potential life of a bearing by one and one-half to two times the life of vacuum-degassed bearings. Bearings of such advanced materials as M-50 steel can offer even further improvement. Cost of the bearing is an important consideration and the application of the bearing considering such factors as loading and velocity must determine the bearing selection.

7.3.4 Inspection Requirements

Design analysis must include the consideration of proper inspection procedures for the assembly of bearings which can enhance their reliability. Besides the obvious dimensional inspection requirements, two additional inspections by the manufacturer should be specified for all high performance drive system bearings:

- Magnetic particle
- Nital etch

Magnetic particle inspection can detect the presence of relatively large surface or near-surface anomalies, such as inclusions, which are often the cause of bearing spalls. Nital etch inspection can detect the presence of grinding burns, which locally change the hardness of the material and cause premature bearing failure.

7.3.5 Bearing Installation and Removal

The installation of bearings should be carefully considered during design not only to prevent assembly errors, but also to permit easy removal of the bearing without damaging it. Lead chamfers are often installed at bearing journals to facilitate installation. When specifying the breakout on the bearing corners, the shaft drawing should be checked to ensure that the maximum radius at the shaft shoulder will be cleared by the bearing. The height of the shaft shoulder should, if possible, be
consistent with that recommended by bearing manufacturers. Where necessary, flats should be machined on the shaft shoulder so that a bearing puller can remove the bearing by contacting the inner race. Many bearings have been damaged in the past where the bearing puller could grab only the cage or rollers of the bearing. Where duplex bearings are used, the bearings should be marked so that the installer can readily determine the proper way for the bearings to be installed. Incorrectly installed duplex bearings will not properly react to the design loads. All bearings that can be separated should have the serial number clearly shown on all of the separable components. This will prevent the inadvertent mixing of components. Every assembly drawing that contains bearings should clearly explain in the drawing notes how the bearing should be installed. It is imperative that the mechanics building up this assembly have this information available.

7.4 BEARING FAILURE MODES

The common bearing failure modes, mechanisms and causes are listed in Table 7-1. One common mechanism of bearing failure is spalling, which is defined as subsurface chipping or breaking. The failure is usually caused by loading of the bearing exceeding the design load. Surface fatigue or peeling is a cracking and peeling of the surface metal. It is usually the result of poor lubrication or surface damage which interrupts the lubricant film. Scores and scratches are usually caused by hard particles being trapped in a bearing. This failure mechanism may also be caused by inadequate sealing, contaminants in the lubricant, or installation damage.

Smearing is surface damage resulting from unlubricated sliding contact within a bearing. Brinelling is the actual indentation of a rolling element under excessive load or impact that causes stresses beyond the yield point of the bearing material. Fretting wear is usually caused by an improper fit between the bearing and the shaft or outer surface of the bearing. This allows movement of the race in relation to the housing or shaft. The surfaces then wear or score, thereby damaging the surfaces and preventing a firm, fixed contact.

Roller and tapered bearings have an additional failure mode defined as scuffing of the bearing surfaces. This failure mode is usually caused by bearing exposure to an excessive load for an extensive period of time. The surfaces of the moving parts are scored or scratched, increasing the roughness of the surfaces, setting up stress concentrations and increasing friction. The scoring also interferes with the normal lubricant film and increases the metal-to-metal contact during use.

Fatigue can occur due to cyclic loads normal to the bearing surface. Wiping occurs from surface to surface contact due to loss of sufficient lubrication film thickness. This malfunction can occur from under-rotation or from system fluid losses. Overheating is indicated by babbitt cracking or surface discoloration. Corrosion is frequently caused by the chemical reaction between the acids in the lubricants and the base metals in the babbitt. Lead based babbitts tend to show a higher rate of corrosion failures.
Table 7-1. Typical Modes of Bearing Failure

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE MECHANISM</th>
<th>FAILURE CAUSE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fatigue damage</td>
<td>- Spalling of ball/roller raceway</td>
<td>- Heavy, prolonged load</td>
</tr>
<tr>
<td></td>
<td>- Brinelling</td>
<td>- Excessive speed</td>
</tr>
<tr>
<td></td>
<td>- Smearing</td>
<td>- Shock load</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Excessive vibration</td>
</tr>
<tr>
<td>Noisy bearing</td>
<td>- Surface fatigue</td>
<td>- Loss of lubricant</td>
</tr>
<tr>
<td></td>
<td>- Glazing</td>
<td>- Housing bore out of round</td>
</tr>
<tr>
<td></td>
<td>- Microspalling of stressed surfaces</td>
<td>- Corrosive agents</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Distorted bearing seals</td>
</tr>
<tr>
<td>Bearing seizure</td>
<td>- Crack formation on rings and balls or rollers</td>
<td>- Inadequate heat removal capability</td>
</tr>
<tr>
<td></td>
<td>- Skidding</td>
<td>- Loss of lubricant</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- High temperature</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Excessive speed</td>
</tr>
<tr>
<td>Bearing vibration</td>
<td>- Scuffing</td>
<td>- Misalignment</td>
</tr>
<tr>
<td></td>
<td>- Fretting</td>
<td>- Housing bore out of round</td>
</tr>
<tr>
<td></td>
<td>- Pitting of surfaces</td>
<td>- Unbalanced/excessive load</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Inadequate housing support</td>
</tr>
</tbody>
</table>

Severe performance requirements may affect the reliability of the bearings if there is a path of heat conduction from the machine or any friction creating components within it to the bearings (for example, brakes or clutches). This condition may cause a decrease in the bearing lubricant's operating viscosity and, consequently, a reduction in bearing life. A lubricant with a higher temperature rating should prevent leakage or excessive wear.

7.5 BEARING FAILURE RATE PREDICTION

Rolling element bearing life is usually calculated using the Lundberg-Palmgren method (Reference 53). This method is a statistical technique based on the subsurface initiation of fatigue cracks through hardened air-melt bearing material. Most mechanical systems are not utilized precisely as the bearing manufacturer envisioned; therefore, some adjustment factors must be used to approximate the failure rate of the bearings under specific conditions.
Experience has shown that the service life of a bearing is usually limited by either excessive wear or fatigue. Excessive wear occurs when the bearings are improperly installed or exposed to hostile operating environments. Inadequate lubrication, misalignment, contamination, shock, vibration, or extreme temperature all cause bearings to wear out prior to their estimated design life. In contrast, a bearing can be expected to perform adequately for the duration of its rated life, given proper operating conditions, until failure occurs due to fatigue.

Rolling element bearings ultimately fail due to fatigue because the load carrying balls, raceways, rollers, etc. are subjected to cyclical contact stresses. Under laboratory conditions the fatigue characteristics of bearings can be quantified in terms of stress magnitude and number of stress cycles, which in turn relates to the bearing load and number of revolutions. A heavily loaded bearing, for example, has a much shorter fatigue life than a lightly loaded one when both are operated at the same low speed. Conversely, a bearing operated under a light load and low speed provides a service life several times greater than the rated life. In this latter case service will generally be terminated by wear.

Attempting to estimate the fatigue life of an individual bearing is not very practical because of the large number of design parameters to consider in relation to the sensitivity of the operating environment. Instead, statistical methods are used to rate bearings based on the results of large groups of the same type of bearing tested to failure under controlled laboratory conditions to establish a fatigue life rating. This rating, known as the $L_{10}$ life, is defined as the number of hours that 90% of the bearings operating at their rated load and speed, can be expected to complete or exceed before exhibiting the first evidence of fatigue. It is important to consider the bearing application before using the published $L_{10}$ life as a reliability estimate. For example, a bearing in a direct drive motor application may have a predicted life of 400,000 hours but the same bearing in a belt drive or pillow block application may have a life of 40,000 hours depending on loading.

Standard equations have been developed to extend the $L_{10}$ rating to determine the statistical rated life for any given set of conditions. These equations are based on an exponential relationship of load to life.

\[
L_{10} = \left( \frac{L_S}{L_A} \right)^{y} \tag{7-1}
\]

where:

- $L_{10} = \text{Bearing life with reliability of 90\%, millions of revolutions}$
- $L_S = \text{Basic dynamic load rating, lbf}$
- $L_A = \text{Equivalent radial load, lbf}$
\( y = \) Constant, 3.0 for ball bearings, 3.3 for roller bearings

The basic dynamic load rating is the dynamic load capacity of the bearing that is established during \( L_{10} \) life testing and can be found in manufacturer’s catalogs. The equivalent radial load is the load the bearing will see in service and can be found in engineering drawings or calculated. Normally \( L_A \) will be approximately 0.5 \( L_S \) depending on the anticipated environmental and maintenance considerations of the design and can be used as a value for preliminary reliability estimates.

The \( L_{10} \) life can be converted to hours with the following:

\[
L_{10h} = 10^6 / (60n) \cdot L_{10}
\]

(7-2)

where: \( L_{10h} \) = Bearing life (at 90% reliability), operating hours \( n \) = Rotational speed, revolutions/ min

In a ball or roller bearing, the rolling elements transmit the external load from one ring to the other. The external force load is generally composed of a radial load \( F_R \) and an axial load \( F_A \) and is distributed over a number of rolling elements. These two components combine to form the equivalent radial load. The equivalent radial load, \( L_A \), is defined as the radial load producing the same theoretical fatigue life as the combined radial and thrust loads. All bearing loads are converted to an equivalent radial load. If only pure radial loads are involved, then the value for \( L_A \) is simply the radial load.

Except for the special case of pure thrust bearings, bearing ratings shown in manufacturers' catalogs are for radial loads. When thrust is present, an equivalent radial load must be determined before estimating reliability. Most bearing manufacturers provide methods of combining thrust and radial loads in accordance with ANSI standards to obtain an equivalent radial load. This relationship can be written as follows:

\[
L_A = XF_R + YF_A
\]

(7-3)

Where: \( L_A \) = Equivalent radial load, lbf \( F_R \) = Radial load, lbf \( F_A \) = Axial load, lbf \( X \) = Radial factor relating to contact angle
\[ Y = \text{Thrust factor relating to contact angle, thrust load and the number and size of balls or rollers in the bearing} \]

A bearing catalog will display separate tables of values to cover single-row, double-row, and angular-contact variations. \( X \) and \( Y \) can be obtained from the manufacturer of the bearing. References 44 and 83 provide design equations to calculate radial and thrust loads, and guidelines for estimating the radial and thrust factors. \( F_A \) should not exceed 30% of the radial load.

Substantial improvements in materials processing and manufacturing techniques have been made since the original development of the \( L_{10} \) concept for predicting bearing life. For instance, high-purity steels that are vacuum degassed or vacuum melted are now widely used for bearings. Also, bearing components are manufactured to tighter tolerances on geometry, and ball/raceways have finer finishes, which help to improve lubricating films. For reasons such as these, bearing manufacturers have modified their \( L_{10} \) ratings with certain adjustment factors.

To adjust the actual failure rate of the bearing from the calculated \( L_{10} \) life, it can be assumed that the failure rate will correspond to 63.21% of the failure probability. Thus a reliability factor of 5.0 can be established to convert the \( L_{10} \) value to \( L_{50} \). The following equation can then be established for the failure rate of a bearing:

\[
\lambda_{BE} = \lambda_{BE,B} \cdot C_V \cdot C_{CW} \cdot C_t \cdot C_{SF}
\]  \hspace{1cm} (7-4)

Where:
- \( \lambda_{BE} \) = Failure rate of bearing, failures/million revolutions
- \( \lambda_{BE,B} = 2 \times 10^5 \) / \( L_{10} \) failures/million revolutions
- \( C_V \) = Multiplying factor for lubricant (See Figure 7.1)
- \( C_{CW} \) = Multiplying factor for water contaminant level (See Section 7.5.2 and Figure 7.2)
- \( C_t \) = Multiplying Factor for operating temperature (See Figure 7.3)
- \( C_{SF} \) = Multiplying factor for operating service conditions (See Table 7-2)

### 7.5.1 Lubricant Multiplying Factor

The lubricant factor, \( C_V \), is a function of the viscosity of the lubricant used in the bearing system at the intended operating temperature. \( C_V \) can be expressed as:
\[ C_v = \left( \frac{v_O}{v_L} \right)^{0.54} \]  \hspace{1cm} (7-5)

Where:  
\( v_O \) = Viscosity of specification lubricant, lb-min/in\(^2\)  
\( v_L \) = Viscosity of lubricant used, lb-min/in\(^2\)

Multiplying factors for the effect of lubrication viscosity on the failure rate of a bearing are shown in Figure 7.1.

### 7.5.2 Lubricant Contamination Multiplying Factor

Less than 10 percent of all ball bearings last long enough to fail due to normal fatigue (Reference 8). Most bearings will fail due to static overload, wear, corrosion, lubricant failure, contamination, or overheating. Water contamination, for example, can have a detrimental effect on fatigue life. A water contamination multiplying factor which accounts for the reduction in fatigue life due to the leakage of water into the oil lubrication is shown in Figure 7.2. This factor is represented as \( C_{CW} \) and is represented by the following equations derived from data in Reference 19.

\[ C_{CW} = 1.04 + 1.03 CW - 0.065 CW^2 \]  \hspace{1cm} (7-6)

Where:  
\( CW \) = Percentage of water in the lubricant

The \( C_{CW} \) multiplying factor will modify the base failure rate as shown in Equation (7-4). For bearings designed for water based lubricants \( CW = 0 \) and \( C_{CW} = 1.04 \)

### 7.5.3 Service Factor

The actual radial or axial load on the bearing may be greater than the calculated load because of vibration and shock present during operation of the equipment. A service factor can be used to adjust the failure rate for various operating conditions as shown in Table 7-2.
Table 7-2. Bearing Service Factors
(Reference 112)

<table>
<thead>
<tr>
<th>Operating Conditions</th>
<th>Typical Applications</th>
<th>Service Factor, $C_{SF}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth operation free from shock</td>
<td>Electric motors, machine tools, air conditioners</td>
<td>1.0 to 1.2</td>
</tr>
<tr>
<td>Normal operation</td>
<td>Air blowers, compressors, elevators, motor vehicles</td>
<td>1.2 to 1.5</td>
</tr>
<tr>
<td>Operation with shock and vibration</td>
<td>Construction, off-road, shipboard equipment</td>
<td>1.5 to 3.0</td>
</tr>
</tbody>
</table>
\[ C_v = \left( \frac{\nu_O}{\nu_L} \right)^{0.54} \]

Where:

- \( \nu_O \) = Viscosity of specification fluid
- \( \nu_L \) = Viscosity of lubricant used

**Figure 7.1 Multiplying Factor for Bearing Lubricant**
\[ C_{CW} = 1.04 + 1.03 CW - 0.065 CW^2 \]

Where: \( CW = \) Percentage of water in the lubricant

**Figure 7.2 Water Contamination Multiplying Factor**
Temperature Multiplying Factor, $C_t$

$C_t = 1.0$ for $T_O < 183^\circ C$

$C_t = \left( \frac{T_O}{183} \right)^3$ for $T_O \geq 183^\circ C$

Where: $T_O = \text{Operating Temperature of the Bearing}$

**Figure 7.3 Operating Temperature Multiplying Factor**
7.6 REFERENCES


CHAPTER 8
GEARS AND SPLINES

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8.1 INTRODUCTION

The reliability of a gear and other gearbox components is an extremely important consideration in the design of a power-transmission system, ensuring that the required loads can be handled over the intended life of the system. Some general design constraints and requirements need to be given special attention because of their potential impact on the long-term reliability of the total system. One is the operating power spectrum and determining the potential requirements for growth. Another is that changing requirements may cause a configuration change where a misalignment could cause vibration that could set up stresses and lead to fatigue failure. Another example is the lubrication system if included as part of the gearbox design, assuring that the
capacity, filter and transferring components are adequate. If superfine filters are required, sufficiently larger traps are needed to accommodate the increase in particles trapped in the element. The lubricant flow should be designed so that the particles within the system are removed prior to reentry into the gearbox area.

Noise and vibration can affect reliability, not only of the gearbox itself, but also of associated components within the complete power-transmission system. Hence, every effort should be made to configure a gearbox that is as quiet and as vibration-free as possible. The design analysis should also include the assurance that critical speeds and gear clash resonance frequencies, which may reinforce each other, are avoided.

In most gearbox applications, especially in airborne systems, weight is usually a constraining and, in some cases, the controlling factor. In general, overdesign means higher reliability, but in weight critical systems, overdesign in one area requires underdesign elsewhere; thereby, defeating the purpose of the overdesign. For example, bearing life should never be sacrificed in the design because bearings are likely to be the main drivers in determining gear system reliability.

When a gearbox is exposed to overstress, several conditions occur that greatly affect the failure rate. Bolted gear flanges will be subject to fretting and high loads will cause bevel gears to shift patterns, making tooth breakage a likely occurrence. Spur gears develop scuffing lines increasing the roughness of the surface as loads are increased. A gear system should be used in a design that exceeds the specification load only after detailed analysis of the impact on each part or component has been made.

8.2 TYPES OF GEARS

8.2.1 Spur and Helical Gears

Spur gears are cylindrical in form and operate on parallel axes with the teeth straight and parallel to the axes. Spur gears are commonly used in all types of gearing situations, both for parallel-axis speed reduction and in coaxial planetary designs. A typical spur gear arrangement is shown in Figure 8.1. In general, the reliability of drive train spur gears is extremely high due to present design standards. There are, however, some considerations that should be addressed because they are frequently overlooked in spur gear design or selection for specific purposes.

Generally, the initial design of a spur gear mesh is one of standard proportions and equal tooth thickness for both pinion and gear. This is, however, rarely the optimum configuration for a spur gear mesh, because this type of design does not have two very desirable characteristics: recess action and a balanced bending stress in pinion and gear. A recess-action gear mesh, shown in Figure 8.2, has a long addendum pinion and short addendum gear. A recess-action mesh is quieter and smoother running than
standard mesh and has a much lower tendency to score due to better lubrication within the mesh.

Figure 8.1 Typical Spur Gear Arrangement

Figure 8.2 Gear Mesh Arrangements

Although the advantage of having balanced bending stresses on a pinion and gear is primarily lower weight, it does have an indirect effect on reliability. As stated earlier, whenever there is an inefficient use of weight, reliability is compromised somewhat. For example, even a fraction of a pound saved in the optimization of a spur gear mesh could be applied to a bearing where the life could perhaps be doubled. While
overemphasis of weight reduction can be detrimental to reliability, the carrying of excess weight can have a far-reaching effect; therefore, a balanced gear system must be the goal for efficient and reliable systems. Fortunately, it is usually a simple task to achieve recess-action and balanced bending stress in most spur gear designs. This is accomplished by experimentally shifting the length of contact up the line of action toward the driver gear, while increasing the circular tooth thickness of the pinion and decreasing that of the gear.

There are four design criteria that are used to evaluate the adequacy of spur or helical design: bending stress, hertz or vibrational stress, flash temperature index and lubrication film thickness. The first three have long been used in gear design and methods of calculation are well documented in many publications. It is obvious that if an oil film of a greater thickness than the contact surface asperities can be maintained scoring will not occur, since a metal-to-metal contact will not be experienced.

An important parameter to evaluate lubrication effectiveness is the lubricant film thickness. The equation below is a non-dimensional expression for lubricant film thickness:

\[
H_L = \frac{2.65 G^{0.54} U^{0.7}}{W^{0.13}} \tag{8-1}
\]

Where:
- \( H_L \) = Dimensionless film thickness factor
- \( G \) = Viscosity and material parameter
- \( U \) = Speed parameter
- \( W \) = Load parameter

Since it is often difficult to obtain these parameters directly, this expression will only be used for a qualitative evaluation. The major impact of the formula is to establish the dependence of lubricant film thickness (\( H_L \)) from the various parameters.

Allowable tooth stress is the subject of much uncertainty and most gear manufacturers have a proprietary method for establishing this criterion. Therefore, it is usually a stated parameter from the manufacturer that is used. The use of allowable stress published by the American Gear Manufacturers Association (AGMA) will usually result in satisfactory gear performance.

To ensure smooth operation of the gear mesh under load, it is generally the practice to modify the involute profile, usually with tip relief, to correct for the deflection of the gear tooth under load. The various parameters affecting gear wear are shown in Figure
8.3. Too little relief will result in the gear teeth going into mesh early and going out of mesh late. This condition results in higher dynamic loads with the accompanying stress, vibration, noise and possible non-involute contact that can lead to hard-lines, scuffing or scoring of gear teeth. Too much tip relief lowers the contact ratio of the gear set and again can result in less than optimum performance with respect to stress, vibration, and noise.

![Diagram of Gear Tooth Designs](image)

**Figure 8.3 Typical Gear Tooth Designs**

Crowning is generally applied to spur gears to ensure full contact across the face of the gear without end loading. With insufficient crowning, end loading will occur and result in higher than predicted vibrational stresses.

Helical gears, shown in Figure 8.4, are usually quieter and have a greater load-carrying capacity per inch of face than spur gears. The major disadvantage is that a thrust load is introduced along the gear shaft, thereby requiring larger and stronger bearings. Analysis of helical gears is very similar to that used for spur gears. The stress analysis is performed using an equivalent spur tooth. AGMA standard procedures have been developed for strength analysis of spur and helical gears.

### 8.2.2 Spiral Bevel Gears

The geometry of spiral bevel gears is considerably more complex than the spur or helical gear; therefore spiral bevel gears are probably the most difficult type of gear to design and analyze. The gear spiral is designed, if possible, so that axial forces tend to push both the pinion and gear out of mesh. If this design is impossible, then the spiral is chosen so the pinion is forced out of mesh. The face contact ratio of the mesh should be as high as possible to ensure quiet running. The face width of the spiral bevel gear should never exceed one-third of the outer cone distance to prevent load concentration on the toe of the gear and possible tooth breakage.
8.2.3 Planetary Gears

Planetary gear units are used in many designs, because they offer relatively large speed reduction in a compact package. The load shared among the pinions and the face width of the planetary gear is much less than that which would be required for a single mesh reduction. From a design point of view, it is desirable to use as many pinions as possible. It is normally desired to refrain from equally spaced planets meshing in unison with a sun or ring gear. The most common problem with this design is thrust washer wear. The excessive wear generally results from an inadequate supply of lubricant to the thrust washer area. The spherical bearing type support is generally preferred from a reliability point of view, since there are fewer parts and the thrust washer problem is eliminated. The spherical bearings also allow the pinions to maintain alignment with the sun and ring gears despite the deflection of the pinion posts. Despite the advantage of this design, it may be impossible to provide adequate support for cantilevered pinions in high torque situations, thereby requiring a two-plate design.

8.2.4 Spline Gear

Spline gears are used to transfer torque between shafts and flanges, gears and shafts, and shafts and shafts. A typical spline arrangement is shown in Figure 8.5. A splined shaft usually has equally spaced teeth around the circumference, which are most often parallel to the shaft's axis of rotation. These teeth can be straight sided, an involute form or included angle form (serrations). The teeth on a straight sided spline have an equal tooth thickness at any point measured radially out from the axis of rotation. Conversely, the internal parallel spline keys are integral to the shaft and equally spaced around the circumference. The involute spline has equally spaced teeth but they have an involute form like a gear tooth. The teeth do not have the same
proportions as a gear tooth. They are shorter in height to provide greater strength. Involute splines provide a more smooth transition through a radius as opposed to straight sided splines decreasing the possibility of fatigue cracks. Involute splines are usually crowned. The serration type of spline has a tooth that is non-involute. The male teeth are in the form of an included angle, with the female serration having spaces of the same included angle. Serrations are generally used on smaller diameter shafts where the included angle form permits more teeth to be used on a smaller circumference, providing a greater contact area.

The most common problem associated with splines is wear due to fretting; particularly, with loose splines. Strict attention must be given to the maintenance of bearing stress below the allowable limit. Tight splines should have an adequate length pilot to react with bending loads. Lubrication is a particular factor in the reliability of loose splines and, if at all possible, should remain flooded with oil at all times. Crowning is usually required to prevent excessive wear.

![Figure 8.5 Typical Spline Gears](image)

**Figure 8.5 Typical Spline Gears**

### 8.3 GEAR FAILURE MODES

The definition of failure for a gear is not very precise because of the wearing pattern of the gear. During the initial period of operation, minor imperfections in the gear will be smoothed out, and the working surfaces will polish up, provided that proper conditions of installation, lubrication and application are being met. Under continued normal conditions of operation, the rate of wear will be negligible. A gear has failed when it can no longer “efficiently” perform the job for which it was designed. Thus the definition of failure may be determined by the amount of vibration, noise, or results of a physical inspection.

The more common modes of gear failure are wear, surface fatigue, plastic flow and breakage. In the shear mode, the gear immediately ceases to transmit power while in the wear mode it degrades gradually before complete failure.
8.3.1 Wear

Wear is the removal of metal, worn away normally in a uniform manner from the contacting surface of the gear teeth. The first stage of wear is the polishing phase during wear-in of the gear when asperities of the contacting surfaces are gradually worn off until very fine, smooth surfaces develop. Moderate wear of the gear occurs during its design life. Moderate wear occurs most commonly when the gear is operating in or near the boundary lubrication regime. Many gears, because of practical limits on lubrication viscosity, speed and temperature, must of necessity operate under such conditions. Contamination in the lubrication system can accelerate this wear. Excessive wear is similar to moderate wear but the gear teeth are experiencing a considerable amount of material being removed from the surfaces. During this phase the tooth-surface profile is being destroyed so that high dynamic loads are encountered which in turn accelerates the wear rate until the gear is no longer usable.

Specific types of gear wear include abrasive wear caused by an accumulation of abrasive particles in the lubrication; corrosive wear caused by water or additives in the lubricating oil resulting in a deterioration of the gear surface from chemical action; and scoring caused by failure of the lubricant film due to overheating resulting in metal-to-metal contact and alternate welding and tearing of the surface metal.

8.3.2 Surface Fatigue

Surface fatigue is the failure of gear material as a result of repeated surface or subsurface stresses that are beyond the endurance limit of the material. Surface fatigue results in removal of metal and the formation of cavities. This pitting can be caused by the gear surfaces not properly conforming to each other due to lack of proper alignment. Spalling is similar to pitting except that the pits are larger, shallower and very irregular in shape. Spalling is usually caused by excessively high contact stress levels. The edges of the initial pits break away and large irregular voids are formed.

8.3.3 Plastic Flow

Plastic flow is the cold working of the tooth surfaces, caused by heavy loads and the rolling and sliding action of the gear mesh. The result of these high contact stress levels is the yielding of the surface and subsurface material and surface deformation. This same failure mode in a slow speed operation combined with an inadequate lubricating film can result in a rippled surface. The cold working action of the gear surface leads to deteriorated gear box operation.

8.3.4 Breakage

Breakage is a failure caused by the fracture of a whole tooth or a substantial portion of a tooth. Gear overload or cyclic stressing of the gear tooth at the root beyond the endurance limit of the material causes bending fatigue and eventually a crack
originating in the root section of the gear tooth and then the tearing away of the tooth or part of the tooth. Gear overload can be caused by a bearing seizure or sudden misalignment of a failed bearing, system dynamic loading, or contaminants entering the mesh area. Stress risers can sometimes subject the gear to higher root stress levels than originally predicted. These stress risers include such abnormalities as notches in the root fillet and small cracks from the heat treating or grinding process.

### 8.3.5 Summary of Gear Failure Modes

Table 8-1 provides a summary of possible failure modes for gears and splines

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pitting</td>
<td>Cyclic contact stress transmitted through lubrication film</td>
<td>Tooth surface damage</td>
</tr>
<tr>
<td>Root fillet cracking; Tooth end cracks</td>
<td>Tooth bending fatigue</td>
<td>Surface contact fatigue and tooth failure</td>
</tr>
<tr>
<td>Tooth shear</td>
<td>Fracture</td>
<td>Tooth failure</td>
</tr>
<tr>
<td>Scuffing</td>
<td>Lubrication breakdown</td>
<td>Wear and eventual tooth failure</td>
</tr>
<tr>
<td>Plastic deformation</td>
<td>Loading and surface yielding</td>
<td>Surface damage resulting in vibration, noise and eventual failure</td>
</tr>
<tr>
<td>Spalling</td>
<td>Fatigue</td>
<td>Mating surface deterioration, welding, galling, eventual tooth failure</td>
</tr>
<tr>
<td>Tooth bending fatigue</td>
<td>Surface contact fatigue</td>
<td>Tooth failure</td>
</tr>
<tr>
<td>Contact fatigue</td>
<td>Surface contact fatigue</td>
<td>Tooth failure</td>
</tr>
<tr>
<td>Thermal fatigue</td>
<td>Incorrect heat treatment</td>
<td>Tooth failure</td>
</tr>
<tr>
<td>Abrasive wear</td>
<td>Contaminants in the gear mesh area or lubrication system</td>
<td>Tooth scoring, eventual gear vibration, noise</td>
</tr>
</tbody>
</table>

### 8.4 GEAR RELIABILITY PREDICTION

The previous paragraphs have provided an insight into the specific characteristics and failure modes of the more common gear types. Gears, fortunately, are designed to
a specification and through the standardization of the American Gear Manufacturer's Association (AGMA), gears of various manufacturers and designs can be compared. The best approach for the calculation of failure rates for a gear system is to use the manufacturer's specification for each gear as the base failure rate, and adjust the failure rate for any difference in the actual usage from that purpose for which the gear was designed. If the manufacturer's failure rate is not available, a gear or spline is usually designed for a life of 100 million revolutions for the particular application, the application including such factors as operating speed, temperature, lubrication and torque. Either way, the gear failure rate can be expressed as:

\[ \lambda_G = \lambda_{G,B} \cdot C_{GS} \cdot C_{GP} \cdot C_{GA} \cdot C_{GL} \cdot C_{GT} \cdot C_{GV} \]  

(8-2)

Where:

- \( \lambda_G \) = Failure rate of gear under specific operation, failures/million operating hours
- \( \lambda_{G,B} \) = Base failure rate of gear, failures/million operating hours*
- \( C_{GS} \) = Multiplying factor considering speed deviation with respect to design (See Section 8.4.1 and Figure 8.6)
- \( C_{GP} \) = Multiplying factor considering actual gear loading with respect to design (See Section 8.4.2 and Figure 8.7)
- \( C_{GA} \) = Multiplying factor considering misalignment (See Section 8.4.3 and Figure 8.8)
- \( C_{GL} \) = Multiplying factor considering lubrication deviation with respect to design (See Section 8.4.4 and Figure 8.9)
- \( C_{GT} \) = Multiplying factor considering the operating temperature (See Section 8.4.5)
- \( C_{GV} \) = Multiplying factor considering the AGMA Service Factor (See Section 8.4.6 and Table 8-1)

* \( \lambda_{G,B} \) can usually be obtained from the manufacturer and it will be expressed in failures/operating hour at a specified speed, load, lubricant, and temperature. Also, a service factor will usually be provided to adjust the normal usage factor for certain specific conditions found in typical industries. These factors include such things as vibration, shock and contaminates. Failure data for similar equipment may also be available or a base failure rate of one failure/10^8 revolutions can be used.
8.4.1 Velocity Multiplying Factor

The speed deviation multiplying factor, $C_{GS}$, can be calculated using the relationship established in Equation (8-1) noting that the lubrication film thickness varies with speed to the 0.7 power. Therefore:

$$C_{GS} = k + \left( \frac{V_o}{V_d} \right)^{0.7} \quad (8-3)$$

Where:
- $V_o = \text{Operating Speed, RPM}$
- $V_d = \text{Design Speed, RPM}$
- $k = \text{Constant, 1.0}$

See Figure 8.6

8.4.2 Gear Loading Multiplying Factor

The gear loading multiplying factor, $C_{GP}$, has a lubricant and a fatigue impact. From Equation (8-1), the impact of load or torque can be expressed as:

$$\text{Change in expected life} = \frac{k}{W^{0.13}} = \frac{k}{\left( \frac{L_O}{L_D} \right)^{0.13}} \quad \text{(lubricant impact)} \quad (8-4)$$

Where:
- $W = \text{Load Parameter}$
- $L_O = \text{Operating Load, lbs}$
- $L_D = \text{Design Load, lbs}$
- $k = \text{Constant}$

and the expression for torque or load on the fatigue rate of the component is:

$$\text{Change in expected life} = k \left( \frac{L_D}{L_O} \right)^{4.56} \quad \text{(fatigue impact)} \quad (8-5)$$
Combining Equations (8-4) and (8-5):

\[ C_{GP} = \left( \frac{L_O}{L_D} \right)^{4.69} \]  

(8-6)

Where: \( k = \) Constant, 0.50

See Figure 8.7

8.4.3 Misalignment Multiplying Factor

The alignment of gears, bearings and shafts can be critical in the operation of a system. \( C_{GA} \), the misalignment factor, can be expressed as:

\[ C_{GA} = \left( \frac{A_E}{0.006} \right)^{2.36} \]  

(8-7)

Where: \( A_E = \) Misalignment angle in radians

See Figure 8.8

8.4.4 Lubricant Multiplying Factor

The lubricant factor \( C_{GL} \) is a function of the viscosity of the lubricant used in a gear system. \( C_{GL} \) can be expressed as:

\[ C_{GL} = \left( \frac{V_O}{V_L} \right)^{0.54} \]  

(8-8)
Where: \( \nu_O \) = Viscosity of specification lubricant, lb-min/in\(^2\)

\( \nu_L \) = Viscosity of lubricant used, lb-min/in\(^2\)

See Figure 8.9

### 8.4.5 Temperature Multiplying Factor

Temperature conditions of the gear system have an impact on other parameters such as \( C_{GL} \) and \( C_{GP} \). As the temperature increases, the lubricant viscosity decreases and the dimensions of the gears, shafts and bearings increase. This change normally causes a closer tolerance between operating units and an increase in the frictional losses in the system. To compensate for the decline in static and dynamic strengths, creep, and thermal expansion at high temperatures, the temperature factor, \( C_T \), represented by Equation (8-9) is applicable for temperatures greater than 160\(^\circ\)F (Reference 19). The multiplying factor for temperature \( C_{GT} \) can be expressed as:

\[
C_{GT} = \frac{460 + T_{AT}}{620} \quad \text{for} \quad T_{AT} > 160^\circ F
\]  \hspace{1cm} (8-9)

and:

\[
C_{GT} = 1.0 \quad \text{for} \quad T_{AT} \leq 160^\circ F
\]

where: \( T_{AT} \) = Operating temperature, \(^\circ\)F

### 8.4.6 AGMA Multiplying Factor

The AGMA has developed service factors for most industrial applications of gears, bearings, and gearbox designs whereby the expected extent of usage in vibration and shock environments can be taken into account when a gear system is selected for use. This service factor can be used as a multiplying factor for determining the inherent reliability or expected failure rate (\( C_{GV} \)) for a specific gearbox or bearing in a particular environment. Most manufacturers provide service factor data for each of their products. An example of a service factor for a speed-decreasing drive is shown in Table 8-1.

\[
C_{GV} = \text{AGMA Service Factor}
\]  \hspace{1cm} (8-10)
8.5 SPLINE RELIABILITY PREDICTION

The failure rate in failures per million revolutions of spline gears can be calculated by:

\[
\lambda_{GS,B} = \frac{10^6}{\theta}
\]  

(8-11)

where:  
\(\lambda_{GS}\) = Failure rate of spline gear in failures/million revolutions  
\(\theta\) = Life of spline gear in revolutions

An analytical expression for the spline gear life, \(\theta\), has been devised by Canterbury and Lowther (Reference 11). This equation is expressed as:

\[
\theta = 7.08 \times 10^{-10} \left( \phi \frac{G_L}{G_D} \right)^{4.56} \left( A_E \right)^{-2.36}
\]  

(8-12)

Where:  
\(G_L\) = Spline length, in  
\(G_D\) = Spline diameter, in  
\(A_E\) = Misalignment angle, radians  
\(\phi\) = Load Factor = \(\frac{4.85 \times 10^3 G_B \left( G_D \right)^3}{G_T}\)  
\(G_T\) = Torque, in-lbs  
\(G_B\) = Tooth hardness (Brinell), lbs/in²

Substituting the expression for the spline gear base failure rate into Equation (8-11) yields:

\[
\lambda_{GS} = \lambda_{GS,B} \cdot C_{GS} \cdot C_{GL} \cdot C_{GT} \cdot C_{GV}
\]  

(8-13)

where \(C_{GS}\), \(C_{GL}\), \(C_{GT}\), and \(G_{GV}\) are calculated by Equations (8-3), (8-8), (8-9), and (8-10) respectively.
Figure 8.6  Gear Velocity Multiplying Factor

\[ C_{GS} = 1.0 + \left( \frac{V_o}{V_d} \right)^{0.7} \]

Where:

\( V_o \) = Operating speed, RPM

\( V_d \) = Design speed, RPM
\[ C_{GP} = \left( \frac{L_O}{L_D} \right)^{4.69} \left( \frac{0.5}{L_D} \right) \]

Where: 
- \( L_O \) = Operating load, lbs
- \( L_D \) = Design load, lbs

\textbf{Figure 8.7 Gear Load Multiplying Factor}
Misalignment Multiplying Factor, $C_{GA}$

\[ C_{GA} = 12.44 \times A_E^{2.36} \]

Where: \( A_E \) = Misalignment angle, degrees

**Figure 8.8 Gear Misalignment Multiplying Factor**
$C_{GL} = \left(\frac{\nu_O}{\nu_L}\right)^{0.54}$

Where:  
$\nu_O$ = Viscosity of specification fluid  
$\nu_L$ = Viscosity of lubricant used

**Figure 8.9 Gear Lubricant Multiplying Factor**
Table 8-1. Typical AGMA Service Factor, $C_{av}$

<table>
<thead>
<tr>
<th>Prime Mover</th>
<th>Character of Load on Driven Member</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Uniform</td>
</tr>
<tr>
<td>Uniform</td>
<td>1.00</td>
</tr>
<tr>
<td>Medium Shock</td>
<td>1.25</td>
</tr>
<tr>
<td>Heavy Shock</td>
<td>1.50</td>
</tr>
</tbody>
</table>

8.6 REFERENCES


54. AGMA Standard for Surface Durability Formulas for Spiral Bevel Gear Teeth, American Gear Manufacturers Association, Report 216.01 (January 1964)


71. Dennis N. Pratt, “Investigation of Spline Coupling Wear”, Report No. SY-51R-87, Naval Air Warfare Center, Patuxent River, MD (December 1987)


CHAPTER 9

ACTUATORS

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9.1 INTRODUCTION

Actuators provide the means to apply mechanical power to systems when and where it is needed. In general, actuators take energy from pumped fluid and convert it to useful work. This conversion is accomplished by using the pumped fluids to generate a differential pressure across a piston, which results in a force and motion being generated. This chapter will identify some of the more common failure modes and failure causes of actuators, and will develop and discuss a failure rate model for actuators.

In general, there are two types of output motions generated by actuators: linear and rotary. Within these two classifications there are many different types of actuator assemblies.

9.1.1 Linear Motion Actuators

Linear motion actuators are usually a derivative of one of the following four types:

1. Single acting
2. Double acting
3. Ram
4. Telescoping
Single acting actuators are the simplest type of the four. Pressurized fluid acts only on one side of the piston so the single acting actuator is capable of generating motion and power only in one direction and requires an external force to move the piston in the opposite direction.

Double acting actuators have fluid chambers on both sides of the piston, which allows pressurized fluid to both extend and retract the piston/rod and provide a faster response. Double acting actuators may have rods extending from either or both ends of the cylinders. Those with rods extending from both ends are balanced; that is, the piston moves at the same rate and delivers equal forces in each direction.

Ram cylinders are a variation on the single acting design, but in this case, the piston rod is the same diameter as the piston. This design is useful where column loads are extremely high or when the rod hanging in a horizontally mounted cylinder has a tendency to cause sagging.

Telescoping cylinders generate long stroke motions from a short body length. Force output varies with rod extension: highest at the beginning, when the pressurized fluid acts on all of the multiple piston faces; and lowest at the end of the stroke, when the pressurized fluid acts only on the last extension's piston area. Telescoping cylinders may be either single or double acting.

9.1.2 Rotary Motion Actuators

Rotary actuators produce oscillating power by rotating an output shaft through a fixed arc. Rotary actuators are primarily one of two types:

1. Linear motion piston/cylinder with rotary output transmission
2. Rotary motion piston/cylinder coupled directly to output shaft

The first of the two rotary actuator types generally uses one or two linearly moving pistons to drive a transmission to convert the linear motion produced by the piston to a rotary output motion. These rotary actuators generally use crankshafts, gear rack-and-pinions, helical grooves, chains and sprockets, or scotch-yoke mechanisms as transmissions to convert the piston's linear output to rotary output. The piston/cylinder design may be single or double acting.

The second of the two rotary actuator types uses a piston designed to oscillate through a fixed arc to directly drive the output shaft. This design is simpler than the other type of rotary actuator as no transmission is required, but the unusual piston shapes required may create sealing problems.
9.2 ACTUATOR FAILURE MODES

The primary failure mode of an actuator is a reduction in output force or stroke. This reduction in actuator output power can be caused by excessive wear of the piston/cylinder contact surfaces, which results in an increase in fluid leakage past the piston. Reduction in actuator output power can also be caused by external leakage, such as leakage through the piston rod/rod seal interface. Deterioration of the piston rod seal also permits ingestion of contaminants to the gap between the piston and cylinder increasing the rate of wear and probability of problems associated with corrosion.

Another common failure mode for actuators is jamming of the piston caused by stiction or misalignment. This failure can occur if excessive contaminants are ingested or if excessive side loads are encountered. Misalignment also increases the rate of piston/cylinder wear contributing to early failure. Depending on the equipment design, one potential failure mode requiring investigation is the loss of signal that a loss of accurate positioning of an actuator can cause to software programming or valve controls.

Temperature extremes may affect the viscosity characteristics of the pressurized fluid and increased seal wear will result from the resultant change in film lubrication.

Table 9-1. Typical Modes of Actuator Failure

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal leakage</td>
<td>Side loading and piston</td>
<td>Loss/reduction in output force</td>
</tr>
<tr>
<td></td>
<td>wear</td>
<td></td>
</tr>
<tr>
<td>External leakage</td>
<td>Seal leakage, piston</td>
<td>Loss/reduction in output force</td>
</tr>
<tr>
<td></td>
<td>wear</td>
<td></td>
</tr>
<tr>
<td>Spurious position</td>
<td>Stiction, binding</td>
<td>Loss of output control or incorrect signal transmission</td>
</tr>
<tr>
<td>change</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Jamming, seizure</td>
<td>Excessive loading</td>
<td>Loss of load control</td>
</tr>
</tbody>
</table>
forces acting inside the actuator as a result of pressure variations, pressure differentials, friction forces, temperature-related expansion and contraction, and by forces developed and transmitted by the impact of external loads.

Valves often form a part of an actuator assembly and are used for primary movement control of the actuator and also for deceleration of the piston/rod assembly at the ends of their stroke. Failure rate models for valve assemblies are presented in Chapter 6 of this handbook.

The complete failure rate model for the piston/cylinder actuator incorporates modifiers for contamination and temperature effects. The complete model can be expressed as follows:

$$\lambda_{AC} = \lambda_{AC,B} \cdot C_{CP} \cdot C_T$$

Where:

- $\lambda_{AC}$ = Failure rate of actuator, failures/million cycles
- $\lambda_{AC,B}$ = Base failure rate of actuator, failures/million cycles  
  [See Equations (9-14) and (9-15)]
- $C_{CP}$ = Contaminant multiplying factor (See Section 9.3.2)
- $C_T$ = Temperature multiplying factor (See Section 9.3.3)

9.3.1 Base Failure Rate for Actuator

The primary failure effect of internal and external loads on an actuator is wear of the piston and cylinder which results in an increase in leakage past the piston. A criteria of actuator failure would then be a leakage rate resulting from wear which exceeds a maximum allowable leakage rate specified by the user.

Wear of the cylinder and piston will occur in two phases according to the Bayer-Ku sliding wear theory (Reference 6). The first or constant wear phase is characterized by the shearing of the surface asperities due to the sliding action of the piston within the cylinder. During this period the wear rate is practically linear as a function of the number of actuator cycles and the wear depth at the end of the constant wear phase is one half the original surface finish. During the second or severe wear phase, wear debris becomes trapped between the two sliding surfaces and gouging of the surfaces takes place. The wear rate begins to increase very rapidly and failure of the actuator is eminent. Therefore, while equations are presented in this chapter for the severe wear phase, for practical purposes the failure rate or life of the actuator can be estimated as that calculated for the constant wear phase.
The number of cycles to complete the constant wear phase can be predicted analytically by a semi-empirical modification of Palmgren's equation (Reference 6) resulting in the formula:

\[ N_O = 2000 \left( \frac{\gamma F_y}{S_c} \right)^9 \]  \hspace{1cm} (9-2)

Where:
- \( N_O \) = Number of cycles in constant wear phase
- \( \gamma \) = Wear factor
- \( F_y \) = Yield strength of softer material, lbs/in²
- \( S_c \) = Compressive stress between the surfaces, lbs/in²

The wear factor, \( \gamma \), will be equal to 0.20 for materials that have a high susceptibility to adhesive wear, in which the wear process involves a transfer of material from one surface to the other. The wear factor will be equal to 0.54 for materials that have little tendency to transfer material in which the material is subject to micro-gouging of the surfaces by the asperities on the material surface.

The maximum compressive stress caused by the cylinder acting on the piston is computed assuming a linear distribution of stress level along the contact area. The following equation has been derived for compressive stress of an actuator:

\[ S_c = 0.8 \left( \frac{W_S \cdot D_1 - D_2}{L D_1 D_2} \right)^{1/2} \left( \frac{1}{1 - \eta_1^2} + \frac{1}{E_1} + \frac{1}{E_2} \right) \]  \hspace{1cm} (9-3)

Where:
- \( W_S \) = Side load on the actuator, lbf
- \( L \) = Total linear contact between piston and cylinder, in
- \( D_1 \) = Diameter of cylinder, in
- \( D_2 \) = Diameter of piston, in
- \( \eta_1 \) = Poisson's ratio, cylinder
- \( \eta_2 \) = Poisson's ratio, piston
- \( E_1 \) = Modulus of elasticity, cylinder, lbs/in²
- \( E_2 \) = Modulus of elasticity, piston, lbs/in²
Substituting Equation (9-3) into Equation (9-2) and adding a constant for lubrication provides an equation for the number of cycles for an actuator during Phase I wear until the severe wear period begins.

\[ N_O = k_1 \left[ \gamma F_y \left( \frac{W_S \cdot D_1 - D_2}{L \cdot D_1 D_2} \right)^{1/2} \right]^{9} \]  

(9-4)

Where: \( k_1 = 15.4 \times 10^3 \) which includes lubrication constant

In a similar way, if the actuator is subjected to axial stress, equation (9-5) can be used to determine compressive stress. Which equation to use depends on the application of the actuator, axial or side loading.

\[ S_C = 0.9 \left( \frac{W_A \cdot \left( \frac{D_1 - D_2}{D_1 D_2} \right)^2}{\left( \frac{1 - \eta_1^2}{E_1} + \frac{1 - \eta_2^2}{E_2} \right)^2} \right)^{1/3} \]  

(9-5)

Where: \( W_A \) = Axial Load on the actuator, lbf

And:

\[ N_O = k_2 \left[ \gamma F_y \left( \frac{W_A \cdot \left( \frac{D_1 - D_2}{D_1 D_2} \right)^2}{\left( \frac{1 - \eta_1^2}{E_1} + \frac{1 - \eta_2^2}{E_2} \right)^2} \right)^{1/3} \right]^{9} \]  

(9-6)
Where:  \[ k_2 = 17.7 \times 10^3 \] which includes lubrication constant.

Since the second phase of wear is severe and relatively short, it can normally be assumed that the calculated number of cycles, \( N_o \), for the first phase of wear will be the life of the actuator. During the second or severe wear phase, the following equation can be used to determine the rate of wear (Reference 45):

\[
V = \frac{KWd}{H} (N - N_o) \tag{9-7}
\]

Where:  
- \( V \) = Volume of material removed by wear during the second phase, in \(^3\)
- \( K \) = Wear coefficient (See Table 9-3)
- \( W \) = Applied load, lb
- \( d \) = Sliding distance, in
- \( H \) = Penetration hardness, psi
- \( N \) = Total number of cycles to failure
- \( N_o \) = Number of cycles at the end of the initial wear phase

Solving for \( N \) results in the equation:

\[
N = \frac{VH}{KWd} + N_o \tag{9-8}
\]

This second phase of wear is characterized by rapid wear until failure of the actuator occurs usually as a result of poor response due to excessive leakage. The leakage rate past the piston within the cylinder may be modeled as laminar flow between parallel plates (Reference 5).

\[
Q = \frac{\pi D_z a^3 \Delta p}{12 \nu L} \tag{9-9}
\]
Where:

\( Q \) = Leakage rate past piston, in\(^3\)/sec

\( D_2 \) = Piston diameter, in

\( a \) = Gap between piston and cylinder, in

\( \Delta p \) = Pressure differential across piston, psi

\( \nu \) = Fluid viscosity, lbf-sec/in\(^2\)

\( L \) = Piston length, in

The gap between the piston and cylinder, \( a \), as shown in Figure 9.1 is a dynamic term being a function of wear.

\[
a = (D_1 - D_2) + h
\]  \hspace{1cm} (9-10)

Where:

\( D_1 \) = Cylinder diameter, in

\( h \) = Depth of wear scar, in

![Diagram of piston and cylinder](image)

Figure 9.1 Typical Single Acting Actuator

The wear scar depth, \( h \), will be equal to the volume of material lost due to wear, \( V \), divided by the contact surface area, \( A \):

\[
h = \frac{V}{A}
\]  \hspace{1cm} (9-11)
Substituting Equations (9-10) and (9-11) for wear gap into Equation (9-9) results in the following equation for leakage rate between the piston and cylinder:

\[
Q = \frac{\pi D_2 \left[ (D_1 - D_2) + \frac{V}{A} \right]^3}{12 \nu L} \Delta p
\]  

(9-12)

Solving Equation (9-12) for \( V \) and substituting \( V \) in Equation (9-7) results in an equation for the number of cycles to failure.

\[
N = \frac{AH}{KDd} \left[ \left( \frac{12Q \nu L}{\pi D_2 \Delta p} \right)^{1/3} + (D_2 - D_1) \right] + N_0
\]  

(9-13)

A similar equation can be developed by combining equations (9-13) and (9-6) for axial loading of the actuator. A typical plot of wear as a function of the number of cycles is shown in Figure 9.2.

![Failure Rate vs. Cycles](image)

**Figure 9.2** Failure Rate as a Function of Cycles for a Typical Actuator Under Different Side Loads.
Since the first phase of wear is fairly linear as a function of the number of cycles and failure will occur soon after phase one wear, the base failure rate of the actuator can be approximated as follows:

\[
\lambda_{AC,B} = \frac{10^6}{N_O}
\]  

(9-14)

Where:  
\[\lambda_{AC,B}\] = Base failure rate of actuator, failures/million cycles  
\[N_O\] = Number of cycles in constant wear phase

9.3.2 Contaminant Multiplying Factor

As established in Equation (9-1), the failure rate of the actuator can be determined as follows:

\[
\lambda_{AC} = \lambda_{AC,B} \cdot C_{CP} \cdot C_T
\]  

(9-15)

Where:  
\[\lambda_{AC}\] = Failure rate of actuator, failures/million cycles  
\[\lambda_{AC,B}\] = Base failure rate of actuator, failures/million cycles  
\[C_{CP}\] = Contaminant multiplying factor  
\[C_T\] = Temperature multiplying factor

During the time that the actuator is at rest, particles can work their way between the piston and cylinder. Then, when the actuator is put into motion, increased forces are needed to move the piston. This stick phenomenon causes a loss of actuator response and in some severe cases, a completely jammed component.

Three types of wear need to be considered in determining the effects of contaminants on actuator reliability:

1. Erosion - Particles carried in a fluid stream impact against the piston and cylinder surfaces. If the kinetic energy released upon actuator response is large compared to forces binding the piston/cylinder walls, surface fatigue will occur. Hard particles may also cut away surface material.

2. Abrasive Wear - A hard particle entering the gap between the piston and cylinder surfaces can cut away material of the softer surface on a single actuator engagement. The rate of wear will be proportional to the number of particles in contact
with the surfaces and the particle hardness. If the hardness of the piston is significantly less than that of the cylinder, a hard particle, absorbed by the softer material causes severe abrasive wear of the harder actuator surface.

(3) Surface Fatigue - Particulate contaminants interacting with the piston and cylinder surfaces can dent a surface producing plastic deformation. Large numbers of dislocations will increase the surface roughness and deteriorate the surface material. The result is an accelerated rate of wear and a higher probability of leakage between the surfaces.

The deteriorating effects of contaminant particles on the reliability of an actuator must be equated along with the probability of the contaminants entering the gap between the actuator surfaces. The probability of contaminants entering this area will depend on the operating environment, the types and numbers of particles expected to be encountered, and the filtering system to prevent the entrance of particles. The typical actuator contains a bushing to wipe the piston on the return stroke. The life expectancy and reliability of this device must be determined as part of the overall reliability estimate of the actuator.

If the piston surface slides over a hard contaminant particle in the lubricant, the surface may be subject to pitting. The abrasive particle has edges with a characteristic radius, denoted by \( r \). When the depth of penetration of the abrasive particle \( (d) \) reaches a certain critical value, the scratching produces additional wear particles by pitting. This elastic/plastic deformation process occurs when the maximum shear stress in the complex stress distribution beneath the contact surface exceeds the elastic limit. This maximum shear stress occurs beneath the contact at a depth equal to one half the contact radius. The value of this critical depth is given by (Reference 48).

\[
d_{\text{crit}} = r \left\{ \frac{1 - \frac{2f_{s,\text{max}}}{F_{sy}}}{2} \right\}
\]

(9-16)

Where:
- \( f_{s,\text{max}} \) = Maximum shear stress, lbs/in\(^2\)
- \( r \) = Characteristic radius of particle, in
- \( F_{sy} \) = Yield strength of material, lbs/in\(^2\)

If this type of wear should occur, it is so severe that actuator performance would be immediately affected and failure would occur. Actuators are designed to prevent particles of sufficient size to cause this type of failure and the probability of failure from this type of pitting is extremely low. The failure mode is presented here as a design evaluation check on the sealing technique for the piston assembly.
Fatigue wear on the microscopic wear due to contaminants is similar to that for pitting just described except that it is associated with individual asperity contacts rather than with a single large region. The additional material lost due to contaminant wear process can be estimated in the same way as the adhesive wear process was explained earlier in this chapter, the volume $\delta V$ removed on an individual piston stroke proportional to $a^2$ where $a$ is the radius of the individual area of contact. Similarly, the sliding distance $\delta L$ is proportional to $A$.

$$\frac{\delta V}{\delta L} \propto \frac{\delta A}{3} \quad (9-17)$$

Where: $A = \text{Area of contact, in}^2$

Summing for all contacts provides the following equation:

$$\frac{V}{L_1 N} = \frac{1}{3} K_1 A = \frac{K_1 W}{3 H_Y} \quad (9-18)$$

Where: $V = \text{Volume of material lost due to contaminant wear, in}^3$
$L_1 = \text{Sliding distance of the piston, in}$
$N = \text{Number of actuations}$
$K_1 = \text{Wear coefficient, See Table 9-3}$
$A = \text{Area of contact, in}^2$
$W = \text{Transverse load on the actuator, lbs}$
$H_Y = \text{Vickers hardness of the piston, lbs/in}^2$

This expression can be rewritten in the form to include a contaminant multiplying factor, $C_{CP}$:

$$V = \frac{C_{CP} W L_1 N}{H_Y} \quad (9-19)$$

The effect of the additional wear due to contaminant particles may be expressed as an additive term in the basic wear relationship. It will be noted from the derivation of equations for the effect of contaminant particles on actuator surface wear and the possibility of stiction problems that a probability of damaging particles entering the gap
between the piston and cylinder must be estimated. The contaminant factors involved are as follows:

**Hardness** - The wear rate will increase with the ratio of particle hardness to actuator surface hardness. It will normally be the hardness of the piston that will be of concern. If the ratio is less than 1, negligible wear can be expected.

**Number of particles** - The wear rate will increase with a concentration of suspended particles of sufficient hardness.

**Size** - For wear of the piston or cylinder to occur, the particle must be able to enter the gap between the two surfaces. The particle must also be equal to or greater than the lubrication film thickness. With decreasing film thickness, a greater proportion of contaminant particles entering the gap will bridge the lubrication film, producing increased surface damage.

**Shape** - Rough edged and sharp thin particles will cause more damage to the actuator surfaces than rounded particles. As the particles remain in the gap, they will become more rounded and produce less wear. It is the more recent particles being introduced into the gap that cause the damage.

$C_{CP}$ can be estimated by considering these variables and their interrelationship. The following factors can be used to estimate a value for $C_{CP}$:

$$C_{CP} = C_H \cdot C_S \cdot C_N$$

(9-20)

Where:

- $C_H =$ Factor considering ratio of particle to piston hardness
  
  (See Table 9-1)

- $C_S =$ Factor considering particle size entering gap between piston and cylinder

- $C_N =$ Factor considering the number of particles meeting hardness, size and shape parameters entering the gap (See Table 9-2)

### 9.3.3 Temperature Multiplying Factor

The effect of the temperature of the surface on the wear rate is a complicated phenomenon, because the corrosion of the wear debris at different temperatures produces different oxidation products. Chemical interactions with the metal surfaces result in different wear rates as the temperature of the surface is changed (Reference 51). For example, the formation of Fe$_3$O$_4$ is likely to predominate when steel is subject to wear in the temperature range between 570 °F and 930 °F (300 °C to 500 °C).
Wear of metals has been related to the heat of absorption of molecules of debris (Reference 51). The basic relationship in this treatment is:

\[ V = \frac{KWL}{Hv} C_o e^{\theta/T} \]  

(9-21)

Where:  
\( C_o \) = Arrhenius constant  
\( \theta \) = Activation energy constant, K  
\( T \) = Operating temperature, K

Values for the parameter \( \theta \) are in the range between 1200 K and 6000 K.

The effect of variation of temperature may be determined by eliminating the Arrhenius constant in terms of the value of the exponential at ambient temperature \( T_a \). Making this substitution into Equation (9-21), the following is obtained:

\[ V = \frac{C_T KWL}{Hv} N \]  

(9-22)

From Reference 45, the temperature multiplying factor, \( C_T \), is given by:

\[ C_T = e^{\theta/\theta a[1-(T_a/T)]} \]  

(9-23)

Where:  
\( T_a \) = Ambient temperature, 298.2 K  
\( T \) = Operating temperature, K

It is noted that the ratio \( \theta / T_a \) is in the range between 4.0 and 20.0.
# Table 9-1. Material Hardness

(Use ratio of hardest particle/cylinder hardness for $C_{ij}$)

<table>
<thead>
<tr>
<th>MATERIAL</th>
<th>HARDNESS $(H_v \times 10^6)$</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Plain carbon steels</strong></td>
<td></td>
</tr>
<tr>
<td>- Low strength steel</td>
<td>140</td>
</tr>
<tr>
<td>- High strength steel</td>
<td>220</td>
</tr>
<tr>
<td><strong>Low-alloy Steels</strong></td>
<td></td>
</tr>
<tr>
<td>- 4320</td>
<td>640</td>
</tr>
<tr>
<td>- 4340</td>
<td>560</td>
</tr>
<tr>
<td><strong>Stainless Steels</strong></td>
<td></td>
</tr>
<tr>
<td>- 303</td>
<td>170</td>
</tr>
<tr>
<td>- 304</td>
<td>160</td>
</tr>
<tr>
<td>- 631 (17-7 PH hardened)</td>
<td>520</td>
</tr>
<tr>
<td>- 631 (17-7 PH annealed)</td>
<td>170</td>
</tr>
<tr>
<td>- Austenitic AISI 201 annealed</td>
<td>210</td>
</tr>
<tr>
<td>- Martensitic 440C (hardened)</td>
<td>635</td>
</tr>
<tr>
<td>- 630 (17-4 PH hardened)</td>
<td>470</td>
</tr>
<tr>
<td><strong>Nickel Alloys</strong></td>
<td></td>
</tr>
<tr>
<td>- 201</td>
<td>100</td>
</tr>
<tr>
<td><strong>Nickel-copper Alloys</strong></td>
<td></td>
</tr>
<tr>
<td>- Monel (annealed)</td>
<td>120</td>
</tr>
<tr>
<td>- Monel K-500 (annealed)</td>
<td>162</td>
</tr>
<tr>
<td><strong>Ni-Cr-Mo-Fe Alloys</strong></td>
<td></td>
</tr>
<tr>
<td>- Inconel 625</td>
<td>140</td>
</tr>
<tr>
<td>- Hastelloy</td>
<td>200</td>
</tr>
<tr>
<td><strong>Aluminum</strong></td>
<td></td>
</tr>
<tr>
<td>- AISI 1100 (annealed)</td>
<td>25</td>
</tr>
<tr>
<td>- AISI 1100 (cold worked)</td>
<td>45</td>
</tr>
<tr>
<td>- AISI 2024 (annealed)</td>
<td>50</td>
</tr>
<tr>
<td>- AISI 2024 T4 (heat treated)</td>
<td>125</td>
</tr>
<tr>
<td>- AISI 6061 (annealed)</td>
<td>32</td>
</tr>
<tr>
<td>- AISI 6061 T6 (heat treated)</td>
<td>100</td>
</tr>
</tbody>
</table>
### Table 9-2. Typical Component Generation Rates

<table>
<thead>
<tr>
<th>COMPONENT</th>
<th>EXPECTED RATE OF CONTAMINANT GENERATION</th>
<th>$C_N$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear Pump</td>
<td>7.5 g/gpm rated flow</td>
<td></td>
</tr>
<tr>
<td>Vane Pump</td>
<td>25.0 g/gpm rated flow</td>
<td></td>
</tr>
<tr>
<td>Piston Pump</td>
<td>6.8 g/gpm rated flow</td>
<td></td>
</tr>
<tr>
<td>Directional Valve</td>
<td>0.008 g/gpm rated flow</td>
<td></td>
</tr>
<tr>
<td>Cylinder</td>
<td>3.2 g/in² swept area</td>
<td></td>
</tr>
</tbody>
</table>

* Add total grams of contaminants expected per hour/100 to determine $C_N$

### Table 9-3. Values of Wear Coefficient ($K_1$) In The Severe-Wear Region (Reference 45)

<table>
<thead>
<tr>
<th>MATERIAL</th>
<th>$K_1$</th>
</tr>
</thead>
<tbody>
<tr>
<td>4130 Alloy Steel (piston)</td>
<td>0.0218</td>
</tr>
<tr>
<td>4130 Alloy Steel (cylinder)</td>
<td>0.0221</td>
</tr>
<tr>
<td>17-4 PH Stainless Steel (piston)</td>
<td>0.0262</td>
</tr>
<tr>
<td>4130 Alloy Steel (cylinder)</td>
<td>0.0305</td>
</tr>
<tr>
<td>9310 Alloy Steel (piston)</td>
<td>0.0272</td>
</tr>
<tr>
<td>4130 Alloy Steel (cylinder)</td>
<td>0.0251</td>
</tr>
</tbody>
</table>

### 9.4 REFERENCES


10.1 INTRODUCTION

Pumps are one of the most common types of mechanical components used by today's society, exceeded only by electric motors. Not surprisingly, there are in existence today, an almost endless number of pump types that function in systems with dissimilar operating and environmental characteristics. With so many different pump types it is impossible to establish a failure rate data base based on design parameters, their use, and the materials used to construct them, or the type of fluid they move. All of these categories tend to overlap for the many different pump types. Therefore, a system to differentiate between all types of pumps is necessary. This system uses the way or means by which energy is added to the fluid being pumped, and is unrelated to application, material type, or outside considerations involving the pump. As seen by Figure 10.1, a pump can be classified into two general classes: Centrifugal and Positive Displacement.
These classes represent the two ways in which energy is added to the fluid. Centrifugal pumps consist of a set of rotating vanes, enclosed within a housing or casing, used to impart energy to a fluid through centrifugal force. The centrifugal pump has two main parts: a rotating element which includes an impeller and a shaft, and a stationary element made up of a volute casing, stuffing box, and bearings. With centrifugal pumps, the energy is added continuously by increasing the fluid velocity with a rotating impeller while reducing the flow area. This arrangement causes an increase in pressure along with the corresponding movement of the fluid. The impeller produces liquid velocity and the volute forces the liquid to discharge from the pump converting velocity to pressure. The stuffing box protects the pump from leakage at the point where the shaft passes out through the pump casing.

Centrifugal pumps can be further classified as to one of the following three designs:

Axial Flow - In an axial flow pump, pressure is developed by the propelling or lifting action of the impeller vanes on the liquid. Axial flow pumps are sometimes referred to as propeller pumps.

Radial or Mixed Flow – In a radial flow pump, the liquid enters at the center of the impeller and is directed out along the impeller blades in a direction at right angles to the pump shaft. The pressure is developed wholly by centrifugal force.

Peripheral - Peripheral pumps employ a special impeller with a large number of radial blades. As the fluid is discharged from one blade, it is transferred to the root of the next blade and given additional energy.
Positive displacement pumps differ from centrifugal pump designs in that energy is added to the fluid periodically by the movement of control boundaries with fluid volumes being displaced causing an increase in pressure. Displacement pumps can be subdivided into reciprocating and rotary types.

Reciprocating – A reciprocating pump is characterized by a back-and-forth motion of pistons inside of cylinders that provides the flow of fluid. Reciprocating pumps, like rotary pumps, operate on the positive principle, each stroke delivering a definite volume of liquid to the system. The master cylinder of the automobile brake system is an example of a simple reciprocating pump.

Rotary - Rotary pumps operate on the principle that a rotating vane, screw, or gear traps the fluid on the suction side of the pump casing and forces it to the discharge side of the casing. A rotary displacement pump is different from a centrifugal pump in that in a centrifugal pump, the liquid displacement and pumping action depend on developed liquid velocity.

Reliability models have been developed to address the difference between pump types. Because of the physical design differences between centrifugal and displacement pumps they have specific performance and reliability advantages and disadvantages. As shown in Figure 10.2 for example, centrifugal pumps are limited by pressure but can supply almost any amount of capacity desired. Displacement pumps lose capacity as the pressure increases due to the increase in slip which occurs with an increase in pump pressure. The amount of slip can vary from pump to pump depending on the actual manufactured clearances in the pump chamber. The slip can also increase with time as wear increases. Equation (10-1) shows that since slip "S" increases as the pressure requirements increase, the value of capacity "Q" is thus decreased:

\[
Q = 0.00433 D N - S \tag{10-1}
\]

Where:
- \( Q \) = Capacity, gpm
- \( D \) = Net fluid transferred or displaced by one cycle, ft³
- \( N \) = Rotation speed, revolutions/minute
- \( S \) = Slip, ft³/min (The quantity of fluid that escapes the full rotor cycle through clearances or other "leak paths")

Therefore, high pressure designs are somewhat limited to the amount of capacity, although slip can be reduced. For example, the slip can be reduced by decreasing the tolerances to the extent that the interference will not occur between moving parts. Interference can cause an extremely rapid reduction in pump performance.
10.2 FAILURE MODES

Due to the large number of pump types and applications, some failure modes are more prevalent than others for a specific category of pump. For example, with displacement pumps there is a much greater chance for cyclic fatigue to have an effect on the system than with centrifugal pumps. This is due to the inherent difference in designs. Displacement pumps have pressure transients which cause temporary unbalanced forces to be applied to the pump and its system. The displacement pump and driver shafting can experience much higher stresses during operation due to the uneven torque loading caused by this natural imbalance. On the other hand, the centrifugal pumps are more balanced and aren't as susceptible to large stress variations.

Suction energy is created from liquid momentum in the suction eye of a pump impeller. The Net Positive Suction Head (NPSH) is defined as the static head + surface pressure head – the fluid vapor pressure – the friction losses in the piping, valves and fittings. NPSH margin is defined as the NPSH available (NPSHA) to the pump by the application divided by the NPSH required (NPSHR) by the pump. NPSHR is the amount of suction head required to prevent pump cavitation and is determined by the pump design as indicated on the pump curve. NPSHA is the amount of suction available or total useful energy above the vapor pressure at the pump suction.

Typical failure modes of pumps are shown in Table 10-1 and Table 10-2.
## Table 10-1. Typical Failure Modes of Centrifugal Pump Assemblies

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reduction in suction head</td>
<td>- Pump cavitation</td>
<td>- Loss of pump efficiency</td>
</tr>
<tr>
<td>Reduction in pump pressure</td>
<td>- Pump cavitation</td>
<td>- Eventual erosion of impeller, casing</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Pump noise and vibration</td>
</tr>
<tr>
<td>Component corrosion</td>
<td>- Incorrect fluid</td>
<td>- Eventual catastrophic pump failure</td>
</tr>
<tr>
<td></td>
<td>- Excessive flow rate for fluid</td>
<td></td>
</tr>
<tr>
<td>Shaft deflection</td>
<td>- High radial thrust on pump rotor</td>
<td>- Eventual shaft and pump failure</td>
</tr>
<tr>
<td>Shaft unbalance</td>
<td>- Impeller wear</td>
<td>- Shaft deflection and Misalignment</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Stuffing box leakage</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Seal leakage</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Bearing wear</td>
</tr>
<tr>
<td>Air leak thru gasket / stuffing box</td>
<td>- Damaged gasket</td>
<td>- Loss of pump head</td>
</tr>
<tr>
<td>External Leakage</td>
<td>- Seal failure</td>
<td>- Depends on type of fluid and criticality as to time of failure</td>
</tr>
<tr>
<td></td>
<td>- Worn mechanical seal</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Scored shaft sleeve</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Stuffing box improperly packed</td>
<td></td>
</tr>
<tr>
<td>Mechanical noise</td>
<td>- Debris in the impeller</td>
<td>- Eventual wear of impeller and other components</td>
</tr>
<tr>
<td></td>
<td>- Impeller out of balance</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Bent shaft</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Worn/damaged bearing</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Foundation not rigid</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Cavitation</td>
<td></td>
</tr>
<tr>
<td>Positive suction head too low</td>
<td>- Clogged suction pipe</td>
<td>- Suction cavitation</td>
</tr>
<tr>
<td></td>
<td>- Valve on suction line only partially open</td>
<td>- Noisy operation</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Low discharge pressure</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- High output flow rate</td>
</tr>
<tr>
<td>Pump discharge head too high</td>
<td>- Clogged discharge pipe</td>
<td>- Discharge cavitation</td>
</tr>
<tr>
<td></td>
<td>- Discharge line valve only partially open</td>
<td>- Noisy operation</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Low output flow rate</td>
</tr>
<tr>
<td>Suction line / impeller clogged</td>
<td>- Contaminants</td>
<td>- Loss of pump output / reduced flow</td>
</tr>
<tr>
<td>Worn / broken impeller</td>
<td>- Wrong flow rate, contaminants</td>
<td>- Loss of pump output / reduced flow</td>
</tr>
<tr>
<td>Thrust bearing failure</td>
<td>- Excessive axial load</td>
<td>- Pump failure</td>
</tr>
<tr>
<td>FAILURE MODE</td>
<td>FAILURE CAUSE</td>
<td>FAILURE EFFECT</td>
</tr>
<tr>
<td>---------------------------------------</td>
<td>----------------------------------------------------</td>
<td>-----------------------------------------------------</td>
</tr>
<tr>
<td>Pump cavitation</td>
<td>- Reduction in suction head</td>
<td>- Pump noise and vibration</td>
</tr>
<tr>
<td></td>
<td>- Eventual erosion of rotor, casing</td>
<td></td>
</tr>
<tr>
<td>Component corrosion</td>
<td>- Incorrect fluid</td>
<td>- Eventual catastrophic pump failure</td>
</tr>
<tr>
<td>Low net positive</td>
<td>- Excessive flow rate for fluid</td>
<td>Reduced pump efficiency</td>
</tr>
<tr>
<td>suction head (NPSH)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shaft unbalance</td>
<td>- Torsional vibration</td>
<td>- Shaft deflection and misalignment</td>
</tr>
<tr>
<td></td>
<td>- Seal failure</td>
<td>- Seal leakage</td>
</tr>
<tr>
<td></td>
<td>- Worn mechanical seal</td>
<td>- Bearing wear</td>
</tr>
<tr>
<td></td>
<td>- Scored shaft sleeve</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Stuffing box improperly packed</td>
<td></td>
</tr>
<tr>
<td>External Leakage</td>
<td>- Seal failure</td>
<td>Depends on type of fluid and criticality as to time of failure</td>
</tr>
<tr>
<td></td>
<td>- Worn mechanical seal</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Scored shaft sleeve</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Stuffing box improperly packed</td>
<td></td>
</tr>
<tr>
<td>Mechanical noise</td>
<td>- Bent shaft</td>
<td>- Eventual wear of piston or rotor, and other components</td>
</tr>
<tr>
<td></td>
<td>- Worn/damaged bearing</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Foundation not rigid</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Cavitation</td>
<td></td>
</tr>
<tr>
<td>Positive suction head too low</td>
<td>- Clogged suction pipe</td>
<td>- Suction cavitation</td>
</tr>
<tr>
<td></td>
<td>- Valve on suction line only partially open</td>
<td>- Noisy operation</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Low discharge pressure</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- High output flow rate</td>
</tr>
<tr>
<td>Pump discharge head too high</td>
<td>- Clogged discharge pipe</td>
<td>- Discharge cavitation</td>
</tr>
<tr>
<td></td>
<td>- Discharge line valve only partially open</td>
<td>- Noisy operation</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Low output flow rate</td>
</tr>
<tr>
<td>Suction line clogged</td>
<td>- Contaminants</td>
<td>- No pump output / reduced flow</td>
</tr>
<tr>
<td>Pressure surges</td>
<td>Incorrect NPSH</td>
<td>- Cavitation damage</td>
</tr>
<tr>
<td>Increased fluid temperature</td>
<td>Incorrect fluid viscosity for pump</td>
<td>- Misaligned pump driver</td>
</tr>
</tbody>
</table>

10.2.1 Cavitation

The formation of bubbles and then the later collapse of these vapor bubbles due to the pump’s dynamic motion is the basic definition of cavitation. In order for cavitation to occur, the local pressure must be at or below the vapor pressure of the liquid. When a
fluid flows over a surface having a curvature, there is a tendency for the pressure near the surface to be lowered. There is a separation of fluid flow lines where there are different velocity regions. Between these fluid regions, turbulence can form which may cause bubbles to occur if the pressure is low enough. The collapsing of these bubbles can cause noise and vibrations. Sometimes, these pressure changes can be very dramatic and cause extensive damage to impellers, rotors, casings or shafts. If exposed for a sufficiently long time, pitting or severe erosion can occur. In some instances impeller vanes have experienced 3/8 inch of material loss. This type of damage can cause catastrophic failures.

Cavitation generally occurs in the first stage of a multistage centrifugal pump, although second stages have also been found to be effected when the suction head is substantially reduced. With displacement pumps like the rotary screw, cavitation can also occur. For these pumps it is important to understand the characteristics of entrained and dissolved air with respect to the vapor pressure of the fluid medium. The rotary screw pump shows a greater tendency for cavitation when the total available pressure at the pump inlet is below atmospheric pressure. With both displacement and centrifugal pumps, cavitation can be identified and easily remedied. Many times the inlet piping arrangement can be modified which will cause flow patterns that alleviate the problem.

10.2.2 Vortexing

Vortexing in centrifugal pumps is caused by insufficient fluid height above the suction line entrance or excess fluid velocity at the suction line entrance causing a noisy pump operation and loss of fluid flow. Vortexing of the fluid in a suction sump or pit sounds a lot like cavitation problems and will cause excessive shaft deflection and damage to mechanical seals, bearings and the pump intake structure and piping. Vortexing problems are intermittent as the vortices form as opposed to cavitation which once started tends to be a constant problem. There are several possible causes of a vortexing problem:

- The pump running at a faster speed than original design
- The flow or volume to the pump inlet has changed
- The fluids-solids mixer has changed
- The inlet line is restricted with contaminant solids
- Excess air in the liquid

10.2.3 Operating Environment

The effect of the ambient temperature and altitude on performance is normally independent of the type of pump. Limits for satisfactory performance are established primarily by the effect of the environment on the fluid rather than by the type of pumping action. Humidity only affects requirements for the pump casing. When operating
Temperature extremes are specified for a hydraulic system, the operating temperature of the fluid, not the ambient temperature, is the critical factor.

Minimum operating temperature is normally set by the increase in fluid viscosity as operating temperature is decreased. When the fluid viscosity is increased, to the point where inlet conditions can no longer keep the pump completely full, cavitation, with possible pump damage, occurs. Fire resistant fluids have a higher specific gravity than petroleum oils and higher viscosity at lower temperatures. They may also contain water which can vaporize at lower pressures or higher temperatures. Thus, pump inlet conditions are more sensitive when these fluids are used. High altitudes can produce similar effects when the fluid reservoir is not pressurized.

The pump being designed for specific fluids, failure rates of seals can increase if alternate fluids are used. Above allowable operating temperatures, many oils will be too thin to maintain proper lubrication at high-load points, and may progressively deteriorate as a result of oxidation. Under elevated temperatures, some seals may harden.

10.2.4 Interference

For rotary displacement pumps, the interference problem must be seriously addressed since very small distortions of rotors will decrease the clearance causing rubbing or direct impact between the moving parts of the rotary displacement pump. Thermal expansion can also pose a threat if there is no care taken in the proper selection of materials. Improper installation can also lead to interference problems. With centrifugal pumps, cavitation significantly increases the interference problem because cavitation causes vibration and imbalance. Interference can be avoided by designing the parts with appropriate elastic and thermal properties so that excessive load or temperature won't significantly deflect internal parts. Manufacturing tolerances must be carefully maintained.

10.2.5 Corrosion and Erosion

Consideration must be made for other possible failure modes such as erosion corrosion and intergranular corrosion. Erosion is dependent on the rate of liquid flow through the pump and also the angle of attack at which the fluid impinges on the material. Generally, the way in which materials should be selected is to first determine whether there are abrasive solids in the fluid. If there are, then the base material should be selected for abrasive wear resistance; if not, then the pump must be designed for velocity/corrosion resistance. Intergranular corrosion is the corrosion of the grain boundaries of the material. For austenitic stainless steels, intergranular corrosion can be limited by keeping the carbon content below 0.03 percent.

10.2.6 Material Fatigue

This failure mode, which cycles the material with unequal loadings over time, can be countered by good material selection. Material fatigue occurs with all types of pumps,
but may have more of an effect on displacement pumps, which have higher fluctuating stresses.

10.2.7 Bearing Failure

Although bearings are relatively inexpensive, they can cause costly shutdowns of complete systems. Short bearing life for centrifugal pumps, for example, can be caused by a number of problems including the following: misalignment, a bent shaft, a rotating part rubbing on a stationary part, a rotor out of balance causing vibration, excessive thrust caused by mechanical failure inside the pump, excessive bearing temperature caused by lack of lubrication, dirt or other contaminant in the fluid, excessive grease or oil in an anti-friction bearing housing, and rusting of bearings from water in housing.

Most bearing problems can be classified by the following failure modes: fatigue, wiping, overheating, corrosion, and wear. Fatigue occurs due to cyclic loads normal to the bearing surface. Wiping occurs as a result of insufficient lubrication film thickness and the resulting surface to surface contact. Loss of sufficient lubricant film thickness can occur from under-rotation or from system fluid losses. Overheating is shown by babbitt cracking or surface discoloration. Corrosion is frequently caused by the chemical reaction between the acids in the lubricants and the base metals in the babbitt. Lead based babbitts tend to show a higher rate of corrosion failures.

10.3 MODEL DEVELOPMENT

The impellers, rotors, shafts, and casings are the pump components which should generally have the longer lives when compared to bearings and seals. With good designs and proper material selection, the reliability of impellers, rotors, shafts and casings should remain very high. In order to properly determine total pump reliability, failure rate models have been developed for each pump component.

Pump assemblies are comprised of many component parts including seals, shaft, bearings, casing, and fluid driver. The fluid driver can be further broken down into the various types common to pumps including the two general categories for centrifugal and displacement pumps. For displacement pumps, it will be broken down into two further categories: reciprocating and rotary. For reciprocating pumps the fluid drivers can be classified as piston/plunger type or diaphragm type. For rotary pumps the fluid drive is a vane type for single rotors and for multiple rotors it is common to find a gear, lobe, or screw type of fluid driver. The total pump failure rate is a combination of the failure rates of the individual component parts. The failure rate for centrifugal pumps and displacement pumps can be estimated using equation (10-2).

\[
\lambda_P = \lambda_{SE} + \lambda_{SH} + \lambda_{BE} + \lambda_{CA} + \lambda_{FD}
\]  

\[(10-2)\]
Where:

\[ \lambda_P = \text{Total failure rate of the pump} \]

\[ \lambda_{SE} = \text{Total failure rate for all pump seals, failures/million operating hours} \quad \text{(See Chapter 3)} \]

\[ \lambda_{SH} = \text{Failure rate for the pump shaft, failures/million operating hours} \quad \text{(See Section 10.4 and Chapter 20)} \]

\[ \lambda_{BE} = \text{Total failure rate for all pump bearings, failures/million operating hours} \quad \text{(See Chapter 7)} \]

\[ \lambda_{CA} = \text{Failure rate for the pump casing, failures/million operating hours} \quad \text{(See Section 10.5)} \]

\[ \lambda_{FD} = \text{Failure rate for the pump fluid driver, failures/million operating hours} \quad \text{(See Section 10.6)} \]

### 10.4 FAILURE RATE MODEL FOR PUMP SHAFTS

A typical pump shaft assembly is shown in Figure 10.3. The reliability of the pump shaft itself is generally very high when compared to other components. Studies have shown (Reference 26) that the average failure rate for the shaft itself is about eight times less than mechanical seals and about three times less than that of the ball bearings. The possibility that the shaft itself will fracture, or become inoperable is very unlikely when compared to the more common pump failure modes. Usually the seals or bearings will cause problems first. The effect of the shaft on reliability of other components is of greater importance than the reliability of the shaft itself.

Because operational and maintenance costs tend to rise with increasing shaft deflection, new pump designs try to decrease possible shaft deflection. For centrifugal pumps, there is a large difference in deflection among the types of pump casing design. In a single volute casing, there are varying amounts of fluid pressure distributed about the casing causing unequal distributions of forces on the pump shaft. This imbalance causes shaft deflection and greater seal and bearing wear.

The amount of radial thrust will vary depending on the casing design and on the amount of the operating flow. The thrust load will increase from normal operation for any type of casing design when the pump is not run at its optimum flow rate speed. When the pump is not operating at its optimum rate, then the type of casing design will have a significant effect on the radial load.
The single volute type shows the greatest pressure imbalance and hence, the greatest deflection. Pump designers have learned to decrease this imbalance through different casing designs. The modified concentric casing and the double volute casing both have lower relative radial thrust because they cause a more even pressure distribution across the face of the impeller. The double volute is the most balanced and the design with the least amount of radial thrust. The maximum deflection recommended for a shaft design is approximately 0.001 inches.

The shaft diameter may be stepped up several times from the end of the coupling to its center to facilitate impeller mounting. Starting with the maximum diameter at the impeller mounting, there is a stepdown for the shaft sleeve and another for the external shaft nut, followed by several more for the bearings and the coupling. Therefore, the shaft diameter at the impeller exceeds that required for torsional strength at the coupling by at least an amount sufficient to provide all intervening stepdowns.

Chapter 20 provides the reliability model for pump shafts.

10.5 FAILURE RATE MODEL FOR CASINGS

The pump casing is a very reliable component. Defined as $\lambda_{Cd}$, the casing failure rate will have a greater effect on total pump reliability from the standpoint of how it affects other less reliable components. For instance, for an ANSI pump, the casing may have an average life expectancy of 10 years where a seal or bearing may have only one or two years. However, the type of casing used in the pump can have a large effect on the lifetime of the bearings and seals. This is due to differing loads placed on the pump...
shaft by the fluid flow pattern. The fluid flow patterns are a function of the casing design. The failure rate of the pump casing ($\lambda_{CA}$) itself can be estimated at 0.001 failures/million hours.

10.6 FAILURE RATE MODEL FOR FLUID DRIVER

All pumps require some vehicle to move the fluid from the intakes and expel it through the volutes and output ports to the exhaust opening. The means by which pumps do this is what differentiates most of today's numerous types of pumps. The reliability of these fluid drivers will vary from pump to pump. Impellers will wear out long after the seals. Pump gears for rotary gear pumps will have a lower reliability than impellers due to the nature of the contact between gears and the speed they attain.

Piston-plunger displacement pumps will generally have greater wear rates for the piston walls and rings than for the impellers of centrifugal pumps. The average failure rates in Table 10-4 have been determined from data base information developed from the Navy 3M system. The equations that describe the fluid driver wear rate may vary drastically since the fluid driver varies greatly in design and application. Other chapters of this Handbook can be used to estimate the failure rates for slider-crank mechanisms, mechanical couplings, valves and other components and parts unique to the particular pump design.

The failure rate of a pump fluid driver can be estimated from the following equation:

$$\lambda_{FD} = \lambda_{FD,B} \cdot C_{TLF} \cdot C_{PS} \cdot C_C$$

(10-3)

where:

- $\lambda_{FD}$ = Failure rate for the pump fluid driver, failures/million operating hours
- $\lambda_{FD,B}$ = Base failure rate of pump fluid driver (See Table 10-4)
- $C_{TLF}$ = Thrust Load Multiplying Factor (See Section 10.6.1)
- $C_{PS}$ = Operating Speed Multiplying Factor (See Section 10.6.2)
- $C_C$ = Contaminant Multiplying Factor (See Section 10.6.3)

10.6.1 Thrust Load Multiplying Factor

A centrifugal pump is designed to operate most reliably at one capacity for a given RPM and impeller diameter. This flow rate is called the best efficiency point (BEP). At this flow, hydraulic loads imposed on the impeller are minimized and are steady. At flows greater than or less than the BEP the hydraulic loads increase in intensity and become unsteady due to turbulence in the casing and impeller. As a result, hydraulic loads, which are transmitted to the shaft and bearings, increase and become unsteady.
Shaft deflection changes as a function of the fluid flow rate through the pump. As a pump's capacity increases or decreases, moving away from the point of maximum efficiency, fluid pressures around the impeller become unequal, tending to deflect it. Special casings, such as diffusers and double-volute and concentric casings can greatly reduce the radial thrust and, hence, the deflection.

Also, the severity of these unsteady loads can cause failures of the mechanical seal. Operation at reduced flow rates that put the pump into its recirculation mode can also lead to cavitation damage in high suction energy pumps. The effect on reliability of operating a pump too far from its maximum efficiency point is shown in Figure 10.4.

The thrust load multiplying factor, \( C_{TLF} \), is dependent upon the casing type and pump capacity percentage. The pump capacity percentage is the actual operating flow divided by the maximum pump specification flow. Values for \( C_{TLF} \) are shown in Figure 10.4 and related equations are included in Table 10-3.

### 10.6.2 Operating Speed Multiplying Factor

Operating speed affects the failure rate multiplying factor, caused by the increased RPM leading to accelerated rubbing wear of shaft and mechanical seal faces, increased bearing friction and lubricant degradation. Increased operating speed also increases the energy level of the pump which can lead to cavitation damage. The effects of wear on these components are almost linear as a function of RPM. Equation 10-4 provides a multiplying factor for operating speed based on actual and design RPM See Figure 10.5.

\[
C_{PS} = k \cdot \left( \frac{V_O}{V_D} \right)^{1.3}
\]  
\[\text{(10-4)}\]

Where:
- \( V_O \) = Operating Speed
- \( V_D \) = Maximum Allowable Design Speed
- \( k \) = Constant = 5.00

### 10.6.3 Contaminant Multiplying Factor

The contaminant factor, \( C_C \), was developed from research performed for the Naval Air Warfare Center in Warminster, Pennsylvania on the effect of contamination and filtration level on pump wear and performance. The contamination factor equation is based on the filtration level as follows:

\[
C_C = 0.6 + 0.05 F_{AC}
\]  
\[\text{(10-5)}\]
Where: \( F_{AC} \) = Filter size, microns

\[ Q = \text{Actual operating pump flow, gpm} \]
\[ Q_r = \text{Specified maximum pump flow, gpm} \]

(See equations in Table 10-3)

**Figure 10.4 Thrust Load Multiplying Factor as a Function of Pump Capacity and Casing Design**
\[ C_{PS} = k \cdot \left( \frac{V_O}{V_D} \right)^{1.3} \]

Where:

- \( V_O \) = Operating speed
- \( V_D \) = Maximum allowable design speed
- \( k \) = Constant = 5.00
Table 10-3. Equations for Figure 10.4, Thrust Load Multiplying Factor, $C_{TLF}$

Ordinary volute casings:

For $0.1 \leq \frac{Q}{Q_r} \leq 1.0$:

$$C_{TLF} = 9.94 - 0.90 \left( \frac{Q}{Q_r} \right) - 10.00 \left( \frac{Q}{Q_r} \right)^2 + 1.77 \left( \frac{Q}{Q_r} \right)^2$$

Where: $C_{TLF} =$ Thrust load multiplying factor  
$Q =$ Actual operating flow, gpm  
$Q_r =$ Maximum pump specified flow, gpm

For $1.0 < \frac{Q}{Q_r} < 1.1$:

$$C_{TLF} = 1.0$$

For $1.1 \leq \frac{Q}{Q_r} \leq 1.7$:

$$C_{TLF} = -30.60 + 36.00 \left( \frac{Q}{Q_r} \right) - 4.50 \left( \frac{Q}{Q_r} \right)^2 - 2.20 \left( \frac{Q}{Q_r} \right)^3$$

Modified volute casings:

For $0.1 \leq \frac{Q}{Q_r} \leq 1.7$:

$$C_{TLF} = 5.31 - 0.55 \left( \frac{Q}{Q_r} \right) - 12.00 \left( \frac{Q}{Q_r} \right)^2 + 12.60 \left( \frac{Q}{Q_r} \right)^3 - 4.63 \left( \frac{Q}{Q_r} \right)^4 + 0.68 \left( \frac{Q}{Q_r} \right)^5$$

Double volute casings:

$$C_{TLF} = 1.03 - 0.30 \left( \frac{Q}{Q_r} \right) + 0.04 \left( \frac{Q}{Q_r} \right)^2$$

Displacement Pumps:

$$C_{TLF} = 0.80 + 1.1 \left( \frac{Q}{Q_r} \right)$$
Table 10-4. Base Failure Rates for Pump Fluid Drivers ($\lambda_{FD,B}$)

<table>
<thead>
<tr>
<th>PUMP TYPE</th>
<th>FLUID DRIVER MODE</th>
<th>MODEL TYPE</th>
<th>BASE RATE*</th>
</tr>
</thead>
<tbody>
<tr>
<td>Centrifugal</td>
<td>- Axial flow impeller</td>
<td>- Closed / open impellers</td>
<td>0.20</td>
</tr>
<tr>
<td></td>
<td>- Mixed flow / radial flow impeller</td>
<td>- Open / semi-open / closed impellers</td>
<td>0.12</td>
</tr>
<tr>
<td></td>
<td>- Peripheral</td>
<td>- Single stage / multi-stage</td>
<td>0.20</td>
</tr>
<tr>
<td>Displacement</td>
<td>- Reciprocating</td>
<td>- Piston / plunger</td>
<td>1.18</td>
</tr>
<tr>
<td></td>
<td>- Reciprocating</td>
<td>- Diaphragm</td>
<td>0.58</td>
</tr>
<tr>
<td></td>
<td>- Rotary (single rotor)</td>
<td>- Vane</td>
<td>0.4</td>
</tr>
<tr>
<td></td>
<td>- Rotary (single rotor)</td>
<td>- Piston</td>
<td>1.05</td>
</tr>
<tr>
<td></td>
<td>- Rotary (multiple rotor)</td>
<td>- Gear</td>
<td>0.75</td>
</tr>
<tr>
<td></td>
<td>- Rotary (multiple rotor)</td>
<td>- Lobe</td>
<td>0.45</td>
</tr>
<tr>
<td></td>
<td>- Rotary (multiple rotor)</td>
<td>- Screw</td>
<td>0.58</td>
</tr>
</tbody>
</table>

* Failures/million hours of operation

10.7 REFERENCES


86. Allan Budris, Eugine Subini, R. Barry Erickson, "Pump Reliability – Correct Hydraulic Selection Minimizes Unscheduled Maintenance”, PumpLines, Fall 2001
11.1 INTRODUCTION

Fluid filtration equipment is unique in that the reliability of this equipment is more concerned with the effects of the filter on associated equipment than on the lifetime of the filter itself. This is due to severe wear of fluid system components which can occur when these components are operated with poorly filtered fluid. This chapter will review the conditions which can lead to degradation or failure of the filter. The effect of contamination on the wear of various components is also discussed. A basic failure rate model with correction factors will also be developed.

11.1.1 Filtration Mechanisms

Filters consist of a porous filter media through which fluid is passed. The filter media is typically corrugated to increase the amount of filtration area in the filter volume. Filtration of gases is accomplished by absorption and direct interception of the suspended particles. Filtration of liquids is primarily accomplished by direct interception of the suspended particles. A typical pressure line filter is shown in Figure 11-1.
11.1.2 **Service Life**

The porous structure of a filter media presents a resistance to fluid flow which causes a pressure drop across the filter. This filter differential pressure increases as captured particles or contaminants are collected and plug the porous media. Every system has a maximum differential pressure at which the filter must be cleaned or replaced. The filter service life is the time it takes the filter to reach the maximum allowable differential pressure. Use of the filter beyond its service life could result in catastrophic failure of the filter due to the high differential pressure or it could result in unfiltered fluid bypassing the filter (Reference 33).

11.1.3 **Filter Failure**

A filter is considered to have failed when it releases previously captured contaminants, when it allows unfiltered fluid to pass throughout the filter media, or when the filter collapses and contaminates the fluid with filter media. Plugging of the filter with contaminants, with a resulting increase in filter differential pressure, can be a normal consequence of operation. Failure due to premature plugging can occur for several reasons as described in Section 11.2. Failure of the filter can also be caused by operating conditions such as high differential pressures, cyclic flow, vibration, system startups when cold, and even by the fluid being filtered, if the fluid is incompatible with the filter. Typically, 67% of filter failures are due to leakage and 33% are due to clogging.
11.2 FILTER FAILURE MODES

Typical failure modes and causes for filter assemblies are summarized in Table 11-1. Some of the more common failure modes are described as follows:

Channeling: Excessively high differential pressures can cause filter media pores to enlarge, allowing large amounts of unfiltered fluid to bypass the filter media. Enlargement of the media pores also allows previously captured contaminants to be released. Channeling can also be the result of media fatigue caused by cyclic flow conditions.

Fatigue Cracks: Cyclic flow conditions in the fluid system can cause fatigue cracks in the filter media. Such cracks may occur at the roots of pleats in corrugated filters or within the volume of loose packed media. The cracks will allow the release of contaminants from the filter and will allow some of the fluid to bypass the filter. Media fatigue can result from cyclic flow conditions such as varying system flow requirements, pump ripple, or cold system startups.

Media Migration: Improper bonding of the media fibers or deterioration of the bonding can result in the downstream release of media fibers. This downstream release of the filter media is termed media migration. Media migration during vibration of the filter may result from an improper fit of the filter in the filter housing or may result from the filter media abrading against the filter casing. Media migration can also occur in conjunction with fatigue cracks in the media, as caused by cyclic flow conditions. Media migration can also occur during cold temperature start-ups due to potential large differential pressure generated as a consequence of increased fluid viscosity.

Filter Disintegration: Complete disintegration of the filter can occur as a result of extremely high differential pressures. Disintegration can also be the result of embrittlement of the filter media from exposure to incompatible fluids or cold temperatures.

Plugging: Plugging of the filter media can be either a normal consequence of operation or failure, depending upon when plugging occurs. Failure due to premature plugging can be attributed to several causes other than just the accumulation of wear debris. As an example, Hudgens and Feldhaus (Reference 24) have found that lubrication oil filters in diesel engines can plug by any one of six mechanisms. While some of these are particular to internal combustion engines, the mechanisms may be applicable to any oil-based fluid system. The six mechanisms can be summarized as follows:

(1) Absorption of water in the oil from condensed moisture and/or coolant leakage can cause insoluble contaminants, normally dispersed into the lubricating oil, to dump out of suspension. This condition can also arise when there is a combination of moderate soot load, low pH and a high level of oxidation produced in the oil. A filter plugged under these circumstances will be marked by a sticky, shiny, adherent sludge.
with wavy pleats and the filter will have accumulated from 1/3 to 1/2 of its total contaminant capacity.

(2) Saturation of the oil with excessive amounts of combustion contaminants, due to engine problems or overextended oil drain internals, can also cause filter plugging. The filter will appear to have a thick, loosely-held sludge. The filter will have accumulated from 1/3 to 1/2 of its total contaminant capacity typically but it can accumulate up to 100 percent in extreme cases.

(3) Absorption of oxidation products such as degraded fuel and oil will also cause the filter to plug and the filter will have accumulated 40 percent to 50 percent of its contaminant capacity. The filter does not appear to have sludge buildup but it does appear to have a brown tint and to be covered by blown snow. The problem occurs most often with API CC spec lubricating oils where overheating or fuel dilution is a problem.

(4) Moisture condensation or coolant leakage into the oil reservoir can cause filters to plug as a result of oil additive precipitation. Plugging of the filter can occur at 8 percent to 30 percent of the filter's contaminant capability. The filter will have a gray coloration but no sludge build-up.

(5) Coolant or moisture can also combine with oil additives to form thick, filter-plugging gels. The filter media in such circumstances will be wavy with a sticky feel but will usually look clean. Filters plugged due to gel formation usually reach only 3 percent to 6 percent of their contaminant capacity before plugging.

(6) Accumulation of wear debris also causes filters to plug. In this failure mechanism, the filter plugs by retention of 100 percent or more of its full contaminant capacity. The filter will appear to have a buildup of visible wear particles on the filter media.

11.3 FLUID CONTAMINATION EFFECTS

Fluid system component failures related to particulate contamination of the operating fluid are usually either catastrophic or deterioration failures. Catastrophic failures occur when the system components are operated under intolerable conditions. Catastrophic failures may also be the result of wear occurring over a long period of operation. Failures due to component deterioration typically involve a fairly rapid change in component performance, falling below a satisfactory level after a period of normal operation.

Contamination of the operating fluid with hard particles can cause progressive performance deterioration through an abrasive wear mechanism. This type of wear is characterized by a particle penetrating a softer surface and cutting away material. The rate of wear and thus the rate of performance degradation is dependent on the number
of particles and the particle hardness. Particle contamination can also cause cumulative performance degradation where a rapid decline in performance follows an extended period of apparently normal operations. This type of degradation failure is caused by the creation of surface defects during operation. These surface defects may be caused by abrasion, surface fatigue or adhesion wear processes.

### Table 11-1. Failure Modes of Filters

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECTS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channeling</td>
<td>- High differential pressures</td>
<td>- Release of contaminants</td>
</tr>
<tr>
<td></td>
<td>- Cyclic flow</td>
<td>- Circulation of unfiltered fluid</td>
</tr>
<tr>
<td>Fatigue cracks</td>
<td>- Cyclic flow</td>
<td>- Circulation of unfiltered fluid</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Release of filter media</td>
</tr>
<tr>
<td>Media Migration</td>
<td>- Vibration</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Cyclic flow</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Cold starts</td>
<td></td>
</tr>
<tr>
<td>Filter disintegration</td>
<td>- Embrittlement</td>
<td>- Substantial contamination of fluid with filter media</td>
</tr>
<tr>
<td></td>
<td>- Cold starts</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- High differential pressures</td>
<td></td>
</tr>
<tr>
<td>Plugging</td>
<td>- Condensed moisture</td>
<td>- Increase in filter differential pressure, definition of failure dependent on</td>
</tr>
<tr>
<td></td>
<td>- Oil saturated with contaminants</td>
<td>system design and maintenance schedule</td>
</tr>
<tr>
<td></td>
<td>- Absorption of oxidation products</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Absorption of coolant</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Accumulation of wear debris</td>
<td></td>
</tr>
</tbody>
</table>

Fluid systems requiring filtration typically include components such as pumps, gears, control valves, ball bearings, roller bearings, journal bearings, and seals. Their potential effect on contamination of the fluid system can be described as follows ([Reference 7](#)):  

**Pumps:** In displacement-type piston pumps, the piston face can be damaged by cavitation or corrosion. Contaminant particles can enter the lubricant film between the piston and cylinder and plough the surface several times before being ejected. In swashplate controlled devices, such as variable displacement pumps and hydraulic motors, the piston shoes can cause abrasion-wear-type degradation failures as the
shoes are highly loaded and are in sliding contact with the swashplate. Similar abrasive-wear-type degradation failures can occur to the sliding contact surfaces of the rotating cylinder block and the mating valve pressure plates.

**Gears:** Gear failures are primarily failures of the gear tooth surface. This surface is damaged by rubbing wear, scoring, pitting, and plastic flow. Rubbing wear occurs when the lubricant film is insufficient to separate the tooth surfaces and is generated by both adhesive and abrasive wear mechanisms. Scoring of the tooth surface is generated by the adhesive wear type mechanisms under intense local frictional heating. Pitting and plastic flow both occur as a result of tooth surface fatigue wear.

**Valves:** Particle contamination can cause increased leakage in control valves by severe cutting or by milder abrasive wear mechanisms. Synthetic phosphate ester fluids have been found to cause servo valve erosion by a streaming-potential corrosion process. A brittle corrosion layer is formed on the valve and is abraded by fluid-borne particles, adding additional particulates to the fluid and exposing base metal, allowing further corrosion. Deterioration failures of relief valves can occur from particle contamination caused by erosion. Contamination of hydraulic fluid by water has been shown to cause rust inhibitor additive to attach to servo valve spools and prevent movement of the valve spool within its housing.

**Bearings:** Hard particle contamination of ball and roller bearing lubricants is the cause of two types of abrasive wear of the rolling surfaces. Hard particle contamination causes rolling surface damage that dominates the fatigue life of ball bearings under typical operating conditions. In severe circumstances, hard particle contamination causes indentations and pits which cause rapid failure of the rolling surfaces. Abrasive wear, increasing with particle concentration and hardness, can remove material from the sliding edges of a tapered rolling bearing, reducing the bearing width and allowing increased misalignment. Wear of this type does not stop until the contaminant size is reduced to less than the lubricant film thickness.

The performance of new journal bearings improves with use initially due to better surface conformity caused by wear during boundary lubrication conditions. As wear in the contact region progresses, the performance begins to gradually deteriorate. Wear of the journal bearings is caused by both abrasive and adhesive wear due to the sliding motion in the contact region. Contamination of the lubricant with water can cause the formation of a metal oxide boundary layer on the bearing which can inhibit adhesive wear. However, abrasion of this film can cause bearing failure due to rapid increases in wear, bearing corrosion, and the number of abrasive oxide particles. Maximum bearing life can be achieved by selecting a filter to filter out all particles larger than the minimum lubricating film thickness.

**Seals:** Seal failures are typically caused by fatigue-like surface embrittlement, abrasive removal of material, and corrosion. Elastomeric seals are more sensitive to thermal deterioration than to mechanical wear. However, hard particles can become
embedded in soft elastomeric materials and sliding contact metal surfaces, causing leakage by abrasive wear of the harder mating surfaces. Abrasive particles can also plug lubricant passages which causes seal failure from the lack of lubricant. Seal failures can be reduced by reducing the amount of contamination through filtration and concern for the operating environment.

11.4 FILTER RELIABILITY MODEL

A basic filter reliability model can be developed by modeling the fluid system incorporating the filter. By modeling the flow of particulates through the system, an expression for the rate of retention of particulates by the filter may be developed. This expression, a function of the system and filter parameters, can then be integrated to a form which can be used to calculate the mass of particulates stored within the filter at any time, \( t \). The amount of stored mass at any time, \( t \), can be compared with the filter capacity, typically a known parameter, to determine the filter reliability.

In order to simplify the development of an initial filter model, the following assumptions must be made:

1. The rate of generation of contaminate particulates by system components and the rate of ingestion of environmental contaminates do not vary with time and the particulates are evenly mixed within the system fluid. Furthermore, the rates of generation of contaminant particulates by system components may be modeled using Table 3-4 in Chapter 3.

2. The system fluid volume and flow rates do not change with time.

3. The system fluid volume may be represented as one lumped sum so that the individual components and fluid lines need not be modeled.

4. The filter will not plug or become unusually restricted prior to reaching its maximum capacity.

A typical hydraulic system consists of a reservoir, pump, filter, one or more control valves, and one or more fluid motors. Such a system can be simplified using the above assumptions to resemble the diagram in Figure 11.2.

The complete filter failure rate model should use adjustments, or correction factors, to modify the base failure and to account for potentially degrading effects of off-design operating conditions. Considering the causes of failures as earlier discussed, the failure rate model can be written as:

\[
\lambda_F = \lambda_{F,B} \cdot C_D \cdot C_V \cdot C_{CS}
\]  

(11-1)
Where: \( \lambda_F \) = Failure rate of the filter in failures/million hours

\( \lambda_{F,B} \) = Base failure rate of the filter in failures/million hours (See Section 11.4.1)

\( C_{DP} \) = Multiplying factor which considers the effects of the filter differential pressure on the base failure rate (See Section 11.4.2)

\( C_V \) = Multiplying factor which considers the effects of vibration on the base failure rate (See Section 11.4.3)

\( C_{CS} \) = Multiplying factor which considers the effects of cold start-up conditions on the base failure rate (See Section 11.4.4)

![Figure 11.2 Simplified Fluid System Containing Filter](image)

### 11.4.1 Base Failure Rate

Using a diagram similar to that of Figure 11.2, Hubert, Beck and Johnson (Reference 23) developed an expression for the concentration of contaminant particulates upstream of the system filter at any time, \( t \), as a function of system fluid...
volume, flow rate, filter efficiency and total contaminant ingestion rates. This equation is written as follows:

\[ C_u(t) = \left(C_o - \frac{M_{ci}}{\varepsilon Q}\right)e^{-\frac{\varepsilon Q t}{V}} + \frac{M_{ci}}{\varepsilon Q} \]  

(11-2)

Where:

- \( C_u(t) \) = Concentration of contaminant particulates upstream of the system filter at any time, \( t \), \( \mu g/ml \)
- \( C_o \) = Initial concentration of contaminant particulates, \( \mu g/ml \)
- \( M_{ci} \) = Generation rate of contaminant particulates from all sources and of all sizes, \( \mu g/min \)
- \( \varepsilon \) = Overall filter efficiency
- \( Q \) = Volumetric fluid flow rate through filter, \( ml/min \)
- \( V \) = Volume of fluid, \( ml \)
- \( t \) = Time, \( min \)

The concentration of contaminant particulates downstream of the filter can be calculated knowing the filter efficiency and the concentration upstream:

\[ C_d(t) = (1 - \varepsilon) C_u(t) \]  

(11-3)

The mass of contaminant particles retained by the filter over a time period, \( t \), is then:

\[ M_{filter} = \int_0^t \left[ C_u(t) - C_d(t) \right] Q \, dt \]  

(11-4)

The mass of contaminant particles retained by the filter determines the life of the filter. However, by definition, a clogged filter is not normally classified as a component failure. See Table 11-1 for a list of typical failure modes. A typical value for a base failure rate is given as 2.53 failures per million operating hours. Typically, the failure rate distribution will be 67% of the failures due to internal leakage and 33% of the failures due to plugging. For a more detailed analysis, the following paragraphs describe the multiplying factors to be used in conjunction with this base failure rate.
\[ \lambda_{F,B} = \text{Base failure rate of a filter in normal operation} \quad (11-5) \]

\[ = 2.53 \text{ failures/million hours} \]

### 11.4.2 Filter Differential Pressure Multiplying Factor

Assuming that the filter may be modeled as a thick-walled cylinder, the correction factor for filter differential pressure \( C_{DP} \) may be developed from the following equation for radial stress (Reference 39).

\[
\sigma_r = \frac{P_i a^2 - P_o b^2 + \frac{a^2 b^2 (P_o - P_i)}{r^2}}{b^2 - a^2} \quad (11-6)
\]

Where:
- \( \sigma_r \) = Radial stress, lbs/in\(^2\)
- \( a \) = Inside radius, in
- \( b \) = Outside radius, in
- \( P_i \) = Inside design pressure, psi
- \( P_o \) = Outside design pressure, psi
- \( r \) = Radius corresponding to maximum stress, in

For most filters, the equation for radial stress can be used to model the effects of high differential pressure on the filter media by developing a ratio of operating stress to design strength. By indicating the operating inside and outside pressures by \( P_i' \) and \( P_o' \), and by dividing the corresponding operating stress by the design strength, the following equation for \( C_{DP} \) can be derived:

\[
C_{DP} = \frac{P_i' a^2 - P_o' b^2 + \frac{a^2 b^2 (P_o' - P_i')}{r^2}}{P_i a^2 - P_o b^2 + \frac{a^2 b^2 (P_o - P_i)}{r^2}} \quad (11-7)
\]
where: \( P'_i \) = operating inside pressure, psi
\( P'_o \) = operating outside pressure, psi

In most filter installations, the flow of fluid through the filter is from the outside to the inside. In this case, the maximum stress will be found at the outer radius, i.e., \( r = b \) and \( P_i = 0 \). Substitution of this into Equation (11-7) results in:

\[
C_{DP} = \frac{P'_o}{P_o}
\]

where: \( P_o \) = outside design pressure

Cyclic flow, pressure surges, and pump ripple can also have a significant impact on filter lifetimes. The tensile strength of the filter media needs to be considered in relation to these parameters.

11.4.3 Vibration Multiplying Factor

Most filters are tested for media migration caused by vibration. A typical test is performed with the filter immersed in the system fluid and the filter is exposed to low amplitude, high frequency vibrations for about 100,000 cycles. As a result, most filters will not degrade due to vibration. However, in aircraft environments, failure of the filter housing and seals due to vibration accounts for 80 percent of the total filter failure rate (Reference 15). Thus it appears that in most systems vibration is not a problem, but in aircraft systems excessive vibration can cause filter failure. As a result:

\[
C_V = 1.25 \text{ for aircraft and mobile systems}
\]

\[
C_V = 1.00 \text{ for all other systems}
\]

11.4.4 Cold Start Multiplying Factor

The correction factor for cold start degradation can be calculated using a ratio of the cold start fluid viscosity to the normal operating fluid viscosity. This is:
\[ C_{CS} = \left( \frac{\nu_{\text{cold start}}}{\nu_{\text{normal}}} \right)^x \]  

(11-9)

Where:

\( \nu_{\text{cold start}} \) = Viscosity at cold start temperature, lb-min/in\(^2\)

\( \nu_{\text{normal}} \) = Viscosity at normal operating conditions, lb-min/in\(^2\)

\( x \) = Exponent which varies with type of fluid

Values for \( \nu \) and \( x \) can be obtained from Table 11-2. Values for \( x \) will range from 0.20 for light viscous fluids such as kerosene to 1.1 for heavy viscous fluids such as SAE 70 oil.
<table>
<thead>
<tr>
<th>Liquid</th>
<th>Viscosity in Centistokes, ν</th>
<th>X</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0 C</td>
<td>20 C</td>
</tr>
<tr>
<td>Water</td>
<td>1.8</td>
<td>1.0</td>
</tr>
<tr>
<td>Sea water</td>
<td>1.9</td>
<td>1.1</td>
</tr>
<tr>
<td>Gasoline, 0.68 s.g.</td>
<td>0.51</td>
<td>0.42</td>
</tr>
<tr>
<td>Kerosene, 0.81 s.g.</td>
<td>3.7</td>
<td>2.3</td>
</tr>
<tr>
<td>Light lubricating oil, 0.91 s.g.</td>
<td>390</td>
<td>96</td>
</tr>
<tr>
<td>Heavy lubricating oil, 0.91 s.g.</td>
<td>3492</td>
<td>500</td>
</tr>
<tr>
<td>SAE 10 oil</td>
<td>555</td>
<td>122</td>
</tr>
<tr>
<td>SAE 20 oil</td>
<td>1141</td>
<td>213</td>
</tr>
<tr>
<td>SAE 30 oil</td>
<td>2282</td>
<td>358</td>
</tr>
<tr>
<td>SAE 40 oil</td>
<td>4640</td>
<td>624</td>
</tr>
<tr>
<td>SAE 50 oil</td>
<td>8368</td>
<td>1179</td>
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<tr>
<td>SAE 60 oil</td>
<td>15215</td>
<td>2206</td>
</tr>
<tr>
<td>SAE 70 oil</td>
<td>23203</td>
<td>2853</td>
</tr>
</tbody>
</table>
11.5 REFERENCES


12.1 INTRODUCTION

The principal function of a brake or clutch assembly is to convert kinetic energy to heat and then either to absorb or dissipate heat while simultaneously (through energy transfer) reducing the relative movement between the friction material and the part to which it is engaged. Reliability models for brakes and clutches are presented together in this Handbook because of similar design and operational characteristics; and because one of the most important functional parts of each of these components is the friction material. Section 12.2 addresses the brake model, which includes actuators, springs, friction linings, bearings, seals and housings. An analysis of the energy
transfer materials which are common to both brakes and clutches is included in the brake model. Section 12.3 outlines and describes the reliability model for clutches, which includes the following components: actuators, bearings, friction linings, seals and springs.

12.2 BRAKES

12.2.1 Brake Assemblies

The reliability of a brake system is dependent on the reliability of its component parts, which may include: actuators, bearings, friction linings, housings, seals, and springs. With the exception of friction linings, all of these component parts are addressed in earlier sections of this handbook. The characteristics of these parts that are peculiar to the braking environment will be discussed in this chapter. The reliability of brake and clutch components is sensitive to friction materials used in their assembly. Therefore, an in-depth analysis of these mechanical parts is presented in this chapter.

In brake systems the rubbing elements include the friction material and a counter-surface. The friction material is the sacrificial element, although the essence of good brake design is to minimize wear. The counter-surface is usually metallic, to provide structure and to dissipate the frictional heat. Most counter-surfaces are a cast iron drum or disc. In a few applications, steel rubbing surfaces are used. The counter-surface is also nominally a non-wearing surface. Counter-surfaces typically wear from one to twenty percent of the total volume worn from the friction interface (Reference 16).

Brakes are called upon to convert large amounts of kinetic energy to thermal energy in a very short time. The life of currently used brake lining materials is determined by wear, which in turn is strongly dependent on the temperature experienced by these materials during sliding. This temperature dependence is due largely to softening of the metal binder (usually copper or iron) present in brake lining composite materials.

Some of the systems which use brakes include passenger cars, light trucks, tractors, buses, agricultural equipment, construction equipment, industrial equipment, railroad trains and aircraft. Brake lining materials used in passenger cars and light trucks fall into two categories: drum brake segments, which are less than 3/4" thick, and disk brake pads.

Brake systems used by trucks, truck tractors and trailer combinations are air assisted hydraulic (air brake) systems. The braking systems used by buses are similar to the conventional air brake system used by large trucks.

Agricultural, construction and industrial equipment each have different brake requirements. Agricultural equipment includes all equipment used in farming and forestry, such as tractors, harvesters, and log skidders. Construction equipment is used for the construction of roads, homes and buildings and includes wheeled tractors,
rollers, scrapers, dozers, power truck cranes, hoists and shovel loaders. Industrial equipment encompasses all equipment used in fixed facility or buildings such as overhead cranes or hoists. Hydraulic brake systems used in agricultural and construction equipment are of either the dry or the wet brake type. Dry brakes are the conventional types of drum or disk system. Wet brakes use drum and disk brake assemblies but the friction material is in a fluid environment. This type of brake exhibits decreased heat build up and subsequently less fade, reduced lining and drum or rotor wear and improved reliability.

Industrial equipment normally uses the conventional drum brake systems with organic binder/asbestos linings. In industrial equipment, such as cranes and hoists, wet brake systems are not used. As a result, an improved friction material with longer wear is needed in such systems. One of the major costs for overhead cranes in industrial use is lining maintenance. Lining replacement is required every three to four weeks.

Most railroad trains rely on two braking systems, a dynamic brake and a friction brake. Most self-propelled rail cars have a dynamic brake, which is used either independently or together with the train’s friction braking system down to about 5-10 mph, using complete friction braking for the last distance to a complete stop.

The use of organic friction materials in aircraft brakes is currently limited primarily to small general aviation aircraft. The trend in larger aircraft brake materials has been toward higher energy absorption per unit mass of brake materials. On larger aircraft organic friction materials have been replaced by more expensive copper and iron-based metallics. Disk brakes, with one brake for each of the main landing gears is common.

12.2.2 Brake Varieties

There are numerous brake system types, each with its own parts and reliability characteristics:

**Band Brakes** - Simpler and less expensive than most other braking devices. Component parts include friction band element and the actuation levers. Characterized by uneven lining wear and poor heat dissipation.

**Externally and Internally Pivoted Drum Brakes** - Simple design requiring relatively little maintenance. May become self-locking with extreme wear if not properly designed. Component parts include friction materials, springs, actuators, housings, seals, and bearings. Internal types offer more protection from foreign material.

**Linearly Acting External and Internal Drum Brakes** - These brakes are fitted with shoes that, when activated, approach the drum by moving parallel to a radius through the center of the shoe. Springs between the friction materials may separate both shoes when the brake is released. Lining wear is more uniform in comparison with internal drum brakes. Component parts include friction materials, springs, actuators, seals, housings, and bearings.
Dry and Wet Disk Brakes - Disk brakes have two main advantages over drum brakes: better heat dissipation and more uniform braking action. However, disk brakes require a larger actuation force due to the absence of either a friction moment or servo action. Both annular and pad type disk brakes are modeled here, and include friction materials, springs, actuators, housings, seals, and bearings. Wet disk brakes may operate in an oil bath. Thus, these brakes are isolated from dirt and water, and the circulation of the oil through a heat exchanger usually provides greater heat dissipation than direct air cooling.

Magnetic Particle, Hysteresis, and Eddy-Current Brakes - In all three of these brake types the braking torque is developed from electromagnetic reactions rather than mechanical friction, and therefore requires a source of electrical power.

The various types of brake systems and methods of actuation are listed in Table 12-1. There are numerous brake lining materials, manufacturing processes, brake types and systems in use today.

Table 12-1. Methods of Actuation (Reference 32)

<table>
<thead>
<tr>
<th>TYPE</th>
<th>ADVANTAGES</th>
<th>DISADVANTAGES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mechanical</td>
<td>- Robust, simple operation provides good control</td>
<td>- Large leverage needed</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Potential frictional losses at pins and pivots</td>
</tr>
<tr>
<td>Pneumatic</td>
<td>- Large forces available</td>
<td>- Compressed air supply needed</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Brake chambers may be bulky</td>
</tr>
<tr>
<td>Hydraulic</td>
<td>- Compact</td>
<td>- Special fluid needed</td>
</tr>
<tr>
<td></td>
<td>- Large forces available</td>
<td>- Temperatures must not be high</td>
</tr>
<tr>
<td></td>
<td>- Quick response and good control</td>
<td>- enough to vaporize fluid</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Potential seal problems</td>
</tr>
<tr>
<td>Electrical</td>
<td>- Suitable for automatic control</td>
<td>- On-off operations</td>
</tr>
<tr>
<td></td>
<td>- Quick response</td>
<td></td>
</tr>
</tbody>
</table>

For example, there are at least six basic methods of making brake linings: Dry process, extruded process, wet board process, sheeter process, sintered metal process and woven process. An analysis of typical linings indicates many common constituents. Chrysotile asbestos is found in most linings at roughly 50 percent (by weight). Rubber, resin, or a combination of both is used as lining binders. Brake lining
fillers and friction modifiers include many metals, metallic compounds, graphite, coal rubber and resins. The specific choice of such materials results from controlled test type experimentation in the development of a friction material to meet specific performance goals. In addition to actual vehicle testing of a brake lining material, the industry uses several dynamometer laboratory test machines to characterize friction materials (Reference 42).

**Friction Material Test Machine** - This apparatus attempts to record the brake lining performance by subjecting it to controlled conditions of pressure and temperature.

**Friction Assessment Screening Test Machine** - The rate of energy dissipation is controlled on a disk while temperature gradually increases.

**Single End Inertial Dynamometer** - An actual brake mechanism is incorporated.

**Dual End Inertial Dynamometer** - This uses the same instrumentation as the single end, but it is operated with one front and one rear brake assembly.

12.2.3 **Failure Modes of Brake Assemblies**

A list of failure modes for a typical brake system is shown in Table 12-2. The brake system friction materials are sacrificial replacements, and they account for most of the "failures". Because friction linings are designed to wear out before the life of the vehicle, service life may be a better measure of their durability than failure rate.

For the purpose of compatibility with the other models developed for mechanical components, the lining life will be converted to a rate of failure. Use of the brake system beyond the life of the friction material results in catastrophic failure of the brake system caused by a loss of braking force due to a drastic reduction in coefficient of friction. Descriptions of the counter-surface failure characteristics are included in Table 12-3.

12.2.4 **Brake Model Development**

The brake system will be reduced to its component parts. Brake systems will normally contain some combination of the following components:

- Actuators
- Springs
- Brake friction linings
- Bearings
- Seals
- Housings
Table 12-2. Failure Modes of a Mechanical Brake System
(Reference 9)

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sticking piston</td>
<td>Contamination</td>
<td>Low output pressure</td>
</tr>
<tr>
<td>Leaking cylinder</td>
<td>Contamination</td>
<td>Low output pressure</td>
</tr>
<tr>
<td>Broken/weak spring</td>
<td>Fatigue activation</td>
<td>Unable to adjust pressure</td>
</tr>
<tr>
<td>Sticking bleeder valve</td>
<td>Contamination</td>
<td>Inadequate dissipation of air</td>
</tr>
<tr>
<td>Deteriorated lining</td>
<td>Aged/heat</td>
<td>Exposed metal-on-metal contact reduces arresting capability</td>
</tr>
<tr>
<td>Worn bearing</td>
<td>Lack of lubrication</td>
<td>Low rotary motion</td>
</tr>
<tr>
<td>Worn seals</td>
<td>Aged</td>
<td>External leakage</td>
</tr>
<tr>
<td>Cracked housing</td>
<td>Vibration, fatigue</td>
<td>External leakage</td>
</tr>
</tbody>
</table>

Table 12-3. Metal Countersurface Failures (Reference 32)

<table>
<thead>
<tr>
<th>FAILURE</th>
<th>CHARACTERISTICS</th>
<th>CAUSES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat spotting</td>
<td>Often cracks are formed in these regions owing to structural changes in the metal.</td>
<td>Friction material not sufficiently conformable to the metal member</td>
</tr>
<tr>
<td>Crazing</td>
<td>Randomly oriented cracks</td>
<td>Overheating and repeated stress cycling</td>
</tr>
<tr>
<td>Scoring</td>
<td>Scratches in the line of movement</td>
<td>Metal too soft for friction material. Abrasive debris embedded in the lining material.</td>
</tr>
</tbody>
</table>

Components such as brake shoes, which are primarily structural, should be modeled using a Finite Element Analysis. The total brake system failure rate is the sum of the failure rates of each of the above component parts in the system:
\[ \lambda_{BR} = \lambda_{AC} + \lambda_{SP} + \lambda_{FR} + \lambda_{BE} + \lambda_{SE} + \lambda_{HO} \]  

Where:

- \( \lambda_{BR} \) = Total failure rate for the brake system, failures/million hours
- \( \lambda_{AC} \) = Total failure rate for actuators, failures/million hours (See Chapter 9)
- \( \lambda_{SP} \) = Total failure rate for springs, failures/million hours (See Chapter 4)
- \( \lambda_{FR} \) = Total failure rate for brake friction materials, failures/million hours (See Sections 12.2.5 and 12.2.6)
- \( \lambda_{BE} \) = Total failure rate for bearings, failures/million hours (See Chapter 7)
- \( \lambda_{SE} \) = Total failure rate for seals, failure/million hours (See Chapter 3)
- \( \lambda_{HO} \) = Total failure rate for brake housing, 3.0 failures/million hours, from Navy Maintenance and Material Management Information System

In the hydraulic drives of brake systems, seals are used to prevent leakage of brake fluid. The hardness and swelling of the seals, when exposed to brake fluid, must remain within limits such that the seals will give reliable operation.

The reliability of springs associated with brake systems is generally very high when compared to other components. Some of the spring assemblies in a brake system may be static, maintaining a constant tension on a part, other springs may be cyclic or dynamic depending on their function.

Severe performance requirements may affect the reliability of the bearings if there is a path of heat conduction from the friction surface to the bearings. This conduction may cause a decrease in the bearing lubricants operating viscosity and, consequently, a reduction in bearing life. A lubricant with a higher temperature rating should prevent leakage or excessive wear.

The reliability of brake actuators normally is very high. Under severe brake performance, conditions of increased temperature and excessive vibration may decrease the reliability of these components. Refer to the appropriate sections of this Handbook for the reliability models for individual parts comprising the brake assembly. In some cases the result in failure/million cycles will have to be converted to failures/million hours by multiplying by the number of cycles per hour.
12.2.5 Friction Materials

As stated in the introduction, the major functional components of brake equipment and clutch equipment are the friction materials. The reliability of brakes and clutches is concerned with the wear of these friction materials. For brake assemblies, the friction lining provides the friction necessary to slow down or stop a vehicle. Friction materials used in clutches are placed in the power-transmission system to couple it together so it rotates as one unit.

Friction materials that are used in brake and clutch linings have severe performance requirements. The necessary energy conversion must be accomplished with a minimum of wear on the contacting parts. For a particular type of brake or clutch, the amount of heat and friction generated varies according to 5 conditions: (1) the amount of pressure applied between the sliding surfaces, (2) the operating environment, (3) the roughness of the surfaces, (4) the material from which the surfaces are made, and (5) the frequency of application.

The reliability of these high energy components is important for a variety of reasons: economy, operational readiness and, most important, safety. In today's modern machinery and equipment, a vast number of friction materials have become available to fulfill the very diverse requirements of this equipment group. However, a material which is exceptional in all areas of friction material criteria does not yet exist.

In design it is necessary to have equations for the prediction of the wear life of clutches and brakes. Lining wear properties are generally considered in terms of system life under several different conditions of use severity. Consequently, lining life is often the last performance character to be quantified. Thus, knowledge of lining wear behavior from laboratory testing can be of great value.

Friction modifier additives, such as cashew resin, graphite, etc. have been used for many years in order to control friction properties in brake and clutch composites. Friction composites are composed of a balanced mixture of resin plus additives and generally contain over a dozen ingredients in order to achieve desired characteristics.

In the past, materials such as wood, leather and felt were used, but it was found that the usable temperature range was inadequate to cope with the ever increasing demands made upon them by the industrial world. Today, friction materials can be divided into five main groups:

1. Woven cotton
2. Solid-woven asbestos
3. Rigid molded asbestos, semi-flexible molded asbestos, flexible molded asbestos
4. Sintered metal
5. Cermets
Refer to Table 12-4 for a summary of friction material types and applications.

Table 12-4. Friction Material Types And Applications
(Reference 32)

<table>
<thead>
<tr>
<th>TYPE</th>
<th>MANUFACTURE</th>
<th>APPLICATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>Woven cotton</td>
<td>Closely woven belt of fabric is impregnated with resins which are then polymerized</td>
<td>Industrial drum brakes, mine equipment, cranes, lifts</td>
</tr>
<tr>
<td>Woven asbestos</td>
<td>Open woven belt of fabric is impregnated with resins which are then polymerized. May contain wire to scour the surface</td>
<td>Industrial band and drum brakes, cranes, lifts, excavators, winches, concrete mixers, mine equipment</td>
</tr>
<tr>
<td>Molded flexible</td>
<td>Asbestos fiber and friction modifiers mixed with thermo-setting polymer and mixture heated under pressure</td>
<td>Industrial drum brakes; heavy duty brakes, excavators, tractors, presses</td>
</tr>
<tr>
<td>Semi-flexible rigid</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sintered metal</td>
<td>Iron and/or copper powders mixed with friction modifiers</td>
<td>Heavy duty brakes and clutches, press brakes, earthmoving equipment</td>
</tr>
<tr>
<td>Cermets</td>
<td>Similar to sintered metal pads, but large portion of ceramic material present</td>
<td>Heavy duty brakes and clutches, press brakes, earthmoving equipment</td>
</tr>
</tbody>
</table>

Friction material manufacturers are usually very reluctant to disclose the composition or formulation of their products. Some basic information is, however, necessary to properly analyze and carefully select the friction material for a given application. Formulation of a lining is defined as a specified mixture of materials from which the lining is made and the corresponding sequence of production processing which together determine the characteristics of the lining.

Organic linings are generally comprised of six basic ingredients:

1. Asbestos for heat resistance and high coefficient of friction
2. Friction modifiers such as the oil of cashew nutshell to give desired friction qualities
3. Fillers such as rubber chips for controlling noise
(4) Curing agents to produce the required chemical reactions in the ingredients
(5) Materials such as powdered lead, brass chips, and aluminum powders for improving the overall braking performance
(6) Binders such as phenolic resins for holding the ingredients together

Organic linings designed for heavy-duty use generally have higher inorganic contents to improve their high temperature wear resistance and fade resistance. Abrasives are generally added to achieve a higher friction coefficient.

Friction materials containing conventional organic binding agents exhibit poor frictional stability under varying temperature conditions. The thermal degradation of such binders results in inferior frictional characteristics, giving rise to fade and often resulting in increased wear. Furthermore, organic materials, particularly resins, tend to have a short shelf life, and are not always easy to reproduce.

In an attempt to overcome the deleterious effects of poor thermal resistance in a friction material having an organic binder, various sintered metal and ceramic materials, in which the sintering affects the bonding, have been developed. In comparison with friction materials produced with organic, resinous binding agents, sintered friction materials have the primary advantage of being able to withstand considerably higher thermal stresses. They are produced from an intimate mixture of powdered metals and nonmetals by pressing and sintering.

These friction materials commonly consist of sintered lead bronzes and iron powders with additions of dry lubricants and so-called friction reinforcers. Graphite and molybdenum disulfide, for example, are suitable as dry lubricants; on the other hand, ceramic additives and minerals, such as quartz and corundum, may be used to increase the coefficient of friction. By appropriate variation in the additives it is possible to make adaptations for all applications, particularly concerning the coefficient of friction.

Semi-metallics rely heavily on iron, steel, and graphite substitutions for the organic and asbestos materials. Some organic components are, however, used to obtain desirable properties. The use of abrasives must be minimized to maintain acceptable mating surface compatibility. Semi-metallics have distinct advantages over conventional organics such as:

- Improved frictional stability and fade resistance
- Excellent compatibility with rotors and high temperature wear resistance
- High performance with minimal noise

The cost of raw material mix represents the major factor in the premium prices of semi-metallics, and as such, widespread use of semi-metallics is not yet found. Metallic linings withstand more severe loads, higher temperatures, and have less tendency to fade. Sintered metallic-ceramic friction materials have successfully been used for
specialized applications such as jet aircraft. See Table 12-5 for a summary of the brake friction material surface failures.

**Table 12-5. Brake Friction Material Failure Modes** *(Reference 32)*

<table>
<thead>
<tr>
<th>PROBLEM</th>
<th>CHARACTERISTICS</th>
<th>CAUSES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat spotting</td>
<td>Heavy gouging resulting in rapid lining wear</td>
<td>Material rubbing against a heat spotted metal member</td>
</tr>
<tr>
<td>Crazing</td>
<td>Randomly oriented cracks on the friction material, resulting in a high wear rate</td>
<td>Overheating of the braking surface</td>
</tr>
<tr>
<td>Scoring</td>
<td>Grooves formed on the friction material, resulting in a reduction of life</td>
<td>Metal member needs regrinding</td>
</tr>
<tr>
<td>Fade</td>
<td>Material degrades or flows at the friction surface, resulting in a temporary loss of performance</td>
<td>Overheating caused by excessive braking</td>
</tr>
<tr>
<td>Metal pick-up</td>
<td>Metal from the mating member embedded in the lining</td>
<td>Unsuitable combination of materials</td>
</tr>
<tr>
<td>Grab</td>
<td>Lining contacting at ends only giving high servo effect and erratic performance</td>
<td>Incorrect radiusing of lining</td>
</tr>
<tr>
<td>Strip braking</td>
<td>Braking over a small strip of the rubbing path giving localized heating and preferential wear at those areas</td>
<td>Distortion of the brake path</td>
</tr>
<tr>
<td>Neglect</td>
<td>Material completely worn off the shoe reducing performance</td>
<td>Failure to provide any maintenance</td>
</tr>
<tr>
<td>Misalignment</td>
<td>Excessive grooving wear at preferential areas of lining surface</td>
<td>Lining not fitted correctly to the shoe platform</td>
</tr>
</tbody>
</table>

**12.2.6 Disk / Brake Lining Reliability Model**

There are several factors that affect the wear rate of friction components of brakes, including:

- (a) nominal pressure
- (b) elastic properties of the materials
- (c) strength properties of the materials
- (d) surface roughness of the mating surfaces
- (e) temperature of the material
(f) compatibility of the lining/drum or disk/pad materials

The wear of the brake lining or disk pad material has been correlated by the following equation (Reference 77):

\[ V = k_o \cdot P \cdot s \quad (12-2) \]

Where:
- \( V \) = Volume of material lost by wear, in\(^3\)
- \( k_o \) = Wear coefficient, (lb/in\(^2\))\(^{-1}\)
- \( P \) = Applied load, lbf
- \( s \) = Sliding distance during braking, in = \( v_s \cdot t_b \)
- \( v_s \) = Sliding velocity, in/sec
- \( t_b \) = Braking time, sec

If the effective thickness of the lining or pad is \( d \) (inches), pad life is commonly given by the following linear relationship:

\[ \text{Life} = \frac{d}{W_p} \quad (12-3) \]

Where:
- \( \text{Life} \) = Number of applications before friction material is completely worn
- \( d \) = Lining thickness, in
- \( W_p \) = Pad wear per application, in

Equation (12-2) can be written in terms of surface pad/lining wear per application:

\[ W_p = \frac{k_o \cdot P \cdot v_s \cdot t_b}{A} \quad (12-4) \]

where:
- \( A \) = Brake lining area, in\(^2\)

and:
\[
\lambda_{FR,B} = \frac{1}{\text{Life}} = \frac{W_p}{d} = \frac{k_a P v_s t_b}{d A}
\]  \hspace{1cm} (12-5)

By normalizing Equation (12-5) to those values for which historical failure rate data are available, the following failure rate model can be derived:

\[
\lambda_{FR} = \lambda_{FR,B} \cdot C_{BT} \cdot C_{RD} \cdot C_T
\]  \hspace{1cm} (12-6)

Where:
- \(\lambda_{FR} = \) Failure rate of the brake friction material in failures/million hours
- \(\lambda_{FR,B} = \) Base failure rate of the brake friction material, failures/million hours (See Section 12.2.6.1)
- \(C_{BT} = \) Multiplying factor which considers the effect of brake type on the base failure rate (See Section 12.2.6.2)
- \(C_{RD} = \) Multiplying factor which considers the effect of dust contaminants on the base failure rate (See Section 12.2.6.3)
- \(C_T = \) Multiplying factor which considers the effect of ambient temperature on the base failure rate (See Section 12.2.6.4)

12.2.6.1 Base Failure Rate for Brake Lining / Disk Material

The brake friction material base failure rate, \(\lambda_{FR,B}\), may be provided by the lining manufacturer. If not, then the base rate can be calculated from Equation (12-5).

Wear coefficients are included in Table 12-6. It should be noted that \(\lambda_{FR,B}\) is in terms of brake actuations. During a single brake actuation the vehicle wheel or industrial machine will rotate several rotations during the stopping or slowing process.

12.2.6.2 Brake Type Multiplying Factor

A typical disk brake will wear better than a drum type due to the disk brakes ability to dissipate heat more quickly. The friction material for the annular brake is in the shape of an annulus and is bonded to both sides of the rotor disk. The slotted annular brake is nearly the same as the annular brake described above, the only exception being the presence of slots cut through the friction material on both sides of the rotor. The purpose of the slots is to decrease surface temperature and wear rate during braking. The pad brake configuration employs pads of friction material on the brake
stators. Multiplying factors for the specific type of brake design are as follows, based on field performance data (Reference 20):

\[
C_{BT} = \begin{align*}
1.25 & \text{ for drum type brakes} \\
1.25 & \text{ for slotted annular disk type brakes} \\
1.00 & \text{ for pad disk type brakes} \\
0.90 & \text{ for annulus disk type brakes}
\end{align*}
\]

**Table 12-6. Typical Wear Coefficients for Brake Linings**  
(Against cast iron or steel)

<table>
<thead>
<tr>
<th>Lining (pad) Material</th>
<th>Wear Coefficient, (k_o) (psi(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Asbestos-type I Composite</td>
<td>(6.46 \times 10^{-11})</td>
</tr>
<tr>
<td>Asbestos-type II Composite</td>
<td>(8.09 \times 10^{-11})</td>
</tr>
<tr>
<td>Carbon-Carbon Composite</td>
<td>(2.24 \times 10^{-11})</td>
</tr>
<tr>
<td>Sintered bronze (dry)</td>
<td>(2.42 \times 10^{-10})</td>
</tr>
<tr>
<td>Non-asbestos composite (dry)</td>
<td>(9.90 \times 10^{-10})</td>
</tr>
<tr>
<td>Sintered bronze (wet)</td>
<td>(5.02 \times 10^{-13})</td>
</tr>
<tr>
<td>Sintered bronze composite</td>
<td>(9.31 \times 10^{-11})</td>
</tr>
<tr>
<td>Sintered resin composite</td>
<td>(3.03 \times 10^{-11})</td>
</tr>
</tbody>
</table>

**12.2.6.3 Dust Contaminant Multiplying Factor**

Operating conditions with high amounts of dust contaminants affects lining wear depending on the binder resin used in formulating the friction material. The correction factor for dust conditions is shown in Table 12-7 (Reference 42).
Table 12-7. Dust Contamination Multiplying Factor

(Reference 42)

<table>
<thead>
<tr>
<th>Binder Resin</th>
<th>$C_{RD}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Phenolic</td>
<td>3.5</td>
</tr>
<tr>
<td>Oil-modified phenolic</td>
<td>1.2</td>
</tr>
<tr>
<td>Rubber phenolic</td>
<td>1.1</td>
</tr>
<tr>
<td>Cashew</td>
<td>1.1</td>
</tr>
<tr>
<td>Oil-phenolic</td>
<td>1.1</td>
</tr>
</tbody>
</table>

12.2.6.4 Temperature Multiplying Factor

Wear of the friction material will be influenced by the ambient temperature in which the vehicle is operating. The correction factor for temperature is (Reference 3):

\[ C_T = 1.42 - 1.54E - 3X + 1.38E - 6X^2 \]  \hfill \text{(12-7)}

(for sintered metallic linings)

\[ C_T = 2.79 - 1.09E - 2X + 1.24E - 5X^2 \]  \hfill \text{(12-8)}

(for resin-asbestos linings used in light duty automotive and moderate duty industrial brakes)

\[ C_T = 3.80 - 7.59E - 3X + 5.07E - 6X^2 \]  \hfill \text{(12-9)}

(for carbon-carbon linings)

\[ C_T = 17.59 - 6.03E - 2X + 5.34E - 5X^2 \]  \hfill \text{(12-10)}

(for resin-asbestos truck linings)

Where:

\[ X = 590 + T \]

\[ T = \text{Ambient temperature, } ^\circ\text{F} \]
12.3 CLUTCHES

12.3.1 Introduction

The reliability of a clutch system is generally very high and is the result of the low failure rate of its parts, which may include actuators, bearings, clutch friction linings, seals and springs. With the exception of clutch friction linings, these component parts are addressed in other chapters of this handbook. The general characteristics of friction materials are also addressed in the first part of this chapter. Those characteristics of friction materials peculiar to clutches will be discussed in the following paragraphs.

The principal function of friction clutches is to convert kinetic energy to heat and then either to absorb or otherwise dissipate the heat while simultaneously, through friction, reducing the relative movement between the friction material and the part to which it is engaged. In order to achieve these objectives the necessary energy conversion must be accomplished with a minimum of wear on the contacting parts.

12.3.2 Clutch Varieties

Clutches are made up of two basic components - the pressure plate and disc. The pressure plate supplies sufficient force or pressure to the disc so enough friction is developed to transmit torque to the driveline.

Friction clutches, although available in many different forms tend to be of the axial or rim type. Axial clutches operate where the movement is parallel to the axis of the shaft. Rim types operate where the movement is radial. Examples of the former are the plate and cone clutches. Examples of the latter include coil or wrap spring and chain clutches.

Plate clutches are divided into two designs - single and multiplate. The single plate design is the type favored by automotive designers for transmission and light to medium power applications. The single plate is normally provided with a friction lining on each side of the disc. Multiplate designs employ a number of discs lined on both sides which serve to distribute the load over a large area. These types are used for high torque and high load applications. They require only moderate clamping pressures, and are suitable for high speed operation because their relatively small size generates lower centrifugal forces.

Cone clutches are used for smaller, medium power, low speed transmission systems which may be subjected to rough usage. These devices cope well with such treatment because of their simple robust construction, and due to the fact that heat is dissipated more readily than with plate clutches.
Rim and block clutches employ various means of engaging the stationary half of the assembly through radial movement against the rim of the driving member. The action is similar to that of an internally expanding brake shoe.

Centrifugal clutches are often used with squirrel cage motors. The fabric facing may be fitted to shoes or blocks mounted to a spider which is keyed onto the driving shaft. The shoes or blocks are thrown outward by centrifugal force, engagement being automatic when a predetermined speed is reached from starting.

Coil or wrap spring clutches operate on the principle of a spring mounted on a drum being tightened. The action is much like that of a rope tightening around a revolving capstan. The design is compact, simple in construction and is used where high torques are required from low power. For this reason the clutches have found applications in small equipment such as plain paper copiers and, in their larger versions, for haulage gears and rolling mills and presses.

Chain clutches employ inner and outer friction rings in an oil filled housing actuated by cams bearing on chain toggles which force the rings together.

Sprag clutches consist of a number of specially shaped steel springs or wedges which jam inner and outer races in one direction only. This action leads to their use for applications in over-running (where the clutch acts as a free-wheel) and back-stopping. This design is particularly useful for intermittent rotary motion involving, for example, indexing or inching (Reference 34).

Materials classification divides the friction materials into organic and metallic groups. The organic group includes all materials composed of both asbestos and non-asbestos fibers and bound by some resin binder. The metallic group consists of all friction materials containing iron, copper, ceramic bronze, graphite, carbon or other metallic material as the base material.

### 12.3.3 Clutch Failure Rate Model

The clutch system reliability model will contain the following component parts:

- Actuators
- Bearings
- Clutch friction linings
- Seals
- Springs

The total clutch system failure rate is the sum of the failure rates of each of the above component parts in the system:
Where:

\[ \lambda_{CL} = \lambda_{AC} + \lambda_{BE} + \lambda_{CF} + \lambda_{SE} + \lambda_{SP} \]  \hspace{1cm} (12-11)

- \( \lambda_{CL} \) = Total failure rate for the clutch system, failures/million hours
- \( \lambda_{AC} \) = Total failure rate for actuators, failures/million hours (See Chapter 9)
- \( \lambda_{BE} \) = Total failure rate for bearings, failures/million hours (See Chapter 7)
- \( \lambda_{CF} \) = Total failure rate for clutch friction materials, failures/million hours (See Section 12.3.4)
- \( \lambda_{SE} \) = Total failure rate for seals, failures/million hours (See Chapter 3)
- \( \lambda_{SP} \) = Total failure rate for springs, failures/million hours (See Chapter 4)

The failure rates obtained from other chapters of the Handbook may have to be converted from failures/million cycles to failures/million hours by multiplying by the number of cycles per hour. The failure rate model for clutch friction materials is presented in the following paragraphs.

### 12.3.4 Clutch Friction Material Reliability Model

A list of failure modes for clutch friction materials is shown in Table 12-8. By using the clutch system beyond the life of the friction material a drastic reduction of friction coefficient can occur. This rapid deterioration can result in a catastrophic failure of the clutch.

Under normal operating conditions, the friction materials used in clutches are reliable mechanical components. Like brake friction materials, the wear of clutch materials is dependent on the amount of accumulated energy dissipated by the mechanical component.

\[
h = k_o p s \]  \hspace{1cm} (12-12)

Where:

- \( h \) = Change in thickness of the clutch friction material caused by wear, in
- \( k_o \) = Wear coefficient, \((\text{lb}/\text{in}^2)^{-1}\)
- \( p \) = Nominal pressure between the clutch wear plates, \(\text{lbf}/\text{in}^2 = P/A\)
- \( P \) = Applied load, lbf
- \( A \) = Wear plate area, \(\text{in}^2\)
- \( s \) = Sliding distance during clutch actuation, \(\text{in} = v_s \cdot t_a \)
- \( v_s \) = Sliding velocity, \(\text{in/sec}\)
- \( t_a \) = Clutch actuation time, sec
If the effective thickness of the clutch lining is $d$ (inches), life of the clutch friction material is given by the following equation:

$$\text{Life} = \frac{d}{W_p} \quad (12-13)$$

Where:  
- Life = Number of applications before friction material is completely worn  
- $d$ = Lining thickness, in  
- $W_p$ = Friction material wear per application, in

$$W_p = \frac{k_o \cdot P \cdot v_s \cdot t_a}{A} \quad (12-14)$$

where:  
- $A$ = Clutch lining area, in$^2$

and:

$$\lambda_{FR,B} = \frac{1}{\text{Life}} = \frac{W_p}{d} = \frac{k_o \cdot P \cdot v_s \cdot t_a}{d \cdot A} \quad (12-15)$$

By normalizing Equation (12-15) to those values for which historical failure rate data is available, the following failure rate model can be derived:

$$\lambda_{CF} = \lambda_{CF,B} \cdot C_{NP} \cdot C_T \quad (12-16)$$

Where:  
- $\lambda_{CF}$ = Failure rate of the clutch friction material in failures/million hours  
- $\lambda_{CF,B}$ = Base failure rate of the clutch friction material, failures/million hours (See Section 12.3.4.1)  
- $C_{NP}$ = Multiplying factor which considers the effect of multiple plates on the base failure rate (See Section 12.3.4.2)  
- $C_T$ = Multiplying factor which considers the effect of ambient temperature on the base failure rate (See Section 12.3.4.3)
12.3.4.1 Base Failure Rate for Clutch Lining / Disk Material

The clutch friction material base failure rate, \( \lambda_{FR,B} \), may be provided by the manufacturer of the clutch assembly. If not, then the base rate can be calculated from Equation (12-16).

Wear coefficients are included in Table 12-6. It should be noted that \( \lambda_{CF,B} \) is in terms of brake actuations. During a single clutch engagement the equipment will go through several rotations during the engagement process.

As noted previously, clutches can be divided into two design groups: single and multiple plate. Multiplate designs use a number of discs which distribute the load, and will therefore increase the reliability of the system.

12.3.4.2 Clutch Plate Quantity Multiplying Factor

The correction factor for the number of plates is given by:

\[
C_{NP} = \text{Number of disks in the clutch}
\]

12.3.4.3 Temperature Multiplying Factor

Because the temperature of the friction material affects the wear of the material, the ambient temperature to which the clutch is exposed will affect the wear of the friction lining (Reference 3). As a result:

\[
C_T = 1.42 - 1.54E - 3X + 1.38E - 6X^2 \tag{12-17}
\]

(for sintered metallic linings)

\[
C_T = 2.79 - 1.09E - 2X + 1.24E - 5X^2 \tag{12-18}
\]

(for resin-asbestos linings used in light duty automotive and moderate duty industrial brakes)

\[
C_T = 3.80 - 7.58E - 3X + 5.07E - 6X^2 \tag{12-19}
\]

(for carbon-carbon linings)
\[ C_T = 17.59 - 6.03E - 2X + 5.43E - 5X^2 \]  

(12-20)

(for resin-asbestos truck linings)

Where: \[ X = 590 + T \]

\[ T \] = Ambient temperature, °F

Table 12-8. Clutch Friction Surface Failure Modes (Reference 32)

<table>
<thead>
<tr>
<th>PROBLEM</th>
<th>CHARACTERISTICS</th>
<th>CAUSES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dishing</td>
<td>Clutch plates distorted into a conical shape</td>
<td>Lack of conformability. The temp. of the outer region of the plate is higher than the inner region.</td>
</tr>
<tr>
<td>Waviness or Buckling</td>
<td>Clutch plates become buckled into a wavy platter</td>
<td>Lack of conformability. The inner area is hotter than the outer area.</td>
</tr>
<tr>
<td>Banding or Crushing</td>
<td>Loss of friction material at the ends of a band</td>
<td>Crushing and excessive wear of the friction material</td>
</tr>
<tr>
<td>Material Transfer</td>
<td>Friction material adhering to opposing plate, often giving rise to excessive wear</td>
<td>Overheating and unsuitable friction material</td>
</tr>
<tr>
<td>Bond Failure</td>
<td>Material parting at the bond to the core plate causing loss of performance</td>
<td>Poor bonding or overheating, the high temperature affecting bonding agent</td>
</tr>
<tr>
<td>Burst Failure</td>
<td>Material splitting and removed from the spinner plate</td>
<td>High stresses on a facing when working at high speeds</td>
</tr>
<tr>
<td>Grooving</td>
<td>Grooving of the facing material on the line of movement</td>
<td>Material transfer to opposing plate</td>
</tr>
<tr>
<td>Reduced Performance</td>
<td>Decrease in coefficient of friction giving a permanent loss in performance</td>
<td>Excess oil or grease on friction material or on the opposing surface</td>
</tr>
<tr>
<td>Distortion</td>
<td>Facings out of flatness after high operating temperature</td>
<td>Unsuitable friction material</td>
</tr>
</tbody>
</table>
12.4 REFERENCES


13.1 INTRODUCTION

A compressor is a machine for compressing air from an initial intake pressure to a higher exhaust pressure through a reduction in volume. A compressor consists of a driving unit, a compressor unit, and accessory equipment. The driving unit provides power to operate the compressor and may be a gasoline or diesel engine. A compressed air system consists of one or more compressors, each with the necessary power source, control of regulation, intake air filter, aftercooler, air receiver, and connecting piping, together with a distribution system to carry the air to points of use.

The compression of a gas by mechanical means, and the raising of it to some desired pressure above that of the atmosphere, is usually characterized by an approximate adiabatic process. Ideal adiabatic compression of air, relating pressure and volume can be given by:

\[ PV^{1.4} = C \]  

(13-1)

where \( P \) = pressure, \( V \) = volume and \( C \) = constant

A compression of this nature could heat the air to temperatures which would interfere with the reliable action of an air compressor and introduce lubrication difficulties, if there were no provisions for cooling the walls of the compression
chamber. The extraction of heat from a compression cycle modifies the conditions of compression from the ideal to some change more nearly represented by:

$$PV^n = C$$  \hspace{1cm} (13-2)

where the value of $n$ is usually between 1.35 and 1.40.

If the heat of compression is removed by cooling as rapidly as it is formed, an isothermal compression will result. Less work is needed for compression of a pound of gas to the same discharge pressure. Although isothermal compression is desirable, it is not possible to achieve in fast-moving compressors. As a result, finned or jacketed cylinder compression is more nearly adiabatic than isothermal.

Compressors can be classified, in their broadest sense, in two categories: (1) positive-displacement and (2) centrifugal machines. The positive-displacement classification can generally be described as "volume reducing" types. In essence, an increase in gas pressure can be achieved by simultaneously reducing the volume enclosing the gas. The centrifugal classification refers to centrifugal velocity increases. These machines impart energy to the gas, and then stationary diffusors convert the velocity head into pressure. The classification tree in Figure 13.1 further defines the subcategories of compressors.

Figure 13.1 Common Classifications for Compressors
The positive-displacement (volume reducing) machines can be further defined by two sub classifications; rotary and reciprocating. Both types generally feature steep characteristic curves of performance. A nearly-constant capacity coupled with varying discharge pressure is typical, reflecting a machine capable of slight variations in flow over a wide pressure range.

Reciprocating machines can be modeled as adiabatic pressure generating devices. Systems requiring higher pressures and lower volumetric flow rates usually employ these machines. Typical operating ranges for this type of machine are presented in Figure 13.2. In compression to high pressures, the temperature rise may be too great to permit the compression to be carried to completion in one cylinder, even though it is cooled. In such cases, the compression is carried out in stages, with a partial increase of pressure in each stage, and cooling of the gas between stages. Two and three-stage compression is very common where pressures of 300-1000 psi are needed. In determining the number of stages (pistons) within a reciprocating compressor, the change in temperature across a stage, the frame or rod loading, and the change in pressure across a stage are among the parameters taken into consideration. The ratio of the temperature before and after compression can be expressed from a form of the Ideal Gas Law:

\[
\left( \frac{T_2}{T_1} \right) = \left( \frac{V_2}{V_1} \right)^{n-1} = \left( \frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}}
\]

(13-3)

where \(T_1\) and \(T_2\) are expressed in degrees Rankine.

Rotary positive displacement machines incorporate some type of rotating element that displaces a fixed volume during each machine revolution. The characteristics performance curve is basically the same as a reciprocating machine. Typical operating ranges for this type of machine are presented in Figure 13.2.

The rotary lobe compressor is typically constructed with two or three figure eight-shaped rotors, meshed together, and driven through timing gears attached to each shaft. It is a relatively low pressure machine (5 to 7 psig, and up to 25 psig for special types) and is well suited for applications with vacuum pressures. Its performance is notable for a greater throughput capability, with little or no flow pulsation.

The rotary screw compressor yields considerably higher pressures and speed. Again, its performance is characteristic of a greater throughput capability with little or no pulsation.

The sliding vane rotary compressor has a rotor construction which is offset, containing slots for vanes to slide in and out during each revolution. These vanes
gradually reduce the volume of a trapped gas, raising its pressure. This machine is used for relatively low pressure operations (up to 50 psig per stage).

A liquid ring (or piston) rotary is constructed of circular vanes, turning inside a casing sealed with a liquid. Centrifugal forces cause the liquid to form a ring around the periphery of the casing interior, while forcing the gas inward toward the center of the vaned rotor. The gradual decrease in volume increases the pressure of the gas. Any liquid entrained in the gas is separated out. This type of machine is characteristically used in low pressure and vacuum applications.

Centrifugal compressors can be divided into two subcategories based on the direction of flow of the product gas: radial flow and axial flow machines. The characteristic curves of these machines offer a wide range in flow with a corresponding small change in head. Flow is smooth and pulsation-free beyond the surge point on the performance curve. The lack of rubbing parts in the compressed gas stream is a particularly desirable feature of these machines from a designer’s standpoint.

In radial compressors, velocity is imparted to a gas stream through centrifugal forces acting in a radial direction to the shaft. The simplest style of radial centrifugal compressor is the single-stage overhung design. The conventional closed or shrouded

Figure 13.2 Typical Operating Range of Compressors for Process Use
impeller is used for adiabatic heads to about 12,000 ft-lb/lb. The open, radial-bladed impeller develops more head for the same diameter and speed.

In axial flow machines, the gas flow remains parallel to the shaft, without a direction change. These machines are typically used for higher capacities than radial flow machines, but generate much lower head per stage. As a result, these machines are usually built with many stages. The characteristic performance curve is steeper than that of radial flow machines, with a more narrow stability range.

13.2 COMPRESSOR FAILURE MODES

Some failure modes are more prevalent than others as a direct result of the variety of compressor types and differing environmental conditions of operation. Certain compressor parts will fail more frequently than others. An analysis of various failure modes for compressors and certain compressor parts is presented in Table 13-1.

13.3 FAILURE RATE MODEL FOR COMPRESSOR ASSEMBLY

Any compressor, taken as a complete operating system, can be reduced to the following series of models of each of its component parts. Each of these parts will sum to the total compressor failure rate:

\[
\lambda_C = \lambda_{SH} + \lambda_{BE} + \lambda_{CA} + \lambda_{VA} + \lambda_{SE} + \lambda_{DC}
\]  

(13-4)

Where: 
\( \lambda_C \) = Total failure rate of compressor, failures/million hours
\( \lambda_{SH} \) = Total failure rate for the compressor shaft(s), failures/million hours (See Chapter 20)
\( \lambda_{BE} \) = Total failure rate for all compressor bearings, failures/million hours (See Chapter 7)
\( \lambda_{CA} \) = Total failure rate for the compressor casing, failures/million hours (See Section 13.4)
\( \lambda_{VA} \) = Total failure rate for any included valve assembly, failures/million hours (See Chapter 6)
\( \lambda_{SE} \) = Total failure rate for all compressor seals, failures/million hours (See Chapter 3)
\( \lambda_{DC} \) = Total failure rate of compressor design configuration, failures/million hours (See Section 13.5)
The failure rate, \( \lambda \), for each part listed above must be known or calculated before the entire compressor assembly failure rate, \( \lambda_{C} \), can be determined. Failure rate values for each part will incorporate expected operational and environmental factors that exist during normal compressor operation.

Table 13-1. Compressor Failure Modes

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reduction of internal clearances</td>
<td>Distortion of rotor due to cyclic loading; Improper material selection for thermal expansion</td>
<td>Rubbing, increased wear</td>
</tr>
<tr>
<td>Increased vibration</td>
<td>High fluctuating Stresses</td>
<td>Material fatigue</td>
</tr>
<tr>
<td>Increased friction And wear</td>
<td>Contaminants</td>
<td>Decreased performance, increased vibration</td>
</tr>
<tr>
<td>Valve sticking</td>
<td>Over lubrication, moisture in oil</td>
<td>Overheating, increased wear</td>
</tr>
<tr>
<td>Low flow pulsation</td>
<td>Thrust reversal, vibration</td>
<td>Bearing failure, overheating</td>
</tr>
<tr>
<td>Corrosion or cracking of diaphragm</td>
<td>Contaminants</td>
<td>Decreased performance</td>
</tr>
<tr>
<td>Accelerated curing, embrittlement of diaphragm</td>
<td>Extreme high or low temperature</td>
<td>Decreased performance</td>
</tr>
</tbody>
</table>

13.4 FAILURE RATE MODEL FOR CASING

The compressor casing, normally a very reliable component, can have a large effect on the life of other components in the compressor assembly (especially seals and bearings). The value of reliability of compressor casings, through the experience of many different manufacturers, can generally be equated to a \( \lambda_{CA} \) value of 0.001 failures/million hours.
13.5 FAILURE RATE MODEL FOR DESIGN CONFIGURATION

Various reliabilities are inherent in specific designs (types) of compressors. For example, it is expected that the reliability due to wear will be different in a rotary screw compressor compared to a centrifugal compressor due to the nature of metal-to-metal contact and rotor speeds. The various chapters of this handbook can be used to estimate the failure rates of the individual component parts. The parameter $\lambda_{DC}$ can be approximated by data presented in Table 13-4 for various types of fluid drivers, developed from information collected by the U.S. Navy.

As an example, the compressor may be a reciprocating design containing a diaphragm. In this case:

$$\lambda_{DC} = \lambda_{DI}$$

Where $\lambda_{DI} =$ Total failure rate of the diaphragm assembly

The following example is a method of estimating the failure rate of a compressor diaphragm.

The configuration diaphragm compressor failure rate model can be described by:

$$\lambda_{DI} = \lambda_{BFD} \cdot C_P \cdot C_{AC} \cdot C_{LC} \cdot C_T$$

(13-5)

Where: $\lambda_{BFD} =$ Compressor diaphragm base failure rate, 0.58 failures/million hrs. (See Table 13-4)

$C_P =$ Factor for effects of axial loading (See Section 13.5.1)

$C_{AC} =$ Factor for effects of atmospheric contaminants (See Section 13.5.2)

$C_{LC} =$ Factor for effects of liquid contaminants (See Section 13.5.3)

$C_T =$ Factor for effects of temperature (See Section 13.5.4)

13.5.1 Axial Load Multiplying Factor, $C_P$

Diaphragms, in general, are round flexible plates which undergo an elastic deflection when subjected to an axial loading. In the application of compressors, this axial loading and elastic deflection creates a reduction in volume of the space adjacent to the diaphragm. The gas is compressed and a pressure builds. The diaphragm can be designed in many different ways with variations in such parameters as materials,
size and shape. The model developed for a compressor diaphragm is shown in Figure 13.3. It has a passive area in the center which is rigid. This area transmits a force from the push rod to the diaphragm. To be effective, the thickness of the rigid center should be at least 6 times the thickness of the diaphragm.

![Compressor Diaphragm Model](image)

**Figure 13.3 Compressor Diaphragm Model**

The characteristic equations describing the compressor diaphragm are given in Equations (13-6) through (13-11) and are based on the following restrictive assumptions:

1. Diaphragm is flat and of uniform thickness.
2. Diaphragm material is isotropic and homogeneous.
3. All forces, loads, and reactions are applied normally to the plane of the plate.
4. Diaphragm thickness not greater than 20% of its diameter.
5. The effects of shearing stresses and pressures on planes parallel to the surface of the diaphragm have not been taken into account. They are considered insignificant in diaphragms with thickness to radius ratios \((h/a)\) of less than 0.15.
6. The stresses created in a diaphragm due to bending and tensile loading may be combined by summing their values (method of superposition).
The characteristic equation of a rigid center diaphragm loaded by a force for any magnitude of deflection is given by Equation (13-6). It is applicable for \((b/a)\) ratios greater than 0.05.

\[
F = \frac{\pi E}{a^2} \left[ \frac{h^3 y_o}{K_F} + h y_o^3 B \right]
\]  

(13-6)

Where:

\(F\) = Force applied to rigid disk of diaphragm, lb

\(E\) = Modulus of elasticity, lbs/in\(^2\)

\(a\) = Radius of diaphragm, in

\(h\) = Diaphragm thickness, in

\(y_o\) = Vertical deflection at center of diaphragm, in

\(K_F\) = Modified Stiffness Coefficient based on diaphragm bending loads,

\[
K_F = \frac{3 \left(1 - \eta^2\right)}{\pi} \left[ \frac{c^2 - 1 - \ln^2 c}{4 c^2 - c^2 - 1} \right]
\]  

(13-7)

\(B\) = Stiffness coefficient based on diaphragm tensile loading, as follows:

\[
B = \frac{7 - \eta \left( \frac{1 + b^2}{a^2} + \frac{b^4}{a^4} \right) + \left(3 - \eta\right)^2 \frac{b^2}{(1 + \eta) a^2}}{(1 - \eta) \left(1 - \frac{b^4}{a^4}\right) \left(1 - \frac{b^2}{a^2}\right)^2}
\]  

(13-8)

\(\eta\) = Poisson’s ratio

\(c\) = Ratio of radii (diaphragm-to-disk), \(a/b\), in/in

\(b\) = Radius of rigid center plate of diaphragm, in

The maximum radial stress for a force-loaded diaphragm with rigid center occurs at the inner perimeter of the diaphragm \((b)\):

\[
\sigma_r = \frac{F K_F B_F}{2 \pi h^2}
\]  

(13-9)
Where: \( \sigma_r \) = Maximum radial stress, lbs/in\(^2\)

\( B_F \) = Modified stiffness coefficient, based on diaphragm tensile loading

\[
B_F = \frac{2}{1 - \eta^2} \frac{c^2 \left(2c^2 \ln c - c^2 + 1\right)}{(c^2 - 1)^2 - 4c^2 \ln^2 c} \quad (13-10)
\]

At equilibrium, where the force transmitted by the push rod in Figure 13.3 generates a maximum pressure in the chamber above the diaphragm (i.e., the rod has completed its stroke), a balance of forces in the vertical direction is established.

If the increased performance of a compressor is to be evaluated and the change in shaft power requirements are known, the following equation, in combination with Equation (13-9), can be used to evaluate the maximum induced stress in the diaphragm:

\[
\sigma_r = \frac{396,000 \text{ hp} K_F B_F}{2\pi L \omega h^2} \quad (13-11)
\]

Where: \( \text{hp} \) = Shaft output horsepower

\( L \) = Offset of eccentric shaft, in

\( \omega \) = Output shaft speed, rpm

The maximum stress is calculated from Equation (13-11) for the compressor rated condition. Then the maximum stress for the actual operating condition is calculated in the same manner.

Empirical studies show that for moderate to high strains, a mechanical tearing of rubber, referred to as "mechanical-oxidative cut growth", can be the mechanism of failure for rubber diaphragms. The cut growth may greatly increase in the presence of oxygen. For this mode of failure, the fatigue life is inversely proportional to a power of the strain energy of the rubber. The strain energy is a characteristic of each type of rubber, and in turn, inversely proportional to the strain experienced by rubber under cyclic stressing. Figure 13.4 shows the stress-strain relationship for natural rubber compounds. Unlike many other engineering materials, rubber can be manufactured with a wide range of elastic moduli. Stiffness variations can be attained with no dimensional changes by varying the incorporation of fillers (reinforcing carbon blacks). This "hardness" variable is essentially a measurement of reversible elastic penetration (International Rubber Hardness Degrees or IRHD).
The stress developed in a rubber diaphragm can be calculated from Equation (13-9). Although rubber is flexible, (i.e., has low elastic and shear moduli), it is highly incompressible in bulk and its Poisson’s ratio, $\eta$, can be approximated as 0.5. This will facilitate the use of these equations. From the stress calculated, Figures 13.4 and 13.7 provide a corresponding load multiplying factor, $C_P$.

The value for strain obtained from Figure 13.4 must exceed 75%. Below this strain, the mechanical-oxidative cut growth mode of failure does not apply, and the $C_P$ factor becomes 1.0.

![Figure 13.4 Tensile Stress-Strain Curves for Four Natural Rubber Compounds of Different Hardness (Ref 31)](image)

13.5.2 Atmospheric Contaminant Multiplying Factor

The very small concentration of ozone in the atmosphere, normally a few parts per hundred million at ground level, may cause cracking in strained rubber components. Under cyclic conditions of strain below about 75%, ozone cut growth is the major factor in determining fatigue life.
Experimental data presented in Figure 13.8 illustrates that fatigue life is proportional to the concentration of ozone. The stress developed in a rubber diaphragm can be calculated from Equation (13-9). Poisson’s ratio, $\eta$, can be equated to 0.5.

Table 13-2 can be used to determine the strain by dividing the value of stress obtained from Equation (13-9) by Young’s modulus. Figure 13.8 and this strain value are then used to determine the contaminant air performance multiplying factor, $C_{AC}$.

### Table 13-2. Hardness and Elastic Moduli

<table>
<thead>
<tr>
<th>HARDNESS, IRHD</th>
<th>YOUNG’S MODULUS, E, lb/in$^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>130</td>
</tr>
<tr>
<td>35</td>
<td>168</td>
</tr>
<tr>
<td>40</td>
<td>213</td>
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<td>256</td>
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<td>830</td>
</tr>
<tr>
<td>70</td>
<td>1040</td>
</tr>
<tr>
<td>75</td>
<td>1340</td>
</tr>
</tbody>
</table>

#### 13.5.2.1 Adjustment to Atmospheric Contaminant Multiplying Factor

In ozone-dominant failure potentials, the use of chemical anti-ozonant (coating) on the surface of the rubber diaphragm can reduce crack growth by a factor of 3. If a coating is used, the multiplying factor, $C_{AC}$, obtained from Figure 13.8 should be multiplied by 1/3.
13.5.3 **Liquid Contaminant Multiplying Factor**

Water absorption does not usually cause any significant deterioration of rubber, but the absorption of oil and solvents cause rubber to swell with a consequent deterioration in certain properties. Thin components can be expected to fail rapidly if the major surfaces are exposed to oil. Thick components are effectively protected by their bulk. Such components can last many years in an oily environment. Diffusion theory predicts that the mass of liquid absorbed per unit area of rubber (in the early stages of swelling) is proportional to the square root of the time taken for the absorption.

The rate of movement of the boundary between swollen and unswollen rubber is calculated from:

\[
PR = \frac{L}{\sqrt{t}}
\]  

(13-12)

Where: 
\( PR \) = Penetration rate, in/sec\(^{0.5}\)

\( L \) = Depth of the swollen layer, in

\( t \) = Time that a given mass of liquid is absorbed by a given surface, sec

The failure rate for a rubber diaphragm is dependent on the presence of liquid contaminants and the viscosity of the liquid in contact with it. Typical penetration rates are shown in Figure 13.5. Figure 13.5 reveals that the penetration rate into natural rubber decreases as the viscosity of the swelling liquid increases.

An adjustment for various types of diaphragm materials can be made using the multiplying factors presented in **Table 13-3**. These factors should be multiplied by the penetration rate obtained from Figure 13.5 prior to using the nomograph in **Figure 13.6**.

13.5.4 **Temperature Multiplying Factor**

The variations in ambient temperature commonly occurring in practice are unlikely to greatly affect fatigue behavior. Experiments over a range of -32 to 212 F indicate only a slight effect of temperature on the fatigue life of crystallizing natural rubber. In general, rubbers become weaker as the temperature is raised. There is a steady fall in strength up to a critical temperature at which an abrupt drop occurs. For natural rubber, this temperature is about 212 °F.

A temperature multiplying factor, \( C_T \), can be developed as follows:
For: $-32 \leq T \leq 212 \text{ F}$, $C_T = 1.0$

and for: $T > 212 \text{ F}$, $C_T = 6.7$

Figure 13.5 Effect of Liquid Viscosity on the Penetration Rate of Liquids into Natural Rubber
Table 13-3. Contaminant Adjustment Factor for Various Diaphragm Materials

<table>
<thead>
<tr>
<th>RUBBER</th>
<th>X</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural</td>
<td>1.0</td>
</tr>
<tr>
<td>Cis polybutadiene</td>
<td>1.3</td>
</tr>
<tr>
<td>Butyl</td>
<td>0.7</td>
</tr>
<tr>
<td>SBR</td>
<td>0.7</td>
</tr>
<tr>
<td>Neoprene WRT</td>
<td>0.4</td>
</tr>
<tr>
<td>Nitrile (38% acrylonitrile)</td>
<td>0.1</td>
</tr>
<tr>
<td>Metal</td>
<td>0.001</td>
</tr>
</tbody>
</table>

Figure 13.6 Nomograph for the Determination of Liquid Contaminant Multiplying Factor, $C_{LC}$
For $S \leq 75\%$: $C_P = 1.0$

For $S > 75\%$: $C_P = \left(\frac{S}{75}\right)^{1.8}$

Where: $S = \text{Strain, } \%$

**Figure 13.7 Axial Loading Multiplying Factor as a Function of Strain**
For ozone 0.3 pphm:  \[ C_{AC} = \left( \frac{S}{10} \right)^{1.1} \]

For ozone 7.5 pphm:  \[ C_{AC} = \left( \frac{S}{10} \right)^2 \]

Where S = Strain, %

**Figure 13.8 Atmospheric Contaminant Multiplying Factor**
Table 13-4. Failure Rate for Fluid Drivers ($\lambda_{FD}$)

<table>
<thead>
<tr>
<th>COMPRESSOR TYPE</th>
<th>FLUID DRIVER MODE</th>
<th>MODEL TYPE</th>
<th>BASE RATE*</th>
</tr>
</thead>
<tbody>
<tr>
<td>Centrifugal</td>
<td>Axial flow</td>
<td>------</td>
<td>0.10</td>
</tr>
<tr>
<td></td>
<td>Radial flow</td>
<td>------</td>
<td>0.10</td>
</tr>
<tr>
<td>Positive</td>
<td>Reciprocating</td>
<td>Single acting</td>
<td>1.00</td>
</tr>
<tr>
<td>Displacement</td>
<td></td>
<td>Double acting</td>
<td>1.00</td>
</tr>
<tr>
<td>Reciprocating</td>
<td>Piston</td>
<td>1.18</td>
<td></td>
</tr>
<tr>
<td>Reciprocating</td>
<td>Labyrinth</td>
<td>0.60</td>
<td></td>
</tr>
<tr>
<td>Reciprocating</td>
<td>Diaphragm</td>
<td>0.58</td>
<td></td>
</tr>
<tr>
<td>Rotary</td>
<td>Vane</td>
<td>0.40</td>
<td></td>
</tr>
<tr>
<td>Rotary</td>
<td>Screw</td>
<td>0.60</td>
<td></td>
</tr>
<tr>
<td>Rotary</td>
<td>Lobe</td>
<td>0.45</td>
<td></td>
</tr>
<tr>
<td>Rotary</td>
<td>Liquid Ring</td>
<td>1.05</td>
<td></td>
</tr>
</tbody>
</table>

* Failures/million hours of operation

13.6 REFERENCES


78. CDNSWC, "Interim Reliability Report on the MC-2A Compressor Unit", January,
1992


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14.1 INTRODUCTION

Electric motors play a very important part in supplying power for all types of domestic and industrial applications. Their versatility, dependability, and economy of operation cannot be equaled by any other type of a power unit. Many types of motors are available and are, therefore, classified in various ways. There are general purpose, special purpose, and definite purpose types of motors. Motors are also classified according to the type of electricity they require; a motor may operate on direct current (DC) or alternating current (AC). If AC, the motor may be of a single or polyphase design.

This chapter contains failure rate models that apply to all electric motors which can be used to support the development of mechanical equipment and provide a reliability estimate for a new design, proposed design modification, or application other than verified specification parameters. The models are intended to focus attention on further design analysis which should be accomplished to assure the allocated reliability of the motor in its intended operational environment.
14.2 CHARACTERISTICS OF ELECTRIC MOTORS

14.2.1 Types of DC Motors

DC motors are classified as either series-wound, shunt-wound, or compound-wound. In the series-wound motor, field windings which are fixed to the stator frame, and the armature windings which are placed around the rotor, are connected in series so that all current that passes through the field windings also passes through the armature windings. In the shunt-wound motor, the armature and field are both connected across the main power supply (in parallel) so that the armature and field currents are separate. The compound-wound motor has both the series and shunt field windings. These may be connected so that the currents are flowing the same direction in both windings, called "cumulative compounding", or so that the currents are flowing in opposite directions, called "differential compounding".

14.2.2 Types of Polyphase AC Motors

The most extensively used polyphase motors are the induction type. The "squirrel cage" induction motor has a wound stator connected to an external source of AC power and a laminated steel core rotor with heavy aluminum or copper conductors set into the core around its periphery while being parallel to its axis. These conductors are connected together at each end of the rotor by a heavy ring, providing closed paths for currents induced in the rotor to circulate. The rotor windings are not connected to the power supply.

The wound-rotor type of induction motor has a squirrel cage and a series of coils set into the rotor which are connected through slip-rings to external variable resistors. By varying the resistance of the wound-rotor circuits, the amount of current flowing in the circuits, and therefore the speed of the motor, can be controlled. Induction motors are manufactured with a wide range of speed and torque characteristics.

The synchronous motor is the other type of polyphase AC motor. Unlike the induction motor, the rotor of the synchronous motor is connected to a DC supply which provides a field that rotates in step with the AC field in the stator. The synchronous motor operates at a constant speed throughout its entire load range, after having been brought up to this synchronous speed. This speed is governed by the frequency of the power supply and the number of poles in the rotor.

14.2.3 Types of Single-Phase AC Motors

Most of the single-phase AC motors are induction motors distinguished by different arrangements for starting. Single-phase motors are used in sizes up to about 7 1/2 horsepower for heavy starting duty, chiefly in home and commercial appliances for which polyphase power is not available.
The series wound single-phase motor has a rotor winding in series with the stator winding as in the series-wound DC motor. Since this motor may also be operated on direct-current, it is called a "universal motor". The series wound motor has a high starting torque and is used in vacuum cleaners, sewing machines, and portable tools. In the capacitor-start single-phase motor, an auxiliary winding in the stator is connected in series with a capacitor and a centrifugal switch. During the starting and accelerating period the motor operates as a two-phase induction motor. At about two-thirds full-load speed, the auxiliary circuit is disconnected by the switch and the motor then runs as a single phase induction motor.

In the capacitor-start, capacitor-run motor, the auxiliary circuit is arranged to provide high effective capacity for high starting torque and to remain connected to the line, but with reduced capacity during the running period. In the single-value capacitor or capacitor split-phase motor, a relatively small continuously-rated capacitor is permanently connected in one of the two stator windings and the motor both starts and runs like a two-phase motor.

In the repulsion-start single-phase motor, a drum-wound rotor circuit is connected to a commutator with a pair of short-circuited brushes set so that the magnetic axis of the rotor winding is inclined to the magnetic axis of the stator winding. The current flowing in this rotor circuit reacts with the field to produce a starting and accelerating torque. At about two-thirds full load speed the brushes are lifted, the commutator is short circuited and the motor runs as a single-phase squirrel-cage motor. The repulsion motor employs a repulsion winding on the rotor for both starting and running. The repulsion-induction motor has an outer winding on the rotor acting as a repulsion winding and an inner squirrel cage winding. As the motor comes up to speed, the induced rotor current partially shifts from the repulsion winding to the squirrel cage winding and the motor runs partly as an induction motor.

In the split-phase motor, an auxiliary winding in the stator is used for starting with either a resistance connected in series with the auxiliary winding (resistance-start) or a reactor in series with the main winding (reactor-start). The split-phase motor is used in refrigerators, air conditioners, freezers, and other compressors involving high starting loads.

### 14.3 ELECTRIC MOTOR FAILURE MODES

The most prominent failure mode for a motor is shorting of the motor winding. Typical failure modes and their failure causes and effects are listed in Table 14-1. For additional information on individual parts of the motor, the particular chapter for that part should be reviewed as shown below:

1. Bearings (See Section 7.4)
2. Windings (See Table 14.1 below and Section 14.5)
3. Brushes (See Table 14.1 below)
4. Armature (shaft) (See Section 20.2)
5. Stator Housing (casing) (See Table 14.1 below)

### Table 14-1. Electric Motor Failure Modes

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Open winding</td>
<td>- Excessively high temperature</td>
<td>- Motor won’t run</td>
</tr>
<tr>
<td>- Shorted winding</td>
<td></td>
<td>- Sparking at brushes</td>
</tr>
<tr>
<td>- Worn bearing:</td>
<td>- Poor lubrication</td>
<td>- Noisy</td>
</tr>
<tr>
<td>-- spalling</td>
<td>- Contamination</td>
<td>- Heat build-up</td>
</tr>
<tr>
<td>--creeping or spin</td>
<td>- Overloading or high temperature</td>
<td>- Armature rubbing stator</td>
</tr>
<tr>
<td>- Cracked housing</td>
<td>- Fatigue</td>
<td>- Seized</td>
</tr>
<tr>
<td>- Excessively high temperature</td>
<td>- External shock</td>
<td>- Shorted or seized</td>
</tr>
<tr>
<td>- Overloading or high temperature</td>
<td>- Vibration</td>
<td></td>
</tr>
<tr>
<td>- Motor won’t run</td>
<td></td>
<td></td>
</tr>
<tr>
<td>- Sparks at brushes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>- Worn bearing:</td>
<td>- Fatigue</td>
<td>- Overloading or high temperature</td>
</tr>
<tr>
<td>-- spalling</td>
<td>- External shock</td>
<td>- Motor won’t run</td>
</tr>
<tr>
<td>--creeping or spin</td>
<td>- Vibration</td>
<td>- Sparking at brushes</td>
</tr>
<tr>
<td>- Contamination</td>
<td>- Overloading or high temperature</td>
<td>- Noisy</td>
</tr>
<tr>
<td>- Overloading or high temperature</td>
<td>- Vibration</td>
<td>- Heat build-up</td>
</tr>
<tr>
<td>- Motor won’t run</td>
<td></td>
<td>- Armature rubbing stator</td>
</tr>
<tr>
<td>- Sparks at brushes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>- Worn sleeve bearing</td>
<td>- Excessive load (belt tension)</td>
<td>- Seized</td>
</tr>
<tr>
<td>- Worn sleeve bearing</td>
<td>- Frequent starts and stops under heavy loads</td>
<td>- Noisy</td>
</tr>
<tr>
<td>- Poor lubrication</td>
<td>- Improper maintenance</td>
<td>- Heat build-up</td>
</tr>
<tr>
<td>- Improper contact pressure</td>
<td>- Contamination</td>
<td>- Armature rubbing stator</td>
</tr>
<tr>
<td>- Excessive sparking</td>
<td>- High temperature</td>
<td></td>
</tr>
<tr>
<td>- Motor runs too fast or too slow under load</td>
<td>- Proper maintenance</td>
<td></td>
</tr>
<tr>
<td>- High temperature</td>
<td>- Low atmospheric humidity</td>
<td></td>
</tr>
<tr>
<td>- Improper contact pressure</td>
<td>- Proper maintenance</td>
<td></td>
</tr>
<tr>
<td>- Motor runs too fast or too slow under load</td>
<td>- Improper contact pressure</td>
<td></td>
</tr>
<tr>
<td>- Improper contact pressure</td>
<td>- Proper maintenance</td>
<td></td>
</tr>
<tr>
<td>- Motor runs too fast or too slow under load</td>
<td>- High temperature</td>
<td></td>
</tr>
<tr>
<td>- Improper contact pressure</td>
<td>- Improper maintenance</td>
<td></td>
</tr>
<tr>
<td>- Motor runs too fast or too slow under load</td>
<td>- Proper maintenance</td>
<td></td>
</tr>
<tr>
<td>- Improper contact pressure</td>
<td>- Improper maintenance</td>
<td></td>
</tr>
</tbody>
</table>

Additional details of failure modes for those components of a motor such as bearings and shafts are included in the applicable chapters of this Handbook.

### 14.4 MODEL DEVELOPMENT

The failure rate model included in this section is based upon identified failure modes of individual parts. The model developed is based on a fractional or integral
horsepower AC type motor, although it will be general enough to be applied to most motors.

The reliability of an electric motor is dependent upon the reliability of its parts, which may include: bearings, electrical windings, armature/shaft, housing, and brushes. Failure mechanisms resulting in part degradation and failure rate distribution (as a function of time) are considered to be independent in each failure rate model. The total motor system failure rate is the sum of the failure rates of each of the parts in the system:

$$\lambda_M = \lambda_{BE} + \lambda_{WI} + \lambda_{BS} + \lambda_{AS} + \lambda_{ST} + \lambda_{GR}$$

Where:
- \(\lambda_M\) = Total failure rate for the motor system, failures/million hours
- \(\lambda_{BE}\) = Failure rate of bearings, failures/million hours (See Chapter 7)
- \(\lambda_{WI}\) = Failure rate of electric motor windings, failures/million hours (See Section 14.5)
- \(\lambda_{BS}\) = Failure rate of brushes, 3.2 failures/million hours/brush (Reference 68)
- \(\lambda_{AS}\) = Failure rate of the armature shaft, failures/million hours (See Chapter 20, Section 20.4)
- \(\lambda_{ST}\) = Failure rate of the stator housing, 0.001 failures/million hours (Reference 68)
- \(\lambda_{GR}\) = Failure rate of gears, failures/million hours (See Chapter 8)

### 14.5 FAILURE RATE MODEL FOR MOTOR WINDINGS

The life expectancy of a motor winding is primarily dependant on its operating temperature with respect to the permitted temperature rise of the winding. The temperature rise of the winding (and the insulation materials) is a function of the design of the motor. The insulation materials age over time and this aging process is directly related to temperature. Eventually, the materials lose their insulating properties and break down causing one or more short circuits.

Temperature rise occurs in a motor due to the losses that occur in the motor, normally copper and iron losses. The temperature inside the motor will depend on how effectively this heat can be removed by the cooling system of the motor. The difference between the internal and external temperatures is dependent on the thermal gradient and this difference is normally quite low.
The electric motor windings failure rate, $\lambda_{WI}$, is derived by Equation (14-2):

$$\lambda_{WI} = \lambda_{WI,B} \cdot C_T \cdot C_V \cdot C_{alt} \quad (14-2)$$

Where:

- $\lambda_{WI,B}$ = Base failure rate of the electric motor windings, failures/million hours (See Section 14.5.1)
- $C_T$ = Multiplying factor which considers the effects of ambient temperature on the base failure rate (See Section 14.5.2 and Figure 14.1)
- $C_V$ = Multiplying factor which considers the effects of electrical source voltage variations (See Section 14.5.3)
- $C_{alt}$ = Multiplying factor which considers the effects of operation at extreme elevations (See Section 14.5.4 and Table 14-3)

### 14.5.1 Base Failure Rate

$\lambda_{WI,B}$ is the base failure rate of the specific motor winding as supplied by the motor manufacturer. The winding will usually be specified in terms of expected life. The base failure rate is then:

$$\lambda_{WI,B} = \frac{1.0 \times 10^6}{L_i} \quad (14-3)$$

Where: $L_i$ = Expected winding life, hours

If a manufacturer's winding life is not available, a winding life of 20,000 hours can be expected from most manufacturers (References 28 and 89). The multiplying factors for Equation (14-2) are described in the following paragraphs.

### 14.5.2 Temperature Multiplying Factor

Heat is the primary limiting factor of motor windings. Heat causes the windings to age and deteriorate, so after time they break down and lose their insulation quality. When this happens, the related electrical components "short" and the motor burns out.
The manufacturer’s rating of a motor based on insulation and expected life is provided in 25° increments. The temperature rating for each class of insulation is defined as the maximum temperature at which the insulation can be operated to yield the rated winding life. The temperature rating for the various classes of insulation is shown in Table 14-2.

<table>
<thead>
<tr>
<th>Insulation Class</th>
<th>Temperature Rating</th>
<th>Maximum Ambient Temperature</th>
<th>Anticipated Temperature Rise</th>
<th>Hot Spot Allowance</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>105° C</td>
<td>40° C</td>
<td>60° C</td>
<td>5° C</td>
</tr>
<tr>
<td>B</td>
<td>130° C</td>
<td>40° C</td>
<td>85° C</td>
<td>5° C</td>
</tr>
<tr>
<td>F</td>
<td>155° C</td>
<td>40° C</td>
<td>110° C</td>
<td>5° C</td>
</tr>
<tr>
<td>H</td>
<td>180° C</td>
<td>40° C</td>
<td>135° C</td>
<td>5° C</td>
</tr>
</tbody>
</table>

Under normal operating conditions, the insulation material used in the windings of electric motors is generally reliable, thereby making the windings themselves a reliable component. The life of any given insulation material depends on the degree of heat to which it is exposed.

The winding temperature is determined by measuring both the ambient and the hot temperature resistances of the windings. The resistance measurement gives an average temperature which is more representative than spot measurements with a thermometer. This method has become standard because of the dimensional restrictions of so many motor designs, which prevent the use of thermometers.

The equation for determining the motor winding temperature from resistance readings is as follows:

$$T_R = \frac{R_H - R_C}{R_C} (235 + T_C)$$  \hspace{1cm} (14-4)

Where:  
- $T_R = \text{Temperature Rise, °C}$
- $R_H = \text{Hot winding resistance, ohms}$
- $R_C = \text{Cold winding resistance, ohms}$
- $T_C = \text{Ambient temperature, °C}$
The winding life increases by a factor of 2 for every 10 degree of rating. Therefore if manufacturer provides a motor with a insulation class F for a B class environment, the motor can be expected to last twice as long.

The correction factor for the motor winding temperature is given by:

\[
C_r = k \times 10^{\frac{2357}{T_r + 273} - \frac{1}{T_o + 273}}
\] (14-5)

Where: 
- \(T_r\) = Temperature rating of windings, °C (See Table 14-2)
- \(T_o\) = Internal motor temperature during operation, °C
- \(k\) = 1.0

Figure 14.1 shows the effect of temperature on failure rate for various classes of motors.

14.5.3 Voltage Multiplying Factor

The motor horsepower rating on the nameplate may not necessarily indicate the motor's maximum capacity. The motor is often designed with extra capacity built in to allow for variations. A motor will operate successfully when the variation in voltage does not exceed ±10% of normal. A failure rate multiplying factor can be established for those situations when the actual voltage exceeds rated voltage:

For \(V_A > V_R\):

\[
C_V = 1.0 + 0.5 \left( \frac{V_A - V_R}{V_R} \right)
\] (14-6)

Where: 
- \(V_R\) = Rated Voltage
- \(V_A\) = Actual Voltage

For \(V_A \leq V_R\):

\[
C_V = 1.0
\] (14-7)
14.5.4 Altitude Multiplying Factor

The influence of altitude on the life of a fan-cooled motor may be tabulated based on a 50% reduction in life for every $10^\circ$C increase in sea level motor temperature rise. Table 14-3 is a tabulation of failure rate multiplying factor, $C_{alt}$, for altitude/temperature conditions applicable to fan-cooled motors which are not enclosed. For totally enclosed motors, altitudes to 60,000 feet will not influence life as compared to sea level and $C_{alt}$, in this case, will be equal to 1.0.
\[ C_T = k \times 10^{\frac{2357}{T_r + 273} - \frac{1}{T_o + 273}} \]

Where:

- \( T_r \) = Temperature rating of windings, °C (See Table 14-2)
- \( T_o \) = Internal motor temperature during operation, °C
- \( k \) = 1.0

**Figure 14.1 Temperature Multiplying Factor, \( C_T \)**
Table 14-3. Multiplying Factor $C_{alt}$ for the Influence of Altitude on Motor Life for Fan-Cooled Motors

<table>
<thead>
<tr>
<th>ALTITUDE (ft x 1000)</th>
<th>SEA LEVEL MOTOR TEMPERATURE RISE</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20°C</td>
</tr>
<tr>
<td>Sea level</td>
<td>1.0</td>
</tr>
<tr>
<td>25</td>
<td>1.0</td>
</tr>
<tr>
<td>30</td>
<td>1.0</td>
</tr>
<tr>
<td>40</td>
<td>1.0</td>
</tr>
<tr>
<td>50</td>
<td>4.0</td>
</tr>
<tr>
<td>60</td>
<td>16.0</td>
</tr>
</tbody>
</table>

14.6 REFERENCES

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment"


15.1 INTRODUCTION

A hydraulic accumulator is a device for energy storage or a reservoir for pressure storage. The function of an accumulator is to hold the pressurized potential energy in the form of a spring or compressed gas or by raised weight that is used to exert a force against an incompressible fluid. An accumulator is a device used to store energy in such applications as:

- Fluid supply
- Pump delivery pulsation damping
- System pressure surge damping and shock suppression
- Stabilization of pressure fluctuations
- Leakage and thermal expansion compensation
- Emergency and standby power source

In a hydraulic system the energy is stored as a fluid under pressure and often used to smooth out the delivery flow of pumps. A pump generates the required power to be
used or stored in a hydraulic system. The pump may deliver this power in a pulsating flow which can produce pulsations detrimental to a high pressure system. An accumulator properly located in the hydraulic system will substantially cushion these pressure pulsations. In many hydraulic power applications the valve, actuator or other driven component stops or closes suddenly, creating a pressure wave that travels back through the system. This shock wave can develop peak pressures several times greater than normal working pressures causing objectionable noise or pump failure. An accumulator properly located in the hydraulic system will minimize this shock wave.

An accumulator design may include one or more valves, bladder, bleed plug, a high strength shell, piston, and fluid port. Typical accumulator designs are shown in Figure 15.1. A dead load accumulator is comprised of a single acting vertical cylinder which raises a heavy load mass.

![Figure 15.1 Typical Accumulator Designs](image)

A dead load accumulator can be designed for large volumes but correspondingly heavy weights are needed resulting in a large physical size. The advantage of this type of accumulator design is the constant discharge pressure, whereas all other types exhibit a variation in pressure with respect to volume of fluid stored.

A spring loaded accumulator contains a spring which moves within a cylinder. As the volume of fluid in the accumulator is increased, the spring is compressed and the spring force is increased. The minimum pressure in the accumulator depends on the designed spring preload. The piston stroke and, therefore, the volume of fluid which can be stored is limited by the physical characteristics of the spring.

A gas loaded accumulator is designed to utilize a compressed gas such as nitrogen or air to pressurize the stored fluid. The accumulator may be a piston type, diaphragm type or bladder type to separate the noncompressable fluid and gas. Gas loaded accumulators can be very large. As discussed in the next section, accumulators are usually designed to be operated in the vertical position. The fluid pressure as a function...
of fluid volume in a gas loaded accumulator depends upon many factors such as the
gas being used, the temperature of the gas and its pressure-volume characteristics.

The three types of gas accumulators are shown in figure 15.2.

![Diagram of gas loaded accumulators]

Figure 15.2  Gas Loaded Accumulator Configurations

15.2 FAILURE MODES

Accumulator failure is often defined as the inability to accept and exhaust a
specified amount of fluid when operating over a specific system pressure range. Failure
often results from unwanted loss or gain of precharge pressure. Correct precharge
pressure is probably the single most important fact in prolonging the life of the
accumulator.

In any type of accumulator utilizing a piston the cylinder bore has to be machined,
and wear will occur between the piston and cylinder body. Seals are built into the piston
and these are subject to wear and leakage. Depending on the accumulator application,
response time may be a factor. The response of the dead load accumulator will be
somewhat slow due to the high inertia of the load and piston. The response of spring
loaded accumulators will depend on the age of the spring and its modulus of rigidity. A
response of a piston type accumulator will be adversely affected by the inertia of the
piston and the effect of seal stiction. Typical accumulator failure modes are listed in
Table 15-1.
Table 15-1. Failure Modes for an Accumulator

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seal Leakage</td>
<td>- Embrittlement</td>
<td>- Leakage past piston</td>
</tr>
<tr>
<td></td>
<td>- Wear</td>
<td>- Internal leakage at valve</td>
</tr>
<tr>
<td></td>
<td>- Distortion</td>
<td>- External leakage</td>
</tr>
<tr>
<td></td>
<td>- Incompatibility with medium</td>
<td></td>
</tr>
<tr>
<td>Worn cylinder bore or piston</td>
<td>- Contaminants</td>
<td>- Poor system response</td>
</tr>
<tr>
<td>surface</td>
<td>- Interaction with fluid medium</td>
<td>- Leakage</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Loss of pressure</td>
</tr>
<tr>
<td>Loss of spring tension</td>
<td>- Corrosion</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Fracture</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Spring misalignment</td>
<td></td>
</tr>
<tr>
<td>Piston stiction</td>
<td>- Surface wear</td>
<td>- Poor system response</td>
</tr>
<tr>
<td></td>
<td>- Corrosion</td>
<td></td>
</tr>
<tr>
<td>Loss of pressure</td>
<td>- Ruptured gas bag</td>
<td>- Poor system response</td>
</tr>
<tr>
<td></td>
<td>- External leakage</td>
<td>- External component damage</td>
</tr>
<tr>
<td>Leakage of charge gas into</td>
<td>- Leakage past piston or bag</td>
<td>- Poor system response</td>
</tr>
<tr>
<td>fluid system</td>
<td>- Sudden discharge of fluid from</td>
<td>- Contaminated fluid system</td>
</tr>
<tr>
<td></td>
<td>accumulator</td>
<td></td>
</tr>
<tr>
<td>Inoperative accumulator</td>
<td>- Jammed output valve</td>
<td>- External component damage</td>
</tr>
</tbody>
</table>

If a piston type accumulator is used inclined to a vertical position, the rate of wear will be increased due to the additional side load. Failure of a piston type accumulator tends to be gradual caused by deterioration of piston seals and wear in the cylinder bore. Failure of a bag type gas loaded accumulator will be more sudden caused by the rupturing of the bag or diaphragm. The failure rate of a bag type accumulator may also depend on its physical characteristics.

A spring loaded accumulator must be evaluated closely for reliability to verify compatibility between the spring material and the surrounding medium. Any leakage past the seal could have a deteriorating effect on the spring material and its compression properties or fatigue life.

One of the main applications of an accumulator is the damping of fluid system pulsations or surges. The system effects of these pulsations must be evaluated as part of any reliability analysis. In some applications the pulsations are unimportant as they
are partially smoothed by pipes upstream of the pump. A critical element of the reliability analysis is the effect of an accumulator on the probability of failure of other system components. For example, a failed valve assembly within the accumulator which prevents fluid discharge may not be immediately detected and damage to other components may occur due to pressure transients. Shock waves produced as a result of the sudden closing of a downstream valve, for example, travels through the system fluid to the far end of the system and a decompression wave is formed which travels back to the valve. These waves travel back and forth until the energy is expended. The more rapid the valve closure, the more severe the pressure transient generated. Without detection of an accumulator failure, severe degradation and damage to system components could be occurring without operator or maintainer knowledge.

### 15.3 FAILURE RATE MODEL

The failure rate of an accumulator will depend on several factors:

- Volumetric capacity
- Operating pressure
- Maximum flow rate

The failure rate of an accumulator is dependent on the sum of the failure rates of its component parts:

$$
\lambda_A = \lambda_{SE} + \lambda_{SP} + \lambda_{PC} + \lambda_{VA} + \lambda_{CW}
$$  \hspace{1cm} (15-1)

Where:
- \( \lambda_A \) = Total failure rate of accumulator, failures/million hours
- \( \lambda_{SE} \) = Failure rate of seals, failures/million hours (See Chapter 3)
- \( \lambda_{SP} \) = Failure rate of springs, failures/million hours (See Chapter 4)
- \( \lambda_{PC} \) = Failure rate of piston/cylinder interface, failures/million hours (See Chapter 9)
- \( \lambda_{VA} \) = Failure rate of control valve, failures/million hours (See Chapter 6)
- \( \lambda_{CW} \) = Failure rate of cylinder wall, failures/million hours (See Section 15.4 for thin walled cylinders and Section 15.5 for thick walled cylinders)
15.3.1 Seals

Specific failure modes of seals and procedures to determine their failure rates under different operating environments are discussed in Chapter 3. Of particular interest in the design evaluation of accumulators and other pressure vessels is the compatibility of the fluid medium and the seal material. The position of the accumulator in the fluid system must also be known to determine the side load on the piston and corresponding stress on the seal.

15.3.2 Springs

Specific failure modes of springs and procedures to determine their failure rates under different operating environments are discussed in Chapter 4. For most accumulators the failure rate equations for static springs can be assumed. The reliability of a spring is very sensitive to corrosion and the compatibility of the fluid and spring material must be considered.

15.3.3 Piston and Cylinder

The wear rate of the piston surface and cylinder bore will be sensitive to the position of the accumulator in its operating environment. Tilting of the accumulator from its vertical position will alter the side load of the piston. This parameter and others affecting the reliability of the piston/cylinder are included in the reliability equations contained in Chapter 9.

15.3.4 Valves

The reliability of valve assemblies which may be contained within the accumulator is determined using the equations contained in Chapter 6. One particular failure mode to be considered in the design evaluation is the possibility of a sudden discharge of fluid causing the output valve to operate without fluid and creating an air lock.

15.3.5 Structural Considerations

The fluid contained within an accumulator under pressure creates stresses in the walls as shown in Figure 15.3. The state of stress is triaxial. A longitudinal or meridional stress acts parallel to the meridian; a circumferential, or hoop stress acts parallel to the circumference; and a radial stress acts outward at the surface. If the walls of the accumulator are relatively thin (thickness \( t \) is less than one-tenth the radius \( r \)) and of uniform shape, longitudinal and circumferential stresses will be uniform throughout the thickness of the wall and the radial stress, although varying from zero at the outside surface to a value equal to the internal pressure at the inside surface can be considered negligible. Section 15.4 provides equations for determining the stress levels of thin walled pressure vessels and Section 15.5 provides equations for determining the stress levels of thick walled pressure vessels.
15.4 THIN WALL CYLINDERS

The shell thickness is designed to keep the maximum stresses below the yield strength of the material. The design thickness is the minimum required thickness computed by code formula plus an allowance for corrosion.

The walls of the accumulator will tend to expand in the radial direction when pressurized, causing the walls to stretch circumferentially. As a result of this radial expansion, stresses will occur acting in the circumferential direction. For thin walled cylinders, the circumferential strains are approximately the same at the inside and outside of the cylinder. Consequently, the circumferential stresses will be very nearly uniform throughout the wall thickness. The circumferential stresses can be related to the internal pressure of the accumulator by considering the equilibrium of a half cylinder shown in Figure 15.2. For equilibrium to occur, the resistive force due to circumferential stress acting on the cylinder wall must equal the force acting on accumulator wall as a result of applied fluid pressure. This equilibrium is shown with the following equation (Reference 38):

\[ F_c = 2\pi \sigma_c t L - 2\pi P r_i L = 0 \]  

(15-2)

Where:

- \( F_c \) = Circumferential force, lbs
- \( P \) = Internal pressure, lbs/in\(^2\)
- \( r_i \) = Internal radius, in
\[ L = \text{Cylinder length, in} \]
\[ \sigma_c = \text{Circumferential stress, lbs/in}^2 \]
\[ t = \text{Wall thickness, in} \]

Solving for \( \sigma_c \), the circumferential stress, in the cylinder results in the following equation:

\[ \sigma_c = \frac{Pr_i}{t} \]  \hspace{1cm} (15-3)

Similarly, the equilibrium of forces in the longitudinal direction provides the following equation:

\[ F_l = \pi P \frac{r_i^2}{2} - 2 \pi r_i t \sigma_l \]  \hspace{1cm} (15-4)

Where: \( F_l = \text{Longitudinal force, lbs} \)
\( \sigma_l = \text{Longitudinal stress, lbs/in}^2 \)

and the corresponding longitudinal stress is:

\[ \sigma_l = \frac{Pr_i}{2t} \]  \hspace{1cm} (15-5)

The effects of end plates and joints on the accumulator are a reduction in strength of the accumulator due to riveted joints, welding and other fabrication techniques. This reduction is accounted for by including a joint efficiency parameter in the circumferential and longitudinal stress equations:

\[ \eta = \frac{\text{minimum strength of joint}}{\text{strength of solid material}} \]

Where: \( \eta = \text{Joint efficiency parameter} \)
The addition of $\eta_c$ as a circumferential joint efficiency parameter provides the following equation:

$$\sigma_c = \frac{Pr_i}{t\eta_c}$$  \hspace{1cm} (15-6)

and the addition of $\eta_l$ as a longitudinal joint efficiency factor provides the following equation:

$$\sigma_l = \frac{Pr_i}{2t\eta_l}$$  \hspace{1cm} (15-7)

The relative strength (efficiency) of a joint depends upon its design and type of joint. Table 15-2 provides values of $\eta_c$ and $\eta_l$ based on joint efficiencies that may be expected in the various types of joints if they are well designed. The actual value for the efficiency parameters, $\eta_c$ and $\eta_l$, from Table 15-2 can be adjusted depending on the confidence level in manufacturing techniques and quality control.

Table 15-2. Approximate Efficiencies of Joints \textbf{(Ref. 57, 58)}

<table>
<thead>
<tr>
<th>TYPE OF JOINT</th>
<th>DESIGN (Number of Rows)</th>
<th>$\eta_c$, $\eta_l$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Riveted Lap Joint</td>
<td>Single</td>
<td>0.55</td>
</tr>
<tr>
<td></td>
<td>Double</td>
<td>0.65</td>
</tr>
<tr>
<td></td>
<td>Triple</td>
<td>0.75</td>
</tr>
<tr>
<td>Riveted Butt Joint</td>
<td>Single</td>
<td>0.65</td>
</tr>
<tr>
<td></td>
<td>Double</td>
<td>0.75</td>
</tr>
<tr>
<td></td>
<td>Triple</td>
<td>0.85</td>
</tr>
<tr>
<td></td>
<td>Quadruple</td>
<td>0.90</td>
</tr>
<tr>
<td>Welded Butt Joint</td>
<td>Single</td>
<td>0.75</td>
</tr>
<tr>
<td></td>
<td>Double</td>
<td>0.85</td>
</tr>
</tbody>
</table>

The ends of the accumulator are often hemispheres. The internal pressure in a thin spherical shell will create two mutually perpendicular circumferential stresses of equal magnitude and a radial stress. Again, a thickness/radius ratio of less than 1/10 provides a minimal value of radial stress. The resistive force due to circumferential stress, $\sigma_c$,
acting on the accumulator wall to achieve equilibrium must equal the force on the hemisphere due to internal pressure, \( P \):

\[
F_{hc} = \pi P r_h^2 - 2 \pi r_h t_h \sigma_{hc} = 0
\]  \hspace{1cm} (15-8)

Where:

- \( F_{hc} \) = Circumferential force of the end section, lbs
- \( P \) = Internal fluid pressure, lbs/in\(^2\)
- \( r_h \) = Radius of the end section, in
- \( t_h \) = Thickness of the end section material, in
- \( \sigma_{hc} \) = Circumferential stress in the end section, lbs/in\(^2\)

Solving for \( \sigma_{hc} \), with the addition of a joint efficiency parameter, provides an equation for maximum stress at the hemispherical ends.

\[
\sigma_{hc} = \frac{P r_h}{2 t_h \eta_c}
\]  \hspace{1cm} (15-9)

It will be noted that for the same wall thickness, the spherical ends of the accumulator provide twice the strength. The hemispherical ends, therefore, are sometimes thinner than the cylindrical section. Equations for various shapes of accumulators can be found in standard textbooks.

The failure rate of the cylinder is determined by a base failure rate for the cylinder multiplied by stress level factors:

\[
\lambda_{CW} = \lambda_{CW,B} + \lambda_c + \lambda_l
\]  \hspace{1cm} (15-10)

Where:

- \( \lambda_{CW} \) = Failure rate of cylinder for use in Equation (15-1), failures/million hours
- \( \lambda_{CW,B} \) = Base failure rate of cylinder, 0.001 failures/million hours
- \( \lambda_c \) = Failure rate considering compressive stress (See Section 15.6)
- \( \lambda_l \) = Failure rate considering longitudinal stress (See Section 15.6)
15.5 THICK WALL CYLINDERS

If the wall thickness of the pressure vessel is more than one-tenth the radius, the circumferential and longitudinal stresses cannot be considered uniform throughout the thickness of the wall and the radial stress cannot be considered negligible. Reference 38 provides the equations for different shapes of thick walled containers:

$$\sigma_l = \frac{P r_i^2}{r_o^2 - r_i^2} \quad (15-11)$$

Where:
- $\sigma_l$ = Longitudinal stress, lbs/in$^2$
- $P$ = Internal pressure, lbs/in$^2$
- $r_i$ = Internal radius of the cylinder, in
- $r_o$ = External radius of the cylinder, in

$$\sigma_c = \frac{P r_i^2 (r_o^2 + r^2)}{r^2 (r_o^2 - r_i^2)} \quad (15-12)$$

Where:
- $\sigma_c$ = Circumferential stress, lbs/in$^2$
- $r$ = Average radius of the cylinder, in

$$\sigma_r = \frac{P r_i^2 (r_o^2 - r^2)}{r^2 (r_o^2 - r_i^2)} \quad (15-13)$$

Where:
- $\sigma_r$ = Radial stress, lbs/in$^2$

The failure rate of the cylinder is determined by a base failure rate for the cylinder multiplied by stress level factors:

$$\lambda_{CW} = \lambda_{CW,B} + \lambda_c + \lambda_l + \lambda_r \quad (15-14)$$

Where:
- $\lambda_{CW}$ = Failure rate of cylinder wall for use in Equation (15-1), failures/million hours
\[ \lambda_{CW,B} = \text{Base failure rate of cylinder, 0.001 failures/million hours} \]
\[ \lambda_c = \text{Failure rate considering circumferential stress (See Section 15.6)} \]
\[ \lambda_l = \text{Failure rate considering longitudinal stress (See Section 15.6)} \]
\[ \lambda_r = \text{Failure rate considering radial stress (See Section 15.6)} \]

15.6 FAILURE RATE CALCULATIONS

The structural aspects of the accumulator failure rate depend on the stress/strength relationships of the materials. The standard definition of reliability includes the probability that the strength random variable will exceed the stress random variable as shown in Figure 15.4.

\[ R = P(S > s) = P(S - s) > 0 \quad (15-15) \]

Where:
\[ R = \text{Reliability} \]
\[ P = \text{Probability} \]
\[ S = \text{Strength random variable} \]
\[ s = \text{Stress random variable} \]

Figure 15.4 Stress Strength Relationship
The stress variable includes any parameter that tends to introduce a failure of the accumulator while the strength variable indicates any parameter resisting failure. Failure is defined to have occurred when actual stress exceeds actual strength for the first time. The ratio of strength to stress provides a safety factor:

\[ n = \frac{F_y}{\sigma_x} \]  

(15-16)

Where:  
\( n \) = Factor of safety  
\( F_y \) = Material yield strength, lbs/in\(^2\)  
\( \sigma_x \) = Mean value for the stress, \( \sigma_c \), \( \sigma_l \), or \( \sigma_r \), lbs/in\(^2\)

The designer/analyst must estimate the tail probabilities for stress and strength variables based on previous experience and intimate knowledge of the design and operating environment. The lower and upper limits on these probabilities quantify the uncertainty of the estimates. The probability distributions of yield strengths for steels are found to be normally distributed.

The standard normal variable of \( (S - s) \) will be equal to:

\[ z = \frac{\mu_s - \mu_s}{\sqrt{\sigma_S^2 + \sigma_s^2}} \]  

(15-17)

Where:  
\( z \) = Probability density function  
\( \mu_S \) = Mean value of \( F_y \), lbs/in\(^2\)  
\( \mu_s \) = Mean value of \( \sigma_x \), lbs/in\(^2\)  
\( \sigma_S \) = Standard deviation of strength  
\( \sigma_s \) = Standard deviation of stress

The probability density factor can be converted to reliability and failure rate using cumulative standard normal distribution tables and assuming \( R = e^{-\lambda t} \).
15.7 REFERENCES


CHAPTER 16

THREADED FASTENERS

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16.1 INTRODUCTION

Methods of fastening or joining components include threaded fasteners, welding, brazing, soldering, and various adhesive bonding systems. One advantage of threaded fasteners is that they permit disassembly of the equipment for maintenance and repair. Threaded fasteners also allow the use of automated as well as standard manual tools for assembly and installation procedures even under restricted conditions. And significantly, the multitude of available sizes, materials, finishes, and strength levels of mechanical fasteners provides a strong design support system for optimum structural compatibility, even under wide environmental extremes. The reliability of threaded fasteners in a given operating environment depends on the strength of materials, methods of fabrication and assembly, and the stress levels created by fatigue loads and environmental conditions. This chapter considers the reliability performance of externally and internally threaded fasteners.
16.1.1 **Externally Threaded Fasteners**

Bolts and screws are fasteners with a formed head on one end and an external thread on the other end. Studs are fasteners that incorporate external threads at each end. Structural bolts and screws are installed through prepared holes in the material to be joined. Various lengths of bolts and screws are produced to accommodate the thickness of the material to be fastened and the additional length of thread needed for proper engagement with the nut or with the internal tapped thread. These fasteners are subjected to tensile, shear, bending, and fatigue loads sensed by the joint. They also respond to the environment imposed on the joint, which may include temperature extremes or exposure to various corrosive conditions.

Studs represent a special class of externally threaded fasteners, and a number of configurations have found wide use in design. Double-end studs are threaded at both ends; they can be mated with two nuts, or one end of the stud can be installed in a tapped hole and a nut employed on the other end to tighten or secure the joint. Another configuration is the continuous-threaded stud, which has been utilized both in general-purpose joining and in special classes for high-temperature and/or high-pressure bolting.

16.1.2 **Internally Threaded Fasteners**

Specific fastening devices which incorporate internal (female) threads include nuts and inserts. They are intended to engage with the external threads of bolts, screws, and studs, and should be compatible to develop the full rated strength of the external thread.

16.1.3 **Locking Mechanisms**

One aspect of threaded fastener performance that is of concern is the fastener's susceptibility to loosening as a result of severe vibration or dynamic loadings acting on the joint. Since vibratory stresses cannot be totally eliminated, several methods have been developed which have proven effective in maintaining fastener integrity.

**Self-locking nuts** are integral fasteners which incorporate in the nut element a controlled high-torque feature which is designed to prevent rotation off the external threads, even if the initial tightening torque is completely relaxed. The same principle of an inherent self-locking feature is often extended to screws used in tapped holes.

**Chemical thread-locking systems** include anaerobic and epoxy adhesives which are applied to the fastener threads before installation, and which cure and effect a permanent bond after assembly.

**Cotter pins** are a supplemental locking device used with a slotted or castellated nut. They are installed through a drilled hole in the bolt threads to prevent rotation or movement of the nut after installation.
Safety lock wiring is also a supplemental locking system. Usually two or more fasteners in series must be wired to prevent rotation in the "off" direction. The nut end can be wired using slotted or castellated nuts and bolts with drilled holes. However, the predominant use of safety wiring is to secure screw heads which have been drilled to accommodate the wire where the screws have been installed in tapped or blind holes.

Lockwashers are used to resist the loosening effects of vibration, and come in a wide variety of designs for various applications. Conical spring washers are typically made of hardened and tempered steel that is slightly concaved. These washers deform when the bolt is tightened, acting as a spring that compensates for small losses in bolt tension due to thermal expansion or compression set of the gaskets. Helical spring washers employ a single-coil helical spring that flattens under load. The spring action assists in maintaining the bolt load, while the split edges provide a locking action by biting into the bearing surfaces. Toothed lockwashers provide a gripping action resulting from the teeth biting into the material of the bolt head or nut, and being deformed axially, with the application of tension from bolt torquing.

All locking systems should be compatible. For example, using an elastic stop nut (a nut with a deformable plastic insert) on a male thread drilled for use with a cotter pin, may destroy the holding capability of the nut because of damage to the insert.

16.1.4 Threads

Inch-series thread forms were developed and used by countries employing the English system of measurement. In the United States, standards have been predicated on a 60° screw thread angle, originally standardized as the American National thread. Since 1948, the accepted standard has been the Unified thread form, which also specifies a 60° angle.

The relationship between nominal diameter and the number of threads per inch is referred to as the "diameter-pitch combination". There are several prominent thread forms which cover the majority of standards intended for general engineering use.

Coarse-thread series - This is perhaps the most widely used series of commercial and industrial fasteners. The thread form is particularly advantageous for applications requiring rapid assembly or disassembly, or for threading into lower-strength materials, such as castings, soft metals, and plastics.

Fine-thread series - For the same nominal diameter, this series incorporates more threads per inch. The result is a larger tensile stress area than that of the same size coarse thread, contributing to the greater strength capability of fine-thread fasteners. These fasteners are normally used where the length of thread engagement is short, or where a smaller lead angle is desired. Fine-thread-series fasteners are used extensively in aerospace and in applications where coarse threads would not be
suitable. Other series include the extra-fine-thread, 8-thread, 12-thread, and 16-thread series.

In addition to establishing diameter-pitch combinations, the Unified screw-thread system also defines the distinct profile and identification requirements for several screw-thread forms. The basic thread for the bulk of commercial and industrial fasteners is the Unified form, identified as "UNC" for the coarse-thread series and "UNF" for the fine-thread series.

16.2 FAILURE MODES

16.2.1 Hydrogen Embrittlement

The phenomenon of hydrogen embrittlement in threaded fasteners has been mostly associated with high-strength (over 160 ksi) steel parts which have been furnished with either zinc or cadmium electroplating. During the plating process, atomic hydrogen can be trapped in or under the plating. Other sources of hydrogen can be traced to material pickling or alkaline or acid cleaning. Unless the free hydrogen is removed, when the fastener is used and stressed, as in a structural application, the hydrogen can attack the grain boundaries. The result is rapid crack propagation and often catastrophic failure of the steel fastener.

Two major ways to avoid the problems of hydrogen embrittlement are to (1) bake the fasteners in a subsequent operation after plating to remove the excess hydrogen or (2) use mechanical plating processes in lieu of electroplating. Mechanical plating is the process of using glass beads to cold-weld a ductile metal (e.g., cadmium, zinc, etc.) onto a metal substrate by mechanical energy. Coating thicknesses are more uniform than when the hot-dip process is used. However, hot-dip fasteners have more corrosion protection built-in, due to the greater coating thickness. Usually, it is necessary to chase the hot-dip coated threads with a die, since the coating is not uniform.

16.2.2 Fatigue

The importance of fatigue strength properties is associated with the fact that when failure is encountered, it is invariably catastrophic in nature, and often occurs without warning. Research has established that rolled threads exhibit higher fatigue life than machined or ground threads. Further, threads rolled after heat treatment show better fatigue performance than comparable threads rolled before heat treatment. In addition, factors such as proper bolt head design, cold work of the head-to-shank fillet, quality control of the basic material used, and minimization of possible metallurgical defects all contribute significantly to improved fatigue life.

With respect to fatigue performance, it has been observed that failures normally develop at stress levels well below the static strength of the fastener. The two main
types of joint fatigue loading are shear fatigue and tension fatigue. For shear-loaded joints, fatigue failure normally occurs in the plate or sheet material. The applied fatigue or dynamic stresses, hole preparation, hole clearance, amount of induced bending, and fastener preload are some of the factors which influence shear joint fatigue life.

16.2.3 Temperature

Both high-temperature and low-temperature service exposures are experienced in practice in nuclear systems, aerospace, electronics, transportation, energy systems, construction, and similar applications. Characteristically, materials used at very low temperatures will show an increase in tensile strength, but may sacrifice ductility. Conversely, at elevated temperatures, tensile strength properties are usually reduced, and above critical service temperature limits they may drop off dramatically.

16.2.4 Load and Torque

For every fastener system, there is an optimum torque range to develop the design clamp load. This is normally referred to as the "torque-tension relationship". Over torquing can result in excessive bolt yielding and possible subsequent relaxation, or even thread stripping and failure on installation. Too low an initial torque can contribute to potential fatigue and/or joint loosening with extended service life.

There are several factors which affect and influence the nominal torque-tension relationship, including condition of the threads, condition and squareness of the joint, method and equipment for torquing, installation from the nut or bolt head end, and lubrication. Possibly the most influential factor is the lubrication (plating and/or supplemental lubricant) on the fastener system, since the effective coefficient of friction can alter the installation torque requirements by as much as 50 to 100 percent.

16.2.5 Bolt and Nut Compatibility

Particularly where high-strength bolts are used, critical attention must be given to specifying the correct mating nut. Inadvertent specification of a lower-strength (grade) nut invites the possibility of nut thread stripping under high tensile loading. But more significantly, a weaker nut will not adequately develop the full clamp load capability of a high-strength bolt when subjected to the necessary installation torque. As a rule of thumb, the thickness (height) of a nut should approximate the diameter of the equivalent mating bolt to develop the full tensile strength properties of the bolt, if the bolt and nut materials have the same strength.

16.2.6 Vibration

Whereas fatigue loading is presumed to be relatively high with respect to the strength of the threaded fastener or the joint, vibration loads are relatively low, but may be associated with various ranges of cyclic frequencies. Critical combinations of frequency, loading, and amplitude can force a structure into resonance, often with
catastrophic results. While the overwhelming majority of operating structures are not subjected to conditions of resonance, the vibration forces present (including random and steady-state vibration, shock, and impact) are sometimes serious enough to drastically affect the threaded fastener system.

Under repeated or extensive vibration, there is a tendency for the nut to rotate or loosen off the bolt threads. Continued vibration can actually result in the nut completely disengaging from the bolt, with subsequent loss of the bolt from the joint. Not as severe, but just as important, vibration loosening can reduce or completely relax the original preload in the bolt, causing the bolt to sense increased fatigue loads with continued exposure. What may have first started as vibration loosening may actually end as a fatigue failure because of the complex stress mechanisms involved.

### 16.3 STRESS-STRENGTH MODEL DEVELOPMENT

#### 16.3.1 Static Preload

The most important factor that determines the preload induced in a bolt is the torque applied to tighten the bolt. There are several methods commonly used to apply a predetermined torque. The torque may be applied manually by means of a wrench which has a dial attachment that indicates the magnitude of torque being applied. Pneumatic air wrenches are also widely used. Another method is to tighten the nut by hand and then use a wrench to give the nut a predetermined number of turns.

Figure 16.1 shows a typical bolt with commonly used symbols for the dimensions. An empirical equation can be used to show the relationship between induced preload and applied torque:

\[ T = d F_i c \]  \hspace{1cm} (16-1)

Where:

- \( T \) = Applied torque, in-lb
- \( d \) = Major bolt diameter, in.
- \( F_i \) = Initial preload, lb
- \( c \) = Torque coefficient, and is given by the relationship (Ref. 62):

\[ c = \frac{d_m}{2d} \left( \frac{\mu_i \sec \phi + \tan \psi}{1 - (\mu_i \sec \phi \tan \psi)} \right) + \mu_c \left( \frac{d_c}{2d} \right) \]  \hspace{1cm} (16-2)

Where:

- \( d_m \) = Thread pitch diameter, in.
- \( d_c \) = Mean bearing face or collar diameter, in.
\( \mu_l = \) Bolt thread coefficient of friction

\( \mu_c = \) Coefficient of friction at bearing face of bolt or nut

\( \phi = \) Thread half angle, 30 degrees

\( \psi = \) Thread helix angle, degrees

\[
= \tan^{-1}\left( \frac{I}{\pi N d_m} \right)
\]

(16-3)

Where: \( N = \) Number of threads/in

For most applications, the value for the torque coefficient can be approximated by the values listed in Table 16-3. The values presented are typical because of the wide range of values reported from coating manufacturers. The user is cautioned to consult the noted reference in critical applications.

A simple bolted joint can also be dangerous unless it is properly designed for the loading and assembled by a trained mechanic. In any fastening situation, the basic aim is to determine as accurately as possible the least expensive fastener that, when properly tightened, will secure a joint during product life. Properly applied assembly torque produces the wedging action of the fastener threads that elongate the bolt to produce tension. Tension (or preload) induced in a fastener at assembly should always be greater than any external load the joint will experience in service. A preload ensures optimum performance if it prevents the clamped parts from separating in service. Thus a preload should always exceed any external load or payload. The fastener generally remains unchanged until the external load exceeds the preload. Therefore, the higher the preload, the greater potential there is for withstanding larger external loads. This is applicable to perfectly rigid joints, which solid, metal-to-metal joints approximate. A high
preload also helps to retain friction at the joint interface, which is important when shear loads are present.

For the case of the fastener joint loaded statically, if the preload properly exceeds the external loading, a joint will remain in service. However, there are instances as mentioned previously in Section 16.2 that can cause relaxation of the preload or an increase in the external loading, causing premature failure of the fastener joint.

### 16.3.2 Temperature Effects of Fasteners in Clamped Joints

Changes in temperature must be considered in fastener joint design, in that they can act to change the clamping force in joint and/or the tension (preload) in the fastener. If at any time during operation, the external loading exceeds the preload of the fastener at operating temperature, the fastener is considered to have failed. Therefore, the desired (room temperature) preload, with correction factors for temperature effects can be modeled as follows:

\[
F_{i,ambient} = \left[ F_{i,operating} \cdot \frac{E_1}{E_2} \cdot \frac{I}{K_{SR}} \right] - \frac{A_S E_1}{L_E} \left( \Delta L_J - \Delta L_B \right) \quad (16-4)
\]

Where:

- \(F_{i,ambient}\) = Preload at ambient temperature, lb
- \(F_{i,operating}\) = Fastener design preload at elevated operating temperature, lb
- \(E_1\) = Modulus of elasticity at room temperature, lbs/in\(^2\) (See Table 16-1)
- \(E_2\) = Modulus of elasticity at operating temperature, lbs/in\(^2\) (See Table 16-1)
- \(K_{SR}\) = Correction factor accounting for stress relaxation in the fastener at elevated operating temperatures (See Figure 16.3)
- \(A_S\) = Fastener tensile stress area, in\(^2\)

\[
A_S = 0.7854 \left( d - \frac{0.9743}{N} \right)^2 \quad (Reference 59) \quad (16-5)
\]

- \(d\) = Major bolt diameter, in
- \(N\) = Number of threads per inch
- \(L_E\) = Effective length (as defined in Figure 16.2) of fastener, in
\[ \Delta L_J = \text{Change in length or thickness of the joint, in} \]
\[ = L_G \cdot \alpha_J \cdot T_O \quad (16-6) \]

\[ L_G = \text{Thickness of clamped joint, in} \]

\[ \alpha_J = \text{Thermal coefficient of linear expansion for joint material} \]
(See Table 16-4)

\[ \Delta L_B = \text{Change in grip length of the fastener, in} \]
\[ = L_G \cdot \alpha_B \cdot T_O \quad (16-7) \]

\[ \alpha_B = \text{Thermal coefficient of linear expansion for bolt} \]
(See Table 16-4)

\[ T_O = \text{Elevated operating temperature, } ^\circ\text{F} \]

Table 16-1 presents values of material moduli of elasticity vs. various temperatures. It should be noted that the equation above will have lower preloads at room temperature assembly, if the fastener joint is designed to be operated at a lower temperature (lower temperatures increase the stiffness of a bolted assembly). Designers must also be aware that the strength of most bolts decreases with rising temperatures, as illustrated in Table 16-2.

![Figure 16.2 Determining Effective Fastener Length (L_E)](image)

Table 16-4 contains the values of coefficients of linear expansion for typical bolting material, \( \alpha_B \), and joint material, \( \alpha_J \). These values are used in computing \( \Delta L_J \) and \( \Delta L_B \) of Equation (16-4). It is necessary to note that the coefficients themselves, are temperature dependent. The values in the table are based on room temperature. For evaluations of temperatures 400\(^{\circ}\)F and above, the noted reference should be consulted. At elevated temperatures, with dissimilar joint/bolt materials, if the joint material expands more than the bolt material, the bolt will develop more stress or preload.
than it was designed to experience. To the contrary, if the bolt material expands more than the joint material at elevated operating temperatures, this will act to lessen the design bolt preload. The correction term in Equation (16-4) is therefore subtracted from the preload at assembly \((F_{i,ambient})\) to account for these situations.

It has been determined that at greatly elevated temperatures, many materials experience a slow increase in length under a heavy, constant load. This phenomenon is called creep. A bolted joint assembly may experience a slightly different phenomenon, under which a steady loss of stress in a heavily loaded part whose dimensions are fixed or constrained, called stress relaxation. \(K_{SR}\) in Equation (16-4) accounts for this time/temperature dependent "stress relieving" factor when the bolted assembly is to be operated at extreme temperatures. Figure 16.3 contains several plots of correction factors for various bolt materials, derived from stress relaxation data over an elevated temperature range after 1000 hours exposure at each temperature. From the fairly constant initial ranges of each curve in Figure 16.3, it is evident that the stress relaxation effect is of concern only at the higher temperatures (i.e., 600°F and above).

![Figure 16.3 Stress Relaxation Factor, \(K_{SR}\), for Various Operating Temperatures & Materials (After 1000 Hours) (Ref. 60)](image)

**NOTES:**

(1) Equivalent Materials -
ASTM A193 = B7, B8M, B16
NIMONIC 80A = B80A
AISI 660 = B17

(2) Residual Stress Reduction for other materials can be derived via Gieske's Correlation *(Ref. 63)* (Given creep data @ 1000 hrs & spec. temp);
residual stress in a bolt = stress that produces 0.01% creep
Stress relaxation losses are not repetitive in temperature cycling situations. The material stress value stabilizes at some lesser value after some period of time. This is because the tendency to relax decreases as the tensile stress (the driving force) in the bolt, decreases.

16.3.3 Corrosion Considerations

Corrosion is a problem often faced when dealing with bolted assemblies. Excessive corrosion can eventually lead to a reduction in preload, or to the total loss of clamping force through destruction of material. Several methods (Reference 60) used to combat the onset of corrosion are as follows:

1. Select materials in the joint assembly (bolts, nuts, structure) that are identical, or as close together as possible in the galvanic series, minimizing electrical potential differences.

2. In a situation of dissimilar metals, the larger amount of material should act as an anode, while the smaller amount of material behaves as the cathode.

3. Introduce a 'sacrificial' anode, that can be replaced from time to time. This can be a block of material placed in the vicinity of the bolted joint, where material is sacrificed in a galvanic reaction.

4. Minimize stresses and/or stress concentrations in fasteners and joints by providing generous fillets, polishing surfaces, preloading bolts uniformly, etc. Stress tends to aggravate the galvanic reaction; therefore, stress concentrations at the root of a crack will increase the galvanic reaction between the bolt and the adjacent material, aiding to the growth of fatigue cracks.

5. Various coatings can resist corrosion by:
   (a) providing a barrier by isolating the bolt from the corrosive environment; cadmium is a common coating, which provides barrier protection.
   (b) inhibiting the process of corrosion.
   (c) provide galvanic/sacrificial protection of the corrosive materials. Zinc-coated, or galvanized fasteners provide sacrificial protection, along with some barrier protection.

6. Periodic replacement of the bolts, prior to failure, can be a practical solution. This approach intensifies the need to be able to accurately predict the amount of life remaining in a fastener, so that the bolted assemblies are not dismantled prematurely.
Stress corrosion cracking is one of the more serious reliability problems associated with bolting applications. This condition is relatively common in many bolts, and may ultimately lead to sudden and unexpected failure (See Figure 16.4). Although every metallic bolting material is susceptible to stress corrosion cracking under certain conditions, carbon steel and low alloy quenched and tempered fasteners with a hardness below about 35 HRC are generally immune (for environments such as humid air, aqueous chloride, etc.) (Reference 60).

![Figure 16.4 Stress Corrosion Cracking](image)

A mathematical model can be developed to make an approximate assessment of the useable life remaining in a fastener through the manipulation of the empirical relationship (Reference 63) for the determination of safe levels of applied stress to resist stress corrosion cracking:

\[
\sigma = \frac{K_{ISCC}}{C_{thd}} (\pi a)^{-0.5}
\]

(16-8)

Where:
- \( \sigma \) = Nominal stress, lbs/in\(^2\)
- \( K_{ISCC} \) = Threshold stress intensity factor for stress corrosion cracking, ksi(in)\(^{0.5}\) (See Figure 16.5)
- \( C_{thd} \) = Material shape factor, considered 1.5 for fasteners with threads
- \( a \) = Maximum allowable material crack/flaw depth prior to exposure to corrosive environment, in
Equation (16-8) can be rewritten to generate an expression for the maximum preload, $F_{i,ambient}$, which can be safely applied at the time of joint assembly and prevent failure from stress corrosion cracking:

$$F_{i,ambient} = \frac{A_S K_{ISCC}}{C_{thd}}(\pi a)^{-0.5} \cdot C_{dia}$$  \hspace{1cm} (16-9)

Where:

- $A_S$ = Fastener tensile stress area, in$^2$ [See Equation (16-5)]
- $C_{dia}$ = Correction factor for varying bolt diameters & thread pitch
  (See Table 16-5)

Figure 16.5 $K_{ISCC}$ Factor vs. Material Yield Strength. (Low alloy, quenched & tempered material, in humid environment) (Ref.60)

Much work has yet to be done in determining characteristics of the many fastener materials that aid in the prediction of the onset of failure due to stress corrosion cracking. The estimation is, however, subject to the following constraints, due to the limited experimental data available:

1. Susceptibility to stress corrosion cracking can increase significantly with elevated temperatures; model estimation is limited to room temperature applications.
(2) \( K_{\text{ISCC}} \) values shown in Figure 16.5 are limited to low alloy, quenched and tempered fastener materials, such as:

- ASTM A193 B7, B16
- ASTM A490, A307, A540
- SAE J 429 GR.8
- AISI 4340

(3) The corrosive environment used to derive the experimental data was based upon humid air.

A correction term has been added to Equation (16-9), since there is a relationship between the depth of the threads on a fastener and its sensitivity to stress corrosion cracking. Generally, the larger diameter fasteners of a given material will have a lower threshold stress level than small bolts. For the same reason, fasteners with fine pitch threads are less sensitive than those with coarse threads. Various correction factors for bolt sizes are given in Table 16-5.

The line in Figure 16.5 represents the lower bound to the test points indicated. Therefore, \( K_{\text{ISCC}} \) can only be determined empirically and is valid only for a tested material in a given temperature.

16.3.3.1 Estimating the Failure Rate of a Fastener in a Corrosive Environment

The following paragraphs illustrate an approach to the estimation of useable life remaining in a fastener susceptible to the effects of corrosion (in the form of stress corrosion cracking).

- Determine the desired design preload at ambient conditions to be developed by the fastener.
- Using this value, solve Equation (16-9) for the maximum crack depth, \( a \). This value should be between the limits, bolt surface finish tolerance \( \leq a \leq \) twice thread depth.

Once the value, \( a \), has been solved for, and it has been checked to exist between proper boundaries, it can be correlated to a plot of corrosion data. This corrosion data should include samples of the same material, exposed to the same type of atmosphere used to compute the \( K_{\text{ISCC}} \) factor. The weight loss per area of specimen exposure can be computed using the following equation:

\[
W = 1.18 \times 10^{-3} \gamma a
\]

Where: \( W = \) Weight per unit surface area, lbs/in\(^2\)
\( \gamma = \) Bolt material density, lbs/in\(^3\) \\
\( a = \) Maximum bolt material crack/flaw depth prior to failure, in

After solving Equation (16-10) for \( W \), the user may enter the graph in Figure 16.6 to estimate a useful bolt life.

![Figure 16.6 Typical Coupon Corrosion Penetration Test Results, (Low alloy steel in humid air) (Ref. 61)](image)

16.3.4 Dynamic Loading

Machinery in operation is a dynamic situation. Fasteners used in many applications have a small dynamic load superimposed on a much larger static preload. These dynamic or fluctuating loads are augmented by stress concentrations and bending.

16.3.4.1 Determination of Base Failure Rate

Data available from fastener manufacturers that have performed extensive testing can be used to determine the base failure rate, \( \lambda_{F,B} \). However, since this data is somewhat specific to the conditions or environment of the test, it is often difficult to locate data that will yield the base failure rate of a specific type of fastener, under a particular set of loading conditions. This necessitates the development of a procedure
to estimate the base failure rate of a generic fastener, with various characteristics and loading conditions.

Fatigue, as discussed in Section 16.2.2, can limit the useable life of a fastener in a dynamic loading condition. Fatigue limit testing can be valuable in developing a model for fastener failure rates. A large number of tests are necessary to establish the fatigue strength of a material due to the statistical nature of fatigue. One of the most widely used fatigue testing devices is the R.R. Moore high speed rotating beam machine (Reference 39). Figure 16.7 illustrates the specimen and method. A motor spins a slender, round, solid, polished test specimen, supported at each end but loaded in pure bending. The majority of published fatigue strength data was obtained using this method (Ref. 19). The generated bending stress and the number of stress reversal revolutions of the beam, required for failure, is recorded and graphed.

![Diagram of Rotating Beam Machine, with Detail of Specimen](References 19 and 39)

The stress values become less as the data is plotted against an increasing number of stress cycles. The graph becomes horizontal in the case of ferrous metals and alloys after the material has been stressed for a certain number of cycles. This is referred to as the 'endurance' or fatigue limit. Table 16-6 presents endurance limit properties for several bolting materials. Aluminum, or other nonferrous materials do not have a horizontal asymptote, hence limit. To develop a model to predict fastener failures under dynamic loading, it is necessary to correlate the results of standard material fatigue tests to the geometry and loading conditions of the fastener to be used. It must be noted that even when the material of the test specimen and that of the mechanical fastener are identical, there will be significant differences between the fatigue curves for the two. Therefore, although correction factors will be presented in an effort to compensate for this, the user is cautioned that the analytical model developed will not yield absolutely precise results.
If the shape of a fatigue curve is known, the statistical number of cycles to failure, \( N \), or ultimately the expected failure rate, \( \lambda_{F,B} \), can be found. If the S-N diagram is not available, there exists a means to analytically determine it (Reference 39). The equation of the S-N curve is given by:

\[
\sigma_f = a N^b
\]  

(16-11)

Where:
- \( \sigma_f \) = Fatigue stress at failure, lbs/in\(^2\)
- \( N \) = Number of stress reversal cycles at failure
- \( a, b \) = See Equations (16-13) & (16-14)

By taking the log of both sides of Equation (16-11) the following characteristics are noted:

- The endurance limit occurs at \( N = 10^6 \)
- Low cycle fatigue terminates at \( N = 10^3 \)
- S-N curves terminate low cycle fatigue at \( \sigma_f = 90\% \) of material ultimate tensile strength (\( \sigma_{T,ult} \))

Then:

\[
\log \sigma_f = \log a + b (\log N)
\]  

(16-12)

Where:

\[
a = \frac{(0.9 \sigma_{T,ult})^2}{\sigma_c}
\]  

(16-13)

and:

\[
b = -\frac{1}{3} \log \left( \frac{0.9 \sigma_{T,ult}}{\sigma_e} \right)
\]  

(16-14)

Where:
- \( \sigma_c \) = Endurance or fatigue limit, lbs/in\(^2\)
- \( \sigma_{T,ult} \) = Ultimate tensile strength, lbs/in\(^2\)
The actual known fastener endurance limit value (Table 16-6) can be substituted for \( \sigma_e \) in Equations (16-13) and (16-14) above. If the specific endurance limit is not known (which is generally the case) an endurance limit for an S-N test specimen (of the same material) can be used. If this data is not readily available, an approximation can be made using the relationships in Table 16-7.

16.3.4.2 Correction Factors for the S-N Test Specimen Data

The S-N test specimen must be corrected for the specific conditions and geometry of the fastener under consideration. Correction factors have been established to account for the individual contributions by surface finish, size differential, loading, temperature, etc. The following equation should be employed when utilizing S-N data for fasteners based on rotating beam test specimen results:

\[
\lambda_F = \lambda_{F,B} \cdot C_{SZ} \cdot C_L \cdot C_T \cdot C_I \cdot C_{SC} \cdot C_K
\]  
(16-15)

Where:  
\( \lambda_F \) = Failure rate of fastener, failures/million hours  
\( \lambda_{F,B} \) = Base failure rate, failures/million hours = \( 1/\sigma_{e,S-N} \)  
\( \sigma_{e,S-N} \) = Endurance or fatigue limit of S-N test specimen, lbs/in\(^2\)  
(See Table 16-7)  
\( C_{SZ} \) = Multiplying factor considering the effects of size deviation from the S-N test specimen (See Section 16.3.4.3)  
\( C_L \) = Multiplying factor considering the effects of different loading applications (See Section 16.3.4.4)  
\( C_T \) = Elevated temperature multiplying factor (See Section 16.3.4.5)  
\( C_I \) = Multiplying factor considering for the severity of in-service cyclic shock (impact) loading (See Section 16.3.4.6)  
\( C_{SC} \) = Surface coatings multiplying factor (See Section 16.3.4.7)  
\( C_K \) = Stress concentration multiplying factor for fastener threads (See Section 16.3.4.8)

It should be noted that rotating beam data generally carries with it, a great deal of scatter. Therefore, any life determinations based on the data will be statistical at best. No attempt has been made in Equation (16-15) to account for the statistical uncertainty. It has been suggested that rotating beam data reflects a reliability in actual survival of only 50% confidence. A factor of 0.814, applied to Equation (16-15) is encouraged by Reference 19, in order to introduce a 99% reliability confidence level in S-N endurance test data and estimates.
16.3.4.3 Size Multiplying Factor

Smaller machine parts tend to exhibit greater fatigue strength than larger ones, all other configurations and material properties being equal. Since larger surfaces have more defects overall, the probability of failure is greater in larger parts. A correction factor is established to account for this, as well as the bending of a solid circular material, without constant rotation (Reference 39):

For bending or torsional loading:

\[ C_{SZ} = \left( \frac{0.370 \cdot d}{0.3} \right)^{-0.1133} \]  \hspace{1cm} (16-16)

Where: \( d \) = Major diameter of fastener, 2 inches or less

For axial loading:

\[ C_{SZ} = 1.0 \]  \hspace{1cm} (16-17)

16.3.4.4 Alternate Loading Multiplying Factor

Appropriate load factors for \( C_L \) are presented in Table 16-8.

16.3.4.5 Temperature Multiplying Factor

Typical rotating beam data are acquired at room temperature. However, fasteners are often called upon to clamp equipment at higher temperatures. Since a decline in static and dynamic strengths, creep, and thermal expansion must all be taken into account at higher temperatures, the following factor should be applied to the S-N test specimen data to achieve correction:

For steel operating above 160°F:

\[ C_T = \frac{460 + T_O}{620} \]  \hspace{1cm} (16-18)

and:
For $T_O \leq 160^\circ$F: $C_T = 1.0$

Where: $T_O$ = Operating temperature of fastener, °F

16.3.4.6 Cyclic Shock/Impact Loading Multiplying Factor

A correction factor, $C_I$, must be applied when shock loads are present. In general, cyclic loads are less severe than applied shock loading. The factors are presented in Table 16-9.

16.3.4.7 Surface Coatings Multiplying Factor

Surface treatments such as electroplating and spraying act to increase the endurance limit. If any of these operations are to be used, Table 16-10 provides the $C_{SC}$ multiplying factor.

16.3.4.8 Thread Correction Multiplying Factor

Observations of typical bolt failure pattern data from Reference 39 have revealed that only about 15% of failures occur under the head, due to the stress riser caused by the fillet. The risers found in the thread area accounted for the other 85%. Therefore, Table 16-11 presents the multiplying factor, $C_K$, for stress risers caused by the introduction of threads. The effects of notch sensitivity and surface finish have been incorporated.
Table 16-1. Elasticity Modulus ($10^6$ lbs/in$^2$) as a Function of Temperature  
(Reference 60)

<table>
<thead>
<tr>
<th>SPEC</th>
<th>GRADE</th>
<th>TEMPERATURE, degrees F</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
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<td>-325</td>
</tr>
<tr>
<td>ASTM</td>
<td>B5</td>
<td>32.9</td>
</tr>
<tr>
<td>A193</td>
<td>B6</td>
<td>31.2</td>
</tr>
<tr>
<td></td>
<td>B7</td>
<td>31.6</td>
</tr>
<tr>
<td></td>
<td>B8-CL 1</td>
<td>30.3</td>
</tr>
<tr>
<td></td>
<td>B16</td>
<td>31.6</td>
</tr>
<tr>
<td>ASTM</td>
<td>L7</td>
<td>31.6</td>
</tr>
<tr>
<td>A307</td>
<td>L43</td>
<td>31.6</td>
</tr>
<tr>
<td></td>
<td>B8</td>
<td>30.3</td>
</tr>
<tr>
<td>ASTM</td>
<td>Type</td>
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</tr>
<tr>
<td>A325</td>
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<td>ASTM</td>
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<td>A540</td>
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Table 16-2. Bolt Yield Strength (ksi) as a Function of Temperature
(Reference 60)

<table>
<thead>
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<th>TEMPERATURE, degrees F</th>
</tr>
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<tbody>
<tr>
<td></td>
<td></td>
<td>70</td>
</tr>
<tr>
<td>ASTM A193</td>
<td>B8-C1 1</td>
<td>30</td>
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<tr>
<td>ASTM A307</td>
<td>GR B</td>
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<tr>
<td>ASTM A320</td>
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<td>Stainless Steel</td>
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<tr>
<td></td>
<td>CL2</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>CL3</td>
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<td></td>
<td>CL4</td>
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<tr>
<td></td>
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Table 16-3. Typical Torque Coefficients  
(Reference 60)

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<thead>
<tr>
<th>FASTENER MATERIAL/COATING</th>
<th>TORQUE COEFFICIENT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum on AISI 8740 alloy steel</td>
<td>0.52</td>
</tr>
<tr>
<td>Mild or alloy steel on steel</td>
<td>0.20</td>
</tr>
<tr>
<td>Stainless steel on mild/alloy steel</td>
<td>0.30</td>
</tr>
<tr>
<td>1&quot; dia. A490</td>
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<td>1&quot; dia. A490 (rusty)∗</td>
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<td>Black Oxide</td>
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<td>Cadmium plate (dry)</td>
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<tr>
<td>Cadmium plate (waxed)</td>
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<td>Galvanized A325</td>
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<td>Galvanized, hot-dip A325</td>
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<tr>
<td>Gold on stainless steel or beryllium copper</td>
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</tr>
<tr>
<td>Graphitic coatings</td>
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</tr>
<tr>
<td>Machine Oil</td>
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<tr>
<td>Moly paste or grease</td>
<td>0.13</td>
</tr>
<tr>
<td>Solid film PTFE</td>
<td>0.12</td>
</tr>
<tr>
<td>Zinc plate (waxed)</td>
<td>0.29</td>
</tr>
<tr>
<td>Zinc plate (dry)</td>
<td>0.30</td>
</tr>
</tbody>
</table>

∗ Exposed outdoors for two weeks
Table 16-4. Thermal Coefficients, $\alpha$, of Linear Expansion ($10^{-6}$ in/in/$^\circ$F)
Evaluated at 70°F (Reference 60)

<table>
<thead>
<tr>
<th>SPEC</th>
<th>GRADE</th>
<th>TEMPERATURE, degrees F</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>70</td>
</tr>
<tr>
<td>ASTM</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A193</td>
<td>B5</td>
<td>6.5</td>
</tr>
<tr>
<td></td>
<td>B6</td>
<td>5.9</td>
</tr>
<tr>
<td></td>
<td>B7</td>
<td>5.6</td>
</tr>
<tr>
<td></td>
<td>B8</td>
<td>8.5</td>
</tr>
<tr>
<td></td>
<td>B16</td>
<td>5.4</td>
</tr>
<tr>
<td>ASTM</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A307</td>
<td></td>
<td>6.4</td>
</tr>
<tr>
<td>ASTM</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A320</td>
<td>L7</td>
<td>5.6</td>
</tr>
<tr>
<td></td>
<td>L43</td>
<td>6.2</td>
</tr>
<tr>
<td></td>
<td>L7M</td>
<td>6.2</td>
</tr>
<tr>
<td></td>
<td>B8 CL 1</td>
<td>8.5</td>
</tr>
<tr>
<td>ASTM</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A325</td>
<td></td>
<td>6.2</td>
</tr>
<tr>
<td>ASTM</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A354</td>
<td></td>
<td>6.2</td>
</tr>
<tr>
<td>ASTM</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A449</td>
<td></td>
<td>6.2</td>
</tr>
<tr>
<td>ASTM</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A453</td>
<td></td>
<td>9.1</td>
</tr>
<tr>
<td>ASTM</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A490</td>
<td>651</td>
<td>6.2</td>
</tr>
</tbody>
</table>
Table 16-5. Correction Factors for UNC Thread Bolt Sizes, $C_{dia}$
(Derived from Data) (Reference 60)

<table>
<thead>
<tr>
<th>BOLT DIAMETER (inches)</th>
<th>BOLT MATERIAL HARDNESS*</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>22 HRC</td>
</tr>
<tr>
<td>1.0 and below</td>
<td>1.0</td>
</tr>
<tr>
<td>1.5</td>
<td>0.93</td>
</tr>
<tr>
<td>2.0</td>
<td>0.87</td>
</tr>
<tr>
<td>2.5</td>
<td>0.84</td>
</tr>
<tr>
<td>3.0</td>
<td>0.84</td>
</tr>
<tr>
<td>4.0</td>
<td>0.84</td>
</tr>
</tbody>
</table>

Table 16-6. Endurance Limit Properties for Various Bolting Materials
(Reference 60)

<table>
<thead>
<tr>
<th>MATERIAL GRADE OR CLASS</th>
<th>SIZE RANGE</th>
<th>ENDURANCE LIMIT AT $10^6$ CYCLES</th>
</tr>
</thead>
<tbody>
<tr>
<td>SAE 5</td>
<td>¼ - 1 in</td>
<td>16.6 kpsi</td>
</tr>
<tr>
<td></td>
<td>1 – 1 ½ in</td>
<td>16.3 kpsi</td>
</tr>
<tr>
<td>SAE 7</td>
<td>¼ - 1 ½ in</td>
<td>20.6 kpsi</td>
</tr>
<tr>
<td>SAE 8</td>
<td>¼ - 1 ½ in</td>
<td>23.2 kpsi</td>
</tr>
<tr>
<td>ISO 8.8</td>
<td>M16 – M36</td>
<td>129 MPa</td>
</tr>
<tr>
<td>ISO 9.8</td>
<td>M1.6 - M16</td>
<td>140 MPa</td>
</tr>
<tr>
<td>ISO 10.9</td>
<td>M5 – M36</td>
<td>162 MPa</td>
</tr>
<tr>
<td>ISO 12.9</td>
<td>M1.6 – M36</td>
<td>190 MPa</td>
</tr>
<tr>
<td>Metric M16, CL 8.8</td>
<td>M16</td>
<td>10.2 ksi</td>
</tr>
<tr>
<td>Metric M14 x 1.5</td>
<td>M14 x 1.5</td>
<td>7.1 – 11.4 ksi</td>
</tr>
<tr>
<td>SAE J429, GR 8</td>
<td></td>
<td>18 ksi</td>
</tr>
<tr>
<td>Metric CL 10.9</td>
<td>M12 x 1.25</td>
<td>8 ksi</td>
</tr>
<tr>
<td>M10, GR 12.9</td>
<td></td>
<td>6.9 – 10.7 ksi</td>
</tr>
</tbody>
</table>
Table 16-7. Formulas for Estimating the Endurance Limit $\sigma_e$ of S-N Test Specimens (Reference 19)

For Steels, where $\sigma_{T,ult} \neq 200,000$ psi:

$$\sigma_e = 0.5 \sigma_{T,ult}$$

For Steels, where $\sigma_{T,ult} > 200,000$ psi:

$$\sigma_e = 100,000 \text{ psi}$$

For Aluminum Alloys (wrought):

$$\sigma_e = 0.4 \sigma_{T,ult}$$

For Aluminum Alloys (cast):

$$\sigma_e = 0.3 \sigma_{T,ult}$$

* NOTE: The endurance limit for nonferrous alloys is taken to occur at approximately $10^8$ cycles.

Table 16-8. Load Multiplying Factors (Reference 39)

<table>
<thead>
<tr>
<th>TYPE OF LOAD APPLIED</th>
<th>$C_L$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial ($\sigma_{T,ult} \leq 220$ kpsi)</td>
<td>1.09</td>
</tr>
<tr>
<td>Axial ($\sigma_{T,ult} &gt; 220$ kpsi)</td>
<td>1.00</td>
</tr>
<tr>
<td>Bending</td>
<td>1.00</td>
</tr>
<tr>
<td>Torsion and Shear</td>
<td>1.72</td>
</tr>
</tbody>
</table>
Table 16-9. Multiplying Factor for Impact Loading  
(Reference 19)

<table>
<thead>
<tr>
<th>IMPACT CATEGORY</th>
<th>$C_I$</th>
</tr>
</thead>
<tbody>
<tr>
<td>LIGHT (rotating machinery - motors, turbines, centrifugal pumps)</td>
<td>1.00</td>
</tr>
<tr>
<td>MEDIUM (rotary &amp; reciprocating motion machines - compressors, pumps)</td>
<td>1.25</td>
</tr>
<tr>
<td>HEAVY (presses for tools &amp; dies, shears)</td>
<td>1.67</td>
</tr>
<tr>
<td>VERY HEAVY (hammers, rolling mills, crushers)</td>
<td>2.50</td>
</tr>
</tbody>
</table>

Table 16-10. Multiplying Factor for Surface Coatings  
(References 19 and 39)

<table>
<thead>
<tr>
<th>SURFACE TREATMENT</th>
<th>$C_{SC}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electroplating (chromium, nickel, cadmium)</td>
<td>1.54</td>
</tr>
<tr>
<td>Electroplating (zinc)</td>
<td>1.00</td>
</tr>
<tr>
<td>Metal spraying</td>
<td>1.16</td>
</tr>
</tbody>
</table>

Table 16-11. Multiplying Factor for Threaded Elements, $C_k$  
(Reference 39)

<table>
<thead>
<tr>
<th>SAE GRADE BOLT</th>
<th>ROLLED THREADS</th>
<th>CUT THREADS</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$C_k$</td>
<td>$C_k$</td>
</tr>
<tr>
<td>0-2</td>
<td>2.2</td>
<td>2.8</td>
</tr>
<tr>
<td>4-8</td>
<td>3.0</td>
<td>3.8</td>
</tr>
</tbody>
</table>
16.4 REFERENCES


63. Thomas Couplings Applications Manual

64. Bolam, J.R., "Coupling Alignment: The Reverse Indicator Method Simplified", P/PM Technology, July/Aug 90


66. Robertson, R., and Smith, B., "Why Flexible Couplings Fail", Plant Engineering and Maintenance, Jun 89

17.1 INTRODUCTION

A coupling is used to make an semi-permanent axial connection between two shafts. A coupling may be rigid or flexible, rigid couplings being used in applications where misalignment is not a factor or where flexibility is not required. Thus, flexible couplings are more common. Flexible couplings are designed to accommodate various types of load conditions including rotation from drive to load, vibration damping, and misalignment reduction.

Misalignment is an important consideration in determining the failure rate of the total system containing a coupling. The maximum allowable misalignment is a function of the coupling material, the percentage of torque capacity being used, and the transmitted vibration. Misalignment may cause radial forces to be exerted on other components of the system. Excessive radial forces can cause stresses on bearings, seals and other parts to the point of premature failure. No one type of coupling can provide the universal solution to all coupling problems so there are many designs available for specific applications. It is important to check that for systems that frequently absorb the peak torque capacity of the power source that this peak torque does not exceed the normal torque capacity of the coupling.
17.1.1 Rigid Shaft Couplings

Rigid couplings are used when shafts have good collinear alignment. Although simple in design, rigid couplings are usually restricted to relatively low speed applications where good shaft alignment or shaft flexibility can be expected. Three major types of rigid couplings are shown in Figure 17.1.

(1) The clamp/compression type coupling relies on the clamping force developed from the fasteners to connect the two shafts. Torsional forces are normally transmitted via shaft keys.

(2) The sleeve type coupling is generally a single piece housing that transmits torque via shaft keys or tapered bushings. Axial positioning of the coupling is maintained by retaining rings or threaded shaft collars.

(3) The flange coupling mates two coupling halves together in a plane that is perpendicular to the shaft centerline. Torque can be transmitted between shafts either via the bolted fasteners in the flange, the frictional contact between flange faces, and the key.

![Figure 17.1 Rigid Couplings](Reference 58)
17.1.2 Flexible Shaft Couplings

Flexible couplings are used to connect collinear shafts subject to one or more kinds of misalignment, while reducing the effect of shock and impact loads that may be transferred between shafts. Flexible couplings can be classified into three groups: couplings employing material flexibility, couplings employing mechanical flexibility, and couplings utilizing both types of flexibility.

Figure 17.2 shows examples of flexible couplings. Flexible couplings employing rigid parts (mechanical flexibility) transmit torque without backlash or angular play other than that due to manufacturing tolerances and wear. These types of couplings are generally incapable of dampening the transmittal of shock and impact loads. The bellows type coupling is used in applications involving large amounts of shaft misalignment, combined with low radial loading. The disc-type coupling can accommodate a smaller amount of angular misalignment than the bellows type, but by adding additional metallic "disks", radial load (torsional) carrying capacity can be greatly increased.

Flexible couplings containing resilient components (material flexibility) can accommodate shaft misalignment, as well as dampen shock and impact loads. This type of coupling possesses torsional flexibility, often acting as "detuning" devices by
altering the vibration properties of the connected system. The flexible insert type coupling shown in Figure 17.2 transmits torque through an oil resistant rubber spider assembled between two pairs of axially overlapping rigid jaws.

Examples of the third type of flexible coupling type (material and mechanical flexibility) are the metallic grid and the diaphragm coupling. The metallic grid coupling consists of two metal half-bodies with slots cut into the peripheries to seat a serpent like spring steel alloy grid. The deflection of the springs under load help to reduce shock loading; thereby, utilizing the best characteristics of material and mechanical coupling flexibilities.

Table 17-1 indicates the range of performance characteristics possible with flexible couplings. Due to the varying ranges of performance, it is easy to see why some caution must be taken in selecting the correct coupling for the operating conditions to be encountered. Diaphragm couplings are rated for high speed operation, but accept only a small amount of misalignment. Although elastomeric and gear type couplings can accommodate a fair amount of angular misalignment, in general, the gear coupling is rated to accept heavier duty (horsepower). Again, a tradeoff must be made, since gear couplings require periodic maintenance in the replenishment of lubrication and seals.

Table 17-1. Typical Flexible Coupling Performance Characteristics

<table>
<thead>
<tr>
<th>COUPLING TYPE</th>
<th>TOLERABLE MISALIGNMENT</th>
<th>MAXIMUM SPEED (rpm)</th>
<th>HP PER 100 RPM</th>
<th>TORSIONAL RIGIDITY (lb-in/deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>ANGULAR (degrees)</td>
<td>PARALLEL (inches)</td>
<td>AXIAL (inches)</td>
<td></td>
</tr>
<tr>
<td>Bellows</td>
<td>5-10</td>
<td>0.008-0.010</td>
<td>0.035-0.055</td>
<td>34,300-12,300</td>
</tr>
<tr>
<td>Diaphragm</td>
<td>0.17</td>
<td>0.166</td>
<td>0.127</td>
<td>21,250-7,000</td>
</tr>
<tr>
<td>Disc</td>
<td>.5/disc pac</td>
<td>0.06-0.20</td>
<td>5,800-1,900</td>
<td>1.2-170</td>
</tr>
<tr>
<td>Elastomeric</td>
<td>3.0</td>
<td>0.031</td>
<td>0.031</td>
<td>5,500-1,830</td>
</tr>
<tr>
<td>Gear</td>
<td>3.0</td>
<td>0.034-0.145</td>
<td>1,600-1,900</td>
<td>40-2,507</td>
</tr>
<tr>
<td>Metallic Grid</td>
<td>0.06-0.25</td>
<td>0.002-0.02</td>
<td>0.012-0.05</td>
<td>10,000-540</td>
</tr>
</tbody>
</table>

A representative graphic comparison of flexible coupling types is shown in Figure 17.3. Figure 17.3(a) shows the clear superiority of an elastomeric material flexible coupling over the other two types when angular and parallel shaft misalignments are excessive. However, the mechanical flexibility of the disk-type or metallic grid can carry a much greater load at smaller degrees of shaft misalignment. This is illustrated in Figure 17.3(b).
17.2 FAILURE MODES OF COUPLINGS

Table 17-2 lists various failure modes encountered when using rigid or flexible couplings. Many of these failures can be avoided by properly selecting the correct type, size and rating for the intended operational environment.

Coupling reliability is affected by the method used to mount the coupling hub on the shafts of connected equipment. The preferred procedure is to use an interference fit between hub and shaft of approximately 0.005 inches per inch of shaft diameter. Although clearance fit connections work satisfactorily for certain types of machinery, this practice should be avoided on any critical piece of equipment, because the reliability of the system can be affected. The most common cause of failure in this instance would be fretting of the coupling bore and of the shaft, and rolling of the key within the keyway due to looseness in the connection.

![Figure 17.3 Coupling Characteristics](image)

**Figure 17.3 Coupling Characteristics**

Initial alignment of machinery is one of the most critical factors affecting coupling performance and reliability; this is true regardless of the type of coupling employed. It should be remembered that flexible couplings are basically in-line devices which are intended to compensate for small amounts of shaft misalignment caused by bearing wear, foundation settling, thermal growth, etc. The more attention paid to initial alignment, the larger the reserve margin that will exist for accomplishing the intended purpose of the coupling. There are definite advantages to be gained from aligning equipment to more precise values than those recommended by the manufacturer. The
primary advantage, of course, is that the reserve margin for accepting misalignment during the life of the machinery is thereby increased.

Another factor to be considered, and one which is most important to satisfactory performance, is adherence to the manufacturer's bolt torquing recommendations. Loose bolts can induce fretting corrosion, as well as hammering and pounding which will eventually destroy the bolts and coupling discs.

Equipment maintenance is probably the most important factor affecting the life of the operating experience of gear couplings in the petroleum refining industry. Indications from the field show that at least 75% of all coupling failures are due to lack of lubrication. It should be kept in mind that even a well aligned gear type coupling requires periodic replenishment of the lubricant due to heat, oxidation, etc.

Table 17-2. Failure Modes of Couplings

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Worn flexing element or shaft bushings</td>
<td>- Excessive shaft misalignment</td>
</tr>
<tr>
<td>Ruptured elastomeric flexing element</td>
<td>- Torsional shock overload</td>
</tr>
<tr>
<td>Sheared hub pins or teeth</td>
<td>- Torsional shock overload</td>
</tr>
<tr>
<td>Fatigue of flexing element, hub pins, or discs</td>
<td>- Excessive starts and stops</td>
</tr>
<tr>
<td></td>
<td>- Torsional vibration</td>
</tr>
<tr>
<td>Shaft bearing failures</td>
<td>- Lubricant failure</td>
</tr>
<tr>
<td></td>
<td>- Excessive shaft misalignment</td>
</tr>
<tr>
<td></td>
<td>- Operational temperature extremes</td>
</tr>
<tr>
<td>Loose hubs on shaft</td>
<td>- Torsional shock overload</td>
</tr>
<tr>
<td></td>
<td>- High peak-to-peak torsional overload</td>
</tr>
<tr>
<td>Worn gear teeth</td>
<td>- Lubrication failure</td>
</tr>
<tr>
<td></td>
<td>- High peak-to-peak torsional overload</td>
</tr>
<tr>
<td>High pitched, staccato, or clacking noise</td>
<td>- Excessive shaft misalignment</td>
</tr>
<tr>
<td></td>
<td>- High peak-to-peak torsional overload</td>
</tr>
<tr>
<td></td>
<td>- Lubricant failure</td>
</tr>
<tr>
<td>Swollen, distorted, or cracked elastomeric flexing member, severe hub corrosion</td>
<td>- Chemical attack</td>
</tr>
<tr>
<td></td>
<td>- Excessive heat</td>
</tr>
</tbody>
</table>
Some failure modes can be experienced with the accumulation of operating time. Specifically, shaft misalignment can develop after many cycles of operation as a result of:

**Settling Foundations** - Once a coupled system's bed plate is grouted, it may experience settling due to foundation conditions. During welding operations of the bedplate, residual stresses may warp the base, causing difficulty during initial alignment.

**Thermal Growth** - Due to differences between component material thermal expansion coefficients, at elevated operating temperatures, shaft centerlines may "grow" to be farther apart than at room temperature. In situations where a system will normally operate at elevated temperatures (i.e. steam turbine driven equipment, etc.), the zero misalignment condition should be set at these elevated temperatures. In addition, total operational scenarios must be considered. If the system has a substantial cool-down or warm-up period of operation, then consideration must be given as to whether or not operations can be sustained during these periods of misalignment.

**Connecting Piping Reactions** - If, during extended operation, piping braces loosen or fail, the coupled components may have to support excessive reaction loads from connecting pipes. This can put a severe strain on bearings and coupling alignment.

**Vibration** - Excessive vibration can act to bring about material fatigue, fastener loosening, or stress corrosion cracking. After extended operation, component wear can open clearances and augment vibration amplitudes. Increased vibration can act to worsen shaft misalignment.

**Bearing Wear** - Lack of lubrication, contamination of the bearings, and wear can deteriorate performance of a bearing over a period of time. Bearing failure can increase vibration, which can in turn, add to shaft operational misalignment.

### 17.3 CHARACTERISTIC COUPLING EQUATION

In machine design, it often becomes necessary to fasten or join the ends of two shafts axially so that they will act as a single unit to transmit power. This transmission is characteristically described in Equation (17-1):

$$\sigma_c = \frac{3.96 \times 10^5 H_{IH}}{4\pi N I} \quad (17-1)$$

Where:
- $\sigma_c$ = Yield strength of the coupling in shear, lbs/in$^2$
- $H_{IH}$ = Input shaft horsepower ft-lbf/sec
- $d$ = Outside diameter of coupling, in
\[ N = \text{Shaft speed, rpm} \]

\[ I = \text{Polar moment of inertia of coupling, in}^4 \]

Calculated stress from Equation (17-1) can be converted to an expected failure rate for the coupling through various empirical data developed by various manufacturers for different types of couplings. An example of the type of data that coupling manufacturer's develop when testing their coupling designs is shown in Figure 17.4. The data is also design specific and carries speed and load limitations. The stress calculated from Equation (17-1), can be converted to a base failure rate using relationships such as those shown in Figure 17.4.

![Figure 17.4 Stress as a Function of Cycles to Failure for a Disc-type Flexible Coupling (Reference 66)](image)

17.4 FAILURE RATE MODEL FOR COUPLING

In the event that manufacturer's data is not readily available, a base failure rate for a coupling can sometimes be derived from the sum of its component parts. A typical gear coupling unit is shown in Figure 17.5. In this example of a gear coupling unit, the failure rate is given by:

\[
\lambda_{CP} = \lambda_{GR} + \lambda_{SE} + \lambda_{H} \quad (17-2)
\]
Where:  

\[ \lambda_{CP} = \text{Failure rate of coupling, failures/million cycles} \]

\[ \lambda_{GR} = \text{Failure rate of gears, failures/million cycles (See Chapter 8)} \]

\[ \lambda_{SE} = \text{Failure rate of seals, failures/million cycles (See Chapter 3)} \]

\[ \lambda_{H} = \text{Failure rate of coupling housing including hubs, failures/million cycles (See Section 17.8)} \]

Figure 17.5 Gear Coupling  (Reference 62)

A typical failure rate of a coupling is 5.0 failures per million hours. However, the failure rate of a coupling depends more on application than design or the manufacturing process. Rapid wear of the flexible member is typical of excessive shaft misalignment. Torn rubber insert, sheared rings or ruptured elastomer caused by high torsion impact loads or vibration are other failure considerations.

The torque capacity of a coupling is defined as its ability to transmit a required torque load. However, because of other factors, such as backlash, a coupling is selected with rated torque capacity many times greater than needed. Couplings are frequently specified in horsepower capacity (a function of torque and speed) at various speeds. Chapter 17 provides an equation to use this horsepower rating in estimating the coupling failure rate. Table 23.5 below contains some service factors to modify the expected failure rate of a coupling for the expected operating environment.
Table 17.3  Service Factors for a Mechanical Coupling

<table>
<thead>
<tr>
<th>Driven Machinery</th>
<th>Normal Torque Characteristic</th>
<th>High or Non-uniform Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uniform</td>
<td>1.1</td>
<td>1.2</td>
</tr>
<tr>
<td>Light shock</td>
<td>1.2</td>
<td>1.3</td>
</tr>
<tr>
<td>Medium shock</td>
<td>1.3</td>
<td>1.4</td>
</tr>
<tr>
<td>Heavy shock</td>
<td>1.4</td>
<td>1.5</td>
</tr>
</tbody>
</table>

17.5 UNIVERSAL JOINT (INTERSECTING SHAFT CENTERLINE COUPLING)

The universal joint, also known as the Cardan or Hooke Coupling consists of two yokes, connected by a cross through four bearings. The universal joint is sometimes used in place of couplings where the only misalignment between shafts is angular and permanent. It cannot be used to compensate for parallel misalignment of axial play. When this device, pictured in Figure 17.6, is operated at a joint angle, $\delta$, nonuniform motion is developed. When the driving yoke of the joint is operating at a uniform rotational velocity, the driven yoke rotates non-uniformly with respect to angular displacement, velocity, and acceleration.

Figure 17.6  Typical Universal Joint (Ref. 57)
17.6 CHARACTERISTIC EQUATION FOR UNIVERSAL JOINT

A relationship for the output shaft velocity \( \omega_2 \) as a function of the input velocity \( \omega_1 \), the angle between shafts \( \delta \), and the angular displacement of the input shaft \( \theta \) is given by (Reference 57):

\[
\omega_2 = \frac{\omega_1 \cos \delta}{1 - \sin^2 \delta \sin^2 \theta}
\]  

(17-3)

Where:

- \( \omega_2 \) = Output angular velocity of driven shaft, rev/sec
- \( \omega_1 \) = Instantaneous angular velocity of driving shaft, rev/sec
- \( \delta \) = Joint angle, degrees
- \( \theta \) = Instantaneous angular rotation, degrees

The characteristics of Equation (17-3) are plotted in Figure 17.7. The motion of the joint has the following characteristics (Reference 67):

- The average angular displacement and velocity is uniform. That is, if the driving yoke rotates one revolution, the driven yoke also rotates one revolution. However, during this one revolution, the incremental angular displacement \( \theta \) and instantaneous angular velocity \( \omega_2 \) and acceleration are not transmitted uniformly through the joint.

- The angular displacement of the driven yoke \( \theta_2 \) during one revolution lags and leads the driving yoke twice.

- Assuming constant input motion of the driving yoke, the driven yoke has a maximum difference of output angular velocity \( \omega_2 \) with respect to the driving yoke when the driving yoke lies in the plane described by the joint angle \( \delta \), and also, when the driving yoke is normal or perpendicular to this plane. The driven yoke has the same instantaneous angular velocity \( \omega_1 = \omega_2 \) at approximately 45 degrees from the joint angle plane for small joint angles.

- The maximum instantaneous angular acceleration and deceleration of the driven yoke occur when the angular velocity of the driven yoke is the same as that of the driving yoke \( \omega_1 = \omega_2 \). Also, the maximum acceleration and deceleration coincide with maximum lag and lead respectively.
• The incremental angular displacement, velocity, and acceleration increase as the joint angle (δ) increases, but at an increasing rate.

![Figure 17.7 Velocity Characteristics of a Universal Joint](image)

The principal advantages of the universal joint are its relatively low cost to manufacture as well as simple and rugged construction, combined with long life and ease of serviceability. In addition to providing the necessary torque capacity in a limited operating space, the joint has the thrust capability to withstand relatively high, externally imposed axial forces which may be produced, for example, by a sliding spline when shaft length changes are required during vehicle suspension movements.

The universal joint becomes a much more useful coupling device when it is used in tandem with another universal joint. In such applications, a constant velocity ratio between input drive shaft and output drive shaft is established. The restrictions for this to occur include:

• The input and output shaft yokes must lie in the same plane
• The angles between driver, driven, and connecting shafts must be equal (See Figure 17.8)
17.7 FAILURE RATE MODEL FOR UNIVERSAL JOINT

The failure rate of a universal joint can be presented as a sum of the failure rates of its individual component parts:

\[ \lambda_{UJ} = \lambda_{BE} + \lambda_{SE} + \lambda_H + \lambda_F \]  

(17-4)

Where:

- \( \lambda_{UJ} \) = Failure rate of universal joint, failures/million cycles
- \( \lambda_{BE} \) = Failure rate of bearings, failures/million cycles (See Chapter 7)
- \( \lambda_{SE} \) = Failure rate of seals, failures/million cycles (See Chapter 3)
- \( \lambda_H \) = Failure rate of coupling housing, failures/million cycles (See Section 17.8)
- \( \lambda_F \) = Failure rate of coupling fasteners, failures/million cycles (See Chapter 16)

The life expectancy of a joint is a function of the application requirements such as torque, speed, and joint angle, as well as other factors. Therefore, the basic load-
speed-life-stress relationships applicable to rolling element bearings are useful in life computations for universal joints employing similar rolling elements. These relationships have been established in the bearing chapter of this handbook. Although the contribution from terms other than in Equation (17-4) must be examined, in many cases $\lambda_{BE}$ can drive the relationship.

17.8 FAILURE RATE OF THE COUPLING HOUSING

The housing for a mechanical coupling is a very reliable component. Defined as $\lambda_H$, the housing failure rate will have a greater effect on the reliability of the coupling unit from the standpoint of how it affects other less reliable components. For instance, the housing may have an average life expectancy of 10 years where a seal or bearing may have only one or two years. However, the type of housing used in the coupling assembly can have a large effect on the lifetime of the bearings and seals. This is due to differing loads placed on the shaft by the lubricant. The failure rate of the coupling housing ($\lambda_H$) itself can be estimated at 0.001 failures/million cycles.

17.9 REFERENCES


63. Thomas Couplings Applications Manual

64. Bolam, J.R., "Coupling Alignment: The Reverse Indicator Method Simplified", P/PM Technology, July/Aug 90


66. Robertson, R., and Smith, B., "Why Flexible Couplings Fail", Plant Engineering and Maintenance, Jun 89


71. Dennis N. Pratt, "Investigation of Spline Coupling Wear", Report No. SY-51R-87, December 1987, Naval Air Warfare Center, Patuxent River, Maryland
CHAPTER 18

SLIDER CRANK MECHANISMS

18.0 TABLE OF CONTENTS

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18.1 INTRODUCTION

The slider crank mechanism is usually not thought of as an independent mechanical component but rather as an integral part of a more complex piece of equipment such as the piston rod/piston components of an internal combustion engine. Figure 18.1 shows a typical slider crank mechanism, the normal function of this particular device being the conversion of rotational force into a linear force or vice versa.

The typical slider crank mechanism includes bearings, rods, linkages, seals and a sliding surface such as a cylinder wall. Wear of these parts becomes the primary failure mechanism, the failure modes and effects being dependent upon the application. The geometry of the design plays an important part of the reliability analysis since the mechanical advantage and the wear pattern are greatly influenced by the positioning of parts.

18.2 FAILURE MODES OF SLIDER CRANK MECHANISMS

The more predominant failure modes of a slider crank mechanism can be readily identified with frictional action on like or dissimilar materials. The component parts of a slider crank mechanism are subject to wear in varying degrees and the normal approach to reliability analysis is to establish the expected life of the individual parts in the projected operating environment.
Bearing wear will usually be influenced by the lubrication film thickness maintained, the side load on the bearing, the contamination level, and corrosion. Chapter 7 presents an approach to evaluating bearing life for these considerations.

Slider crank wear will manifest itself in several ways to cause the degradation of the slider crank to the point of failure. This threshold of failure must be defined in terms of jamming friction, side movement, limit of travel, alignment of parts, etc. Some of the failure modes to be considered are included in Table 18-1.

### Table 18-1. Typical Failure Modes of Slider Crank Mechanisms

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linkage does not move in intended direction</td>
<td>Bearing parts separate causing jam</td>
</tr>
<tr>
<td>Restricted travel limited of linkage</td>
<td>Excess load combined with loss of bearing material</td>
</tr>
<tr>
<td>Broken linkage</td>
<td>Fatigue of linkage member</td>
</tr>
<tr>
<td>Linkage alignment out of tolerance</td>
<td>Bearing deformed</td>
</tr>
</tbody>
</table>

The function of a mechanical seal is to provide a barrier between the moving or rotating surfaces and prevent the trapped fluid from migrating into undesired areas. For example, in the case of an engine cylinder the rings on the piston prevent the
combustion gases from going into the lubricant and also prevent loss of energy due to combustion gas by-pass. The O-ring and flat seals are designed to prevent the lubricant from contaminating other part areas and also, the loss of the lubricant. Seals and gaskets are described in Chapter 3 and the models contained therein should be used for the analysis of seal and gasket reliability.

Rods and linkages within the slider crank mechanism are subject to fatigue and may crack although this failure mode is rare.

18.3 MODEL DEVELOPMENT

The failure rate model for the slider crank mechanism can be expressed by the following equation:

$$\lambda_{SC} = \lambda_{BE} + \lambda_{RD} + \lambda_{SE} + \lambda_{RI} + \lambda_{SM}$$  \hspace{1cm} (18-1)

Where:
- $\lambda_{SC}$ = Total failure rate for slider crank, failures/million operations
- $\lambda_{BE}$ = Failure rate for bearings, failures/million operations  
  (See Section 18.3.1)
- $\lambda_{RD}$ = Failure rate for rods/shafts, failures/million operations  
  (See Chapter 10, Section 10.4)
- $\lambda_{SE}$ = Failure rate for seals/gaskets, failures/million operations  
  (See Chapter 3, Section 3.2)
- $\lambda_{RI}$ = Failure rate for rings/dynamic seals, failures/million operations  
  (See Chapter 3, Section 3.3)
- $\lambda_{SM}$ = Failure rate for slider mechanism, failures/million operations  
  (See Section 18.3.5)

Failure rates are determined for the individual parts comprising the slider crank mechanism. Then a total failure rate of the slider crank mechanism is determined using Equation (18-1).

18.3.1 Bearings

One of the predominant failure modes of a slider crank mechanism is caused by a malfunctioning bearing surface. Both roller and sliding bearings can be included in a slider crank design. Failure rate equations for roller/ball bearings are included in Chapter 7.
Typical sliding bearings are shown in Figure 18.2. The sliding bearing is usually comprised of three elements including the inner surface member, the outer surface member, and the lubricant film separating the inner and outer members. The sliding bearing is characterized as a shaft rotating within a sleeve. Sliding bearings can be classified by material, load direction, lubrication method, and configuration. Sliding bearings are well suited for large loads encountered in slider crank mechanisms. Although sliding bearings may have less running friction than rolling bearings, their starting friction is much higher. Rolling bearings are also easier to lubricate during service life. Sliding bearings are well suited to low speed applications where shock and vibration occur such as punch presses and steam hoists. And for many applications such as hoists, sliding bearings need only minimal lubrication.

Hydrodynamic sliding bearings are characterized by the load being carried by a film of oil generated by rotation and suitable oil grooves. The friction at start-up is large due to direct contact between the journal and sleeve while the friction during operation is moderate, the bearing acting like a low efficiency pump. The life of the bearing then is limited due to wear at start-up and stopping.

![Figure 18.2 Sliding Bearing Classifications](Reference 19)

The reliability of the sliding crank mechanism will, of course, be affected by the lubricant being used. All liquids provide lubrication but some do better jobs in particular applications. Dry lubricants, for example, will adhere very well to the bearing surfaces but tend to wear quite rapidly as petroleum oils. Their capacity to minimize friction, however, is only fair. The concept of lubricant viscosity is illustrated in Figure 18.3.

A film of lubricant adheres to the stationary plate and supports a moving plate. In order to move the upper plate to the right at a constant velocity \( V \), it is necessary to
exert some constant force $F$. Thus a shear stress is applied at the wetted surface of the moving plate equal to (Reference 19):

$$\tau = \frac{F}{A} \quad (18-2)$$

Where: $\tau =$ Shear stress, lbs/in$^2$

$F =$ Applied Force, lb

$A =$ Area of plate surface in contact with lubricant, in$^2$

The rate of shearing strain $R'$ is defined as the ratio of the velocity $V$ to the thickness $h$ of the lubricant film (Reference 19):

$$R' = \frac{V}{h} \quad (18-3)$$

The ratio of shearing stress to rate of shearing strain is called the dynamic viscosity, $\nu$ (Reference 19):

$$\nu = \frac{\tau}{R'} = \frac{F h}{AV} \quad (18-4)$$
Three types, or regimes, of lubrication occur in practice. They differ in the degree to which the lubrication is carrying the load. A full film lubrication physically separates the shaft and bearing surfaces by a relatively thick lubricant film of about 15 microns. This film prevents any metal-to-metal contact at the operating conditions. The coefficient of friction will be low, usually not above about 0.005. Full film operation implies minimum power losses and maximum life expectancy of the parts.

Complete boundary lubrication means that the bearing and shaft surfaces are being rubbed together with only a very thin lubricant film adhering to each surface and preventing direct contact. The coefficient of friction is high, in the range of 0.1.

Mixed film lubrication means that there is both boundary and hydrodynamic lubrication. Part of the load is carried by small pools of self-pressurized lubricant. Other areas of the surfaces are rubbing with only a thin film of lubricant separating the peaks. A typical friction coefficient for this regime is 0.02.

Figure 18.4 shows the coefficient of friction plotted against a bearing characteristic number $\frac{\nu n}{P}$. The three operating variables in this bearing characteristic number are the lubricant's viscosity, shaft speed, and unit bearing load:

$$\alpha = \frac{\nu n}{P}$$  (18-5)

Where

- $\alpha$ = Bearing characteristic number
- $\nu$ = Lubrication viscosity, lb-min/in$^2$
- $n$ = Shaft speed, revolutions/sec
- $P$ = Unit bearing load, lbs/in$^2$

The unit bearing load is defined as the ratio of the bearing's load to its projected area (Reference 19):

$$P = \frac{W}{D \times L}$$  (18-6)

Where:

- $W$ = Bearing load, lb
- $D$ = Bearing diameter, in
- $L$ = Bearing length, in
This bearing characteristic number provides a method of determining any potential problem with lubricant film. Any low viscosity, low shaft speed, or high unit bearing load implies a low value for $\nu n/P$. Conversely, the higher $\nu n/P$, the easier it is to establish a full-load-supporting film.

For the largest values of $\nu n/P$, there is full-fluid-film, or hydrodynamic, lubrication. In this regime of operation the coefficient of friction attains a minimum of about 0.001. A greater $\nu n/P$ value will assure an adequately thick film and a margin of safety with a somewhat greater power loss. The lowest values of $\nu n/P$ correspond to the regime of complete boundary lubrication. The friction coefficient remains constant throughout this regime; its actual value depends on the character of the surfaces and the lubricant. The mid-regime is that of mixed-film lubrication. In this regime a decrease in $\nu n/P$ is accompanied by a sharp increase in friction coefficient.

Most of the bearings utilized in mechanical devices are considered light service and operate in the mixed-film or boundary lubrication regimes. Typical office equipment and appliances with latching mechanisms contain bearings with little or even no lubrication and without the proper operating conditions to develop a full-lubricant film. Yet they
survive and they provide a low-cost solution to the problem of supporting and controlling machine members in relative motion.

Manufacturers of bearings for light service usually base bearing selection on the PV factor, the product of unit bearing load $P$ and rubbing velocity $V$. This factor indicates what bearing temperature will be reached and what rate of wear can be expected. Temperature rise and wear rate are maintained within reasonable limits by controlling the PV factor.

The unit bearing load $P$, already defined by Equation (18-6) is the ratio of the bearing’s load to its projected area. The rubbing velocity $V$ must be calculated differently for oscillating shaft motion than for continuous rotation. For continuous rotation the rubbing speed is (Reference 19),

$$ V = 5\pi D n $$

(18-7)

Where: $V$ = Rubbing velocity, ft/min  
$D$ = Bearing diameter, in  
$n$ = Shaft speed, revolutions/sec

However, if the shaft is oscillating relative to the bearing, the design value for $V$ is based on the average rubbing speed (Reference 19).

$$ V = \frac{\pi D \theta f}{72} $$

(18-8)

Where: $\theta$ = Total angle traveled per cycle, degrees  
$f$ = Frequency of oscillation, cycles/sec

The use of sleeve or journal bearings in severe service requires a full bearing or a thick lubrication film to support the load. While an external pump may be used to supply a lubricant under pressure to the bearing’s feed hole, within the bearing itself it is the shaft that acts as a pump and pumps the oil adhering to it into the wedge-shaped oil film that supports the load. With the shaft stationary, the shaft simply rests on the bottom of the bearing. But at start-up the shaft begins to roll up the bearing wall. As it climbs, it also begins to pump oil between itself and the bearing. As this oil is pumped, the shaft lifts off the bearing surface and moves in the direction of rotation. At operating speed, the shaft has developed a wedge-shaped film between itself and the bearing that supports the shaft and its load. The radial displacement of the shaft’s center from the bearing’s center is the eccentricity, $e$. The pressure distribution in the oil film achieved
depends on factors such as shaft speed, load, lubricant viscosity, bearing clearance, and length-to-diameter ratio.

The failure rate of a bearing is obtained using the equations contained in Chapter 7. The information obtained from this section can be used in conjunction with the procedures in Chapter 7 to estimate the failure rate of the bearing.

18.3.2 Rods/Shafts

The reliability of the rod or shaft is generally very high when compared to the components in the sliding-action system. Generally, the life expectancy will be at least three times that of the bearing. The possibility that the rod or shaft will fracture can best be determined using finite element techniques. The effects of the rod or shaft breakage on adjacent components are of greater importance than the reliability of the rod or shaft itself.

Procedures for determining the base failure rate and the multiplying factors for a shaft can be found in Chapter 10. The failure rate of a rod can be estimated at 0.001 failures/million hours for fracture.

18.3.3 Seals/Gaskets

The failure rate of a seal or gasket is determined by ability of the seal to restrict the flow of fluid from one region to another for the intended life is prescribed operating environment. Section 3.2 of Chapter 3 contains procedures and equations for determining $\lambda_{SE}$.

18.3.4 Rings, Dynamic Seals

The sealing surface of rings and other dynamic seals are perpendicular to the shaft with contact between primary and secondary rings to achieve a dynamic seal at various speeds, pressures and temperatures. The procedures contained in Chapter 3 can be used to determine the base failure rate and multiplying factors for rings.

18.3.5 Slider Mechanism

The wear life of the sliding surface area depends on the correlation of wear of the two surfaces involved with the material strength and the stress imposed on the sliding action mechanism. From a time standpoint, wear of the two surfaces will occur in two phases. The first or constant wear phase is characterized by the shearing of asperities due to sliding action. During this period the wear rate is practically linear as a function of the number of mechanical cycles and the wear depth at the end of the constant wear phase is one half the original surface finish. During the second or severe wear phase, wear debris becomes trapped between the two sliding surfaces and gouging of the surfaces takes place. The wear rate begins to increase very rapidly and failure of the sliding action mechanism is imminent. Chapter 9, Section 9.3 contains the procedure.
and equations to determine the number of cycles at which point the slider crank mechanism is determined to have failed due to wear.

18.4 REFERENCES


19.1 INTRODUCTION

A sensor is a hardware device that measures a physical quantity and produces a signal which can be read by an observer or by an instrument. For example, a thermocouple converts temperature to an output voltage which can be read by a voltmeter. For accuracy, all sensors are calibrated against known standards. A transducer is a device that is actuated by power from one system and supplies power usually in another form to a second system. For example, a loudspeaker is a transducer that transforms electrical signals into sound energy. A standard definition for a sensor or transducer does not exist and the words sensor and transducer are used synonymously, specific names being given depending on the application. Also, suffixed derivatives ending in -meter such as accelerometer, flowmeter and tachometer are used. For convenience, the basic sensor will be used in this chapter to describe these units.

In almost every product for commercial and military application, the number of sensors and transducers continues to increase. The application of “smart sensors” with digital communication techniques and resulting improved accuracy and self-healing sensor networks has created an ever increasing application of sensor technology. Table 19-1 is a partial listing of the applications for sensors and transducers. One of the important newer applications of sensors to system reliability is prognostics and diagnostics. A prognostic/diagnostic system uses sensors to monitor operational and environmental conditions and translates any changes to component remaining life using the equations in this handbook embedded in the prognostic/diagnostic processor. As preestablished threshold values are exceeded, the embedded reliability models calculate component overstress and cumulative damage and provide maintenance personnel of impending malfunctions and the next best maintenance action to be performed.
Table 19-1. Typical Sensor/Transducer Applications

<table>
<thead>
<tr>
<th>Sensor/Transducer Classification</th>
<th>Typical Sensor/Transducer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal</td>
<td>Thermocouple, thermostat, thermister</td>
</tr>
<tr>
<td>Mechanical</td>
<td>Pressure, vibration, shock</td>
</tr>
<tr>
<td>Electromagnetic</td>
<td>Ohmmeter, ammeter, voltmeter</td>
</tr>
<tr>
<td>Fluid</td>
<td>Pressure, flow, viscometer</td>
</tr>
<tr>
<td>Chemical</td>
<td>Oxygen, carbon monoxide</td>
</tr>
<tr>
<td>Optical</td>
<td>Photodiode, phototransistor, optoelectronic</td>
</tr>
<tr>
<td>Acoustic</td>
<td>Ultrasound, sonar, seismometer</td>
</tr>
<tr>
<td>Motion</td>
<td>Accelerometer, tachometer, odometer,</td>
</tr>
<tr>
<td>Presence</td>
<td>Gas, liquid, metal, magnetic, proximity</td>
</tr>
<tr>
<td>Environmental</td>
<td>Humidity, temperature, altitude, smoke</td>
</tr>
</tbody>
</table>

A typical sensor arrangement is shown in Figure 19.1. Here a differential transformer is used in conjunction with a spring to measure weight. The sensing element is the spring which is compressed when the weight is applied against the transducer changing the position of the central core of the differential transformer. The transformer converts the position of the core into an electrical signal the magnitude of which is proportional to the applied weight.

The failure rate of the sensor in Figure 19.1 is essentially determined by the failure rates of the compression spring and the transformer. The failure rate of the spring can be determined by using the procedures in Chapter 4. MIL-HDBK-217 contains procedures for determining the failure rate of the transformer (Reference 28).

Other typical sensors for mechanical flow sensing are shown in Figure 19.2. In Figure 19.2(a), a spring loaded hinged vane is pushed open as fluid flows through the sensor; and in Figure 19.2(b), the fluid pushes against a spring restrained plug. Again the failure rates of the individual parts are determined using procedures in the applicable chapters of this handbook for cylindrical wear, springs, etc.
Figure 19.1  Typical Positioning Sensor Unit

Figure 19.2  Typical Flow Sensor Units
It is important to realize that determining the failure rate of a sensor is more involved than combining failure rates of the individual parts. As shown in Figure 19.3, sensors are used as part of a larger sensing/monitoring/detection system and the reliability of the total system is sensitive to its defined requirements. Accuracy, repeatability, sensitivity, drift, and other performance requirements are an important part of the failure rate estimate. Redundant sensors are often used to back up other sensors that give no warning of imminent failure and which may fail catastrophically. Self test features and scheduled test inputs to check for sensor drift can avoid system failure. Response to sensor data by the operator and/or other parts of the system must also be considered in estimating the failure rate of a sensor system.

![Figure 19.3 Typical Sensor Network](image)

Though sensors provide vital environmental information and feedback to microcontrollers, they also increase processor workload because their signals must be linearized, filtered, temperature compensated, scaled, and converted. New integrated circuits, however, can convert temperature, pressure, position movement, and acceleration signals to computer compatible equivalents. One-chip functions include a/d converters, multiplexers, current-source, and a microcomputer that controls the converter and processor sensor data. Sensor-specific information such as calculated values and calibration factors can be stored in a local memory unit. The fact that each and every sensor system is different requires that the sensor be separated into its individual parts for analysis using the appropriate chapters of the Handbook to estimate each failure rate.

In redundant sensor configurations a possible design method is the use of diversity. Diversity is the use of different technologies to perform a required function. This practice may involve such an approach as sensors from various manufacturers with different sensor technology and software. Diversity has the advantage that it can reduce the probability that two or more sensors fail simultaneously, although this effect
is limited by the fact that two or more sensors may still contain the same faults or experience a common cause failure mode. A common cause failure mode is the failure of multiple items from a single cause which is common to all of them such as a power supply failure. A disadvantage of diversity can be the increased complexity of maintenance, which in itself can lead to a higher probability of failure of the sensor configuration. Whether the use of diversity is advisable depends on the design of the sensor network and the application.

A sensor network can be self healing thus increasing the overall reliability of the sensor network. A sensor network typically has three types of nodes: sensor nodes that monitor the immediate environment, target nodes generate various stimuli for sensor nodes, and user nodes that handle the administration of the sensor network. Typical sensor-to-sensor links

19.2 FAILURE MODES OF SENSORS AND TRANSDUCERS

Typical failure modes for individual sensors and transducers are shown in Table 19-2. These example failure modes must be modified for the particular application. A failure mode and analysis performed on the sensor assembly will help focus on the modifying factors to be used on published failure rates. Failure modes need to be addressed for the specific sensor network being analyzed. Typical subsystems of a sensor network include:

- Data processing subsystem – provide and process the measured quantities for further real-time use by the human and communication interfaces and/or at the electrical output subsystem.
- Sensor subsystem – convert the physical or chemical quantities into conditioned electrical signals for use in the data processing unit.
- Human interface – read the information on a display and enter request for data
- Communications interface – connect the instrument to external systems
- Electrical output subsystem – convert the digital information into one or more analog signals
- Power supply unit – supply power to the other smart sensor subsystems.

Reliability is defined differently in real time applications than for delay tolerant applications. In real time applications reliability is dependent upon the arrival time of sensor data while delay tolerant configurations data loss is the factor determining reliability. There are significant differences in failure modes when evaluating a sensor
network for reliability. Additional failure modes to be considered include but not limited to information overload, timing errors, node interference, power depletion, common cause failure, and increased probability of digitization errors. Table 19-3 lists some typical failure modes of sensor network applications.

Table 19-2. Typical Failure Modes of Individual Sensors

<table>
<thead>
<tr>
<th>FAILURE MODES</th>
<th>FAILURE CAUSES</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Incorrect signal from sensor element</td>
<td>- Reduced signal level</td>
<td>Potential processing error</td>
</tr>
<tr>
<td></td>
<td>- Impedance mismatch</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- A/D conversion error</td>
<td></td>
</tr>
<tr>
<td>Loss of signal from sensor element</td>
<td>- Chip failure</td>
<td>Loss of signal to processor</td>
</tr>
<tr>
<td></td>
<td>- Corroded sensor</td>
<td></td>
</tr>
<tr>
<td>Complete loss of signal in transmission line</td>
<td>- Broken wire</td>
<td>Loss of signal to processor</td>
</tr>
<tr>
<td></td>
<td>- Fiber optic, RF interruption</td>
<td></td>
</tr>
<tr>
<td>Signal error in transmission line</td>
<td>- Power line interference</td>
<td>Potential processing error</td>
</tr>
<tr>
<td></td>
<td>- Contaminants in fluid system</td>
<td></td>
</tr>
<tr>
<td>Incorrect signal in computation device</td>
<td>- Error in algorithm</td>
<td>Processing error</td>
</tr>
<tr>
<td>Power supply loss of voltage</td>
<td>- Power supply malfunction</td>
<td>Loss of signal to processor</td>
</tr>
<tr>
<td>Improper response to information recipient</td>
<td>- Incorrect interpretation</td>
<td>Potential system malfunction</td>
</tr>
<tr>
<td></td>
<td>- System malfunction</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Error in algorithm</td>
<td></td>
</tr>
<tr>
<td>Calibration error</td>
<td>- Software error</td>
<td>Potential system malfunction</td>
</tr>
<tr>
<td></td>
<td>- Error in algorithm</td>
<td></td>
</tr>
<tr>
<td>Battery energy depletion</td>
<td>- Battery malfunction</td>
<td>Loss of signal to processor</td>
</tr>
</tbody>
</table>

Timing in conventional sensors is not normally an issue. The sensor continuously measures the parameter and presents a resulting 4-20mA signal. The time lag between measurement and output is normally negligible. In a sensor network architecture based on interrupts, the sensor might receive many interrupts of high priority and might subsequently fail to attend to the tasks of lower priority. For example, the lower priority task involves updating the sensor’s display and the user gets an incorrect reading on the display. Another potential problem with a sensor network is data integrity. With conventional sensors, parameters are set using fixed components such as resistors,
and are therefore highly insensitive to external influences. In the case of a sensor network, the sensor may lose the parameter settings stored in memory during a power failure. Failure modes of the communications link in sensor networks include information overload of the communications interface causing events happening simultaneously to be communicated sequentially, thus confusing the operator when deciding what caused a disturbance. Conventional sensors use a 4-20mA loop for communication which can be used over very long distances. Digital buses often require repeaters increasing sensor network complexity and higher failure rate.

19.3 SENSOR FAILURE RATE MODEL DEVELOPMENT

The unique designs of sensors and special applications in terms of sensitivity, drift, etc. make it difficult to assign an accurate base failure rate for a sensor. A reliability model can be written as follows:

\[ \lambda_{TD} = \lambda_{TD,B} + \lambda_S + \lambda_T + \lambda_C + \lambda_P + \ldots + \lambda_X \] (19-1)

Where:
- \( \lambda_{TD} \) = Failure rate of sensor, failures/million hours
- \( \lambda_{TD,B} \) = Base failure rate of sensor, failures/million hours
- \( \lambda_S \) = Failure rate of sensing element (See Table 19-3)
- \( \lambda_T \) = Failure rate of the transmission line (See Reference 28)
- \( \lambda_C \) = Failure rate of the computational device (See Reference 28)
- \( \lambda_P \) = Failure rate of the power source (See Reference 28)
- \( \lambda_X \) = Failure rate of other components comprising the sensor or sensor network

Typical published failure rates for sensors are shown in Table 19-4 for commercial applications. Various publications contain historical failure rates for sensors. The data does not usually provide the design information, the environment/operating conditions under which the sensor was used, the fluid medium in contact or any of the characteristics listed in the section on failure modes. As indicated in the table, the failure rate can vary to a great extent mainly because the failure rate of a sensor is mainly dependent upon manufacturing practices and environmental conditions. Use of these failure rates must be adjusted based on experience with similar applications and operating environment. It is recommended that the considerations provided in Table 19-5 be applied in estimating sensor and sensor network failure rates.
Table 19-3. Typical Failure Modes of Sensor Networks

<table>
<thead>
<tr>
<th>FAILURE MODES</th>
<th>FAILURE CAUSES</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Loss of voltage to all components</td>
<td>Power Supply malfunction</td>
<td>Loss of signal history, calibration and settings information</td>
</tr>
<tr>
<td>Loss of voltage to sensor subsystem/ADC</td>
<td>Wire failure</td>
<td>Loss or incorrect measurement of input signal</td>
</tr>
<tr>
<td>Loss of voltage to processor</td>
<td>Wire failure</td>
<td>Loss of signal history, calibration and settings information</td>
</tr>
<tr>
<td>Loss of voltage to transceiver</td>
<td>Wire failure</td>
<td>Loss of or incorrect output or settings</td>
</tr>
<tr>
<td>Loss of voltage to human interface</td>
<td>Wire failure</td>
<td>No display of data, setting of parameters not possible</td>
</tr>
<tr>
<td>Defect in D/A conversion</td>
<td>Hardware error</td>
<td>Incorrect output</td>
</tr>
<tr>
<td>Incorrect signal from sensor</td>
<td>Defect in A/D conversion, software error</td>
<td>Incorrect output</td>
</tr>
<tr>
<td>Loss of signal from sensor</td>
<td>Sensor failure</td>
<td>Node failure</td>
</tr>
<tr>
<td>Timing error from sensor</td>
<td>Software error</td>
<td>Incorrect output</td>
</tr>
<tr>
<td>Conditioning of data from sensor is incorrect</td>
<td>Software error</td>
<td>Incorrect output</td>
</tr>
<tr>
<td>Corrupted signals to data processing or external system</td>
<td>Information overload, software error</td>
<td>Loss of or incorrect output</td>
</tr>
<tr>
<td>Error in signal processing</td>
<td>Software error</td>
<td>Incorrect output</td>
</tr>
<tr>
<td>Error in calibration</td>
<td>Failure of communications interface, software error</td>
<td>Incorrect output</td>
</tr>
<tr>
<td>Error in data storage</td>
<td>Hardware/software error</td>
<td>Incorrect output, incorrect calibration</td>
</tr>
<tr>
<td>Incorrect display of data</td>
<td>Human error</td>
<td>Incorrect action by controller</td>
</tr>
<tr>
<td>Incorrect setting of parameters</td>
<td>Human/software error</td>
<td>Incorrect output</td>
</tr>
<tr>
<td>Loss of clock , wrong or changing frequency</td>
<td>Hardware error</td>
<td>Incorrect output</td>
</tr>
<tr>
<td>SENSOR/TRANSDUCER</td>
<td>TYPICAL FAILURE RATE $\lambda_{TD,B}$ or $\lambda_S$ (failures/10^6 hours)</td>
<td></td>
</tr>
<tr>
<td>-------------------</td>
<td>---------------------------------------------------------------------</td>
<td></td>
</tr>
<tr>
<td>Accelerometer</td>
<td>174.2</td>
<td></td>
</tr>
<tr>
<td>Hall effect switch</td>
<td>0.6</td>
<td></td>
</tr>
<tr>
<td>Pneumatic switch</td>
<td>23.3</td>
<td></td>
</tr>
<tr>
<td>Solid state device</td>
<td>13.2</td>
<td></td>
</tr>
<tr>
<td>Thermocouple</td>
<td>1.00</td>
<td></td>
</tr>
<tr>
<td>Torque sensing device</td>
<td>80.0</td>
<td></td>
</tr>
<tr>
<td>Fluid flow sensor</td>
<td>133.8</td>
<td></td>
</tr>
<tr>
<td>Air flow sensor</td>
<td>32.9</td>
<td></td>
</tr>
<tr>
<td>Pressure sensor</td>
<td>2.0</td>
<td></td>
</tr>
<tr>
<td>Fluid level sensor</td>
<td>77.8</td>
<td></td>
</tr>
<tr>
<td>Humidity sensor</td>
<td>20.4</td>
<td></td>
</tr>
<tr>
<td>Temperature transmitter</td>
<td>0.39</td>
<td></td>
</tr>
</tbody>
</table>
Table 19-5. Sensor and Sensor Network Failure Rate Considerations

1. Battery life, battery backup, uninterruptible power supply
2. Sensor connectors, connections
3. Individual sensor failures/redundant sensor connection
4. Average reading of multiple sensors
5. Sensor accuracy requirements
6. Short term and long term drift
7. Sensor positioning in target area
8. Data routing efficiency
9. Number of sensor nodes in the network
10. Data generation rate of sensor nodes
11. Energy consumption rate
12. Response timing requirements
13. Corrosive environment, protective coating
14. Operating temperature
15. Shock and vibration requirements
16. All potential failure modes listed in Table 19-2 and/or Table 19-3

19.4 REFERENCES


88. Reliability Analysis Center, “Nonelectronic Parts Reliability Data”, NPRD-95


20.1 INTRODUCTION

A shaft is a rotating member, usually of circular cross section, used to transmit power or motion. It provides the axis of rotation, or oscillation of other parts such as gears, flywheels and pulleys and controls the geometry of their motion. One of the functions of a shaft is transmitting torque from one element to another on the shaft. Common torque-transfer elements include keys, splines, setscrews and pins. They transmit power by means of rotational motion and developed torque from one end of the shaft to the other. Shaft loads include such components as fluid drivers, gears, splines and pulleys. Reliability issues for shafts include material strength, rotational speed, shear stress, temperature, and the operating environment.

Flexible shafts provide the capability of transmitting motion around corners. A flexible shaft is made by winding several layers of wire around a central core. Flexible shafts are rated by specifying the torque corresponding to various radii of curvature of casing. A wire rope may be used as a flexible shaft. The wire rope section in Chapter 23 provides some additional information on evaluating a wire rope for reliability.

A spindle is a short shaft. Failure modes to be considered in evaluating a spindle for reliability and the equations and procedures for determining the failure rate for a given application are the same as a typical shaft.
Shafts are usually designed for an infinite life. Beyond 1 million stress cycles the fatigue strength of most steels becomes constant. This value of material endurance strength divided by the shaft load provides a factor of safety for the shaft material. Materials used for the manufacture of shafts less than 3.5 inches in diameter are usually cold-rolled or cold-drawn steel. Larger shafts are usually hot-rolled and then machined to remove the decarburized surface. Shafts larger than 6 inches in diameter are usually forged and machined to size.

After the failure rate of the shaft is determined, it can be added to other failure rates of the component for a total component failure rate.

**20.2 FAILURE MODES**

Most shafts are subject to combined rotational and bending stresses. These in turn may be steady, variable, or a combination of the two. Sometimes flexible couplings are used as shaft connections to reduce bending loads. Shafts are generally designed for an infinite life and will therefore have a very low failure rate. The effect on other components is often a more serious question. For example, in the case of a shaft supporting a gear, excessive bending of the shaft will cause misalignment of gear teeth and uneven distribution of tooth load, in turn, causing excessive tooth wear.

Keyways cut into the shaft reduce its load carrying ability, particularly when impact loads or stress reversals are involved. When a completely reversing stress is applied to a stress concentration point, growth of a crack is accelerated at this point that can lead to shaft fracture. Keys and keyways should be used only to locate components on the shaft avoiding stress concentrations on the key. Changes in shaft radii contribute to stress concentrations. Large radii between shaft radii minimize stress concentrations at the shoulders.

Shafts in rotation can become very unstable at certain speeds and damaging vibrations and deflections can occur. The result is not only failure of the shaft but also to components of the machine of which the shaft is a part. The rotational speed at which this phenomenon occurs is called the critical speed of the shaft.

**20.3 MODEL DEVELOPMENT**

Shafts are primarily designed on the basis of the torsional moment which they transmit. This torque can be calculated using the following expression:

\[
T = \frac{3.96 \times 10^5 \text{ hp}}{2 \pi N} \tag{20-1}
\]
where: \( T \) = torque, in-lbs
\( hp \) = transmitted horsepower
\( N \) = Shaft rpm

and the shear stress on the shaft can be calculated from the following:

\[
S_s = \frac{16 T}{\pi d^3}
\]

(20-2)

where: \( d \) = shaft diameter, in

20.4 FAILURE RATE MODEL FOR SHAFTS

The reliability of the shaft itself is generally very high when compared to other components. Studies have shown (Reference 26) that the average failure rate for the shaft itself is about eight times less than mechanical seals and about three times less than that of the ball bearings. The possibility that the shaft itself will fracture, or become inoperable is very unlikely when compared to the more common component failure modes. Normally, it will be the seals or bearings of the component that will cause the initial problems. The effect of the shaft on reliability of other components is of greater importance than the reliability of the shaft itself.

The shaft reliability model is shown by the following equation:

\[
\lambda_{SH} = \lambda_{SH,B} \cdot C_f \cdot C_T \cdot C_{DY} \cdot C_{SC}
\]

(20-3)

Where:
\( \lambda_{SH} \) = Shaft failure rate, failures/million cycles
\( \lambda_{SH,B} \) = Shaft base failure rate, failures/million cycles
(See Section 20.4.1)
\( C_f \) = Shaft surface finish multiplying factor (See Section 20.4.2)
\( C_T \) = Material temperature multiplying factor (See Section 20.4.3)
\( C_{DY} \) = Shaft displacement multiplying factor (See Section 20.4.4)
\( C_{SC} \) = Stress concentration factor for shaft discontinuities (See Section 20.4.5)
The above reliability equation for shafts assumes a constant rotational torque loading due to transmitted power and completely reversed bending loads on the shaft due to its rotation. In the case of large reciprocating loads such as those in piston engines and compressors, this assumption must be analyzed and the failure rate adjusted accordingly.

### 20.4.1 Base Failure Rate

In a typical fatigue test, the test specimen is stressed to some value and the number of stress fluctuations to fracture noted. The applied stress is then plotted against the number of cycles to failure. For many ferrous metals and thermosetting plastics the $S-N$ (stress vs. cycles) curve approaches an asymptotic value at some stress level called the endurance or fatigue limit. Figure 20.1 shows an example of fatigue data for mild steel subjected to reversed stresses with an endurance limit of 27,000 lbs/in$^2$ (curve A). For many materials there is no definite endurance limit and a value is determined at a selected number of cycles, usually $10^8$ cycles. The base failure rate, $\lambda_{SH,B}$, is estimated from the material endurance factor by the following relationship:

$$\lambda_{SH,B} = \frac{10^6}{N}$$  \hspace{1cm} (20-4)

Where:  
$N$ = Number of cycles to failure at application stress level, $S_{ED}$

and:  
$S_{ED}$ = Material endurance limit, lbs/in$^2$

The endurance limit, $S_{ED}$, for some common steels and alloys is shown in Table 20-1.

The multiplying factors account for environmental conditions that vary from the normal operation. The base failure rate represents values that can be expected if all conditions during normal operation are those of the original design. The following discussion explains the values for each multiplying factor.

### 20.4.2 Shaft Surface Finish Multiplying Factor

$C_f$ is the shaft surface finish factor that adjusts the base failure rate by an amount depending on the type of manufactured finish. If the design calls for a particular finish, then a variation from this finish during the manufacturing process will alter the reliability of the shaft. Table 20-2 shows the values and equations for the various finishes versus material tensile strength.
20.4.3 Material Temperature Multiplying Factor

Typical material fatigue data are acquired at 160 °F. To compensate for the decline in static and dynamic strengths, creep, and thermal expansion at high temperatures, the temperature factor, $C_T$, represented by Equation (20-5) is applicable for temperatures greater than 160 °F (Reference 19).

\[
C_T = \frac{460 + T_{AT}}{620} \quad \text{for } T_{AT} > 160^\circ F
\]  \hspace{1cm} (20-5)

and:

\[
C_T = 1.0 \quad \text{for } T_{AT} \leq 160^\circ F
\]

Where: $T_{AT}$ = Operating temperature, °F

20.4.4 Shaft Displacement Multiplying Factor

The shaft displacement factor, $C_{DY}$, will vary with the amount of load the shaft will see. Shaft misalignment and excessive deflection have a significant influence on the reliability of shaft bearings and any seals mounted on the shaft. The estimate of shaft deflection depends on the weight of the shaft, its physical dimensions and any
unbalance force caused by fluid flow. The basic equation for determining the shaft displacement multiplying factor is:

\[
C_{DY} = \frac{F l^3}{E I}
\]  \hspace{1cm} (20-7)

Where:

- \( F \) = Fluid radial unbalance force or shaft weight, lb
- \( l \) = Shaft length, in
- \( E \) = Modulus of elasticity of shaft material, lbs/in\(^2\)
- \( I \) = Moment of inertia (\( \pi d^4/64 \)), in\(^4\)
- \( d \) = Shaft diameter, in

A typical shaft assembly is shown in Figure 20.2. Equation (20-8) (Ref. 8) can be used to determine the shaft deflection for this design. Similar equations will apply to the applicable shaft configurations.

\[
C_{DY} = \frac{F}{3E} \left[ \frac{X^3}{I_X} + \frac{L^3}{I_L} + \frac{M^3}{I_M} + \frac{N^3}{I_N} \right]
\]  \hspace{1cm} (20-8)

### 20.4.5 Stress Concentration Multiplying Factor

Most shaft failures originate at stress concentrations caused by discontinuities of shaft geometry and loads. The previous section contains those parameters affecting shaft loading. The total stress concentration factor is determined from the design of the fillet between shaft sections and the extent of shaft grooves if any along each shaft section. The total shaft stress concentration factor is then determined as follows:

\[
C_{SC} = C_{SC,R} + C_{SC,G}
\]  \hspace{1cm} (20-9)

where:

- \( C_{SC} \) = Shaft stress concentration factor
- \( C_{SC,R} \) = Stress concentration factor due to transition between shaft sections
- \( C_{SC,G} \) = Stress concentration factor due to shaft grooves
A stress concentration factor for each shoulder radii can be found using the following equation:

\[ C_{SC,R} = \left( \frac{0.3}{r/d} \right)^{0.2} \sqrt{\frac{D}{d}} \left(1 - \frac{r}{d}\right) \quad (20-10) \]

Where:

- \( r \) = Radius of fillet, in
- \( D \) = Initial shaft diameter, in (for example \( D_L \) in Figure 20.2)
- \( d \) = Transitioned shaft diameter, in (for example \( D_M \) in Figure 20.2)

Table 20-3 provides typical stress concentration factors for shaft grooves, \( C_{SC,G} \). If there are no grooves in the shaft, \( C_{SC,G} \) will be equal to 1.0.

Figure 20.2 Typical Shaft Assembly
Table 20-1. Average Values of Endurance Limits

(Reference 39)

<table>
<thead>
<tr>
<th>MATERIAL</th>
<th>ENDURANCE LIMIT $S_{ED}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel, $\sigma_{T,ult} \leq 200$ kpsi</td>
<td>$0.50 \sigma_{T,ult}$</td>
</tr>
<tr>
<td>Steel, $\sigma_{T,ult} &gt; 200$ kpsi</td>
<td>100 kpsi</td>
</tr>
<tr>
<td>Magnesium</td>
<td>$0.35 \sigma_{T,ult}$</td>
</tr>
<tr>
<td>Nonferrous Alloy</td>
<td>$0.35 \sigma_{T,ult}$</td>
</tr>
<tr>
<td>Aluminum Alloy (wrought)</td>
<td>$0.40 \sigma_{T,ult}$</td>
</tr>
<tr>
<td>Aluminum Alloy (cast)</td>
<td>$0.30 \sigma_{T,ult}$</td>
</tr>
</tbody>
</table>

Note: Alloys are heat treated, hot worked; specimens are smooth, subjected to long life rotating beam tests.

Table 20-2. Shaft Surface Finish Factor

<table>
<thead>
<tr>
<th>FINISH</th>
<th>$C_f$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Polished</td>
<td>1.0</td>
</tr>
<tr>
<td>Ground</td>
<td>0.89</td>
</tr>
<tr>
<td>Hot Rolled</td>
<td>$0.94 - 0.0046 T_S + 8.37 \times 10^{-6} (T_S)^2$</td>
</tr>
<tr>
<td>Machined or Cold Drawn</td>
<td>$1.07 - 0.0051 T_S + 2.21 \times 10^{-5} (T_S)^2 - 3.57 \times 10^{-8} (T_S)^3$</td>
</tr>
<tr>
<td>Forged</td>
<td>$0.75 - 4.06 \times 10^{-3} T_S + 7.58 \times 10^{-6} (T_S)^2$</td>
</tr>
</tbody>
</table>

Note: $T_S = \text{Tensile strength of material, kpsi}$
### Table 20-3  Stress Concentration Factor $S_{SC,G}$ for Shaft Groves

<table>
<thead>
<tr>
<th>h/D</th>
<th>0.1</th>
<th>0.5</th>
<th>1.0</th>
<th>2.0</th>
<th>4.0</th>
<th>6.0</th>
<th>8.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05</td>
<td>1.10</td>
<td>1.45</td>
<td>1.60</td>
<td>2.00</td>
<td>2.05</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>0.10</td>
<td>1.00</td>
<td>1.27</td>
<td>1.40</td>
<td>1.70</td>
<td>2.00</td>
<td>2.25</td>
<td>---</td>
</tr>
<tr>
<td>0.20</td>
<td>1.00</td>
<td>1.10</td>
<td>1.20</td>
<td>1.31</td>
<td>1.60</td>
<td>1.75</td>
<td>2.00</td>
</tr>
<tr>
<td>0.30</td>
<td>1.00</td>
<td>1.10</td>
<td>1.10</td>
<td>1.20</td>
<td>1.35</td>
<td>1.48</td>
<td>1.55</td>
</tr>
</tbody>
</table>

### Table 20-4  Shaft Material Strengths

<table>
<thead>
<tr>
<th>Shaft Material</th>
<th>Tensile Strength (Ultimate) $\sigma_u$ ksi</th>
<th>$\sigma_e / \sigma_u$</th>
<th>Endurance Strength $\sigma_e$ ksi</th>
<th>Modulus of Elasticity $E$ mpsi</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alloy steel</td>
<td>100 - 240</td>
<td>0.44</td>
<td>44 - 106</td>
<td>30</td>
</tr>
<tr>
<td>Stainless steel</td>
<td>80 - 230</td>
<td>0.30</td>
<td>24 - 69</td>
<td>29</td>
</tr>
<tr>
<td>High carbon steel</td>
<td>90-210</td>
<td>0.43</td>
<td>39 - 90</td>
<td>30</td>
</tr>
<tr>
<td>Cast steel, carbon</td>
<td>70 - 100</td>
<td>0.50</td>
<td>35 - 50</td>
<td>30</td>
</tr>
<tr>
<td>Low alloy cast steel</td>
<td>70 - 200</td>
<td>0.50</td>
<td>35 - 100</td>
<td>30</td>
</tr>
<tr>
<td>Cast aluminum</td>
<td>20 - 48</td>
<td>0.38</td>
<td>8 - 18</td>
<td>10.3</td>
</tr>
<tr>
<td>Wrought aluminum</td>
<td>22 - 83</td>
<td>0.35</td>
<td>8 - 29</td>
<td>10.0 – 10.6</td>
</tr>
</tbody>
</table>

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21.1 INTRODUCTION

There are a number of means to mechanically change speeds and transmit power in industrial applications. V-belts, synchronous belts, and chains are three methods of transmitting power between two shafts separated by a wide distance, and they are used over a wide range of speed ratios.

V-belt drives are a common means of transmitting horsepower and reducing speed. They are quiet and require very little maintenance. These drives transmit power through friction created by a wedging action of the belt in the sheave groove. V-belts work optimally with speed ratios of up to 6:1, but are available with drive ratios up to 10:1.

Synchronous belt drives offer positive engagement between mating teeth of a toothed belt and a toothed sprocket. No lubrication is required and they can operate at higher speeds than other positive engaging drives such as chain drives. Additionally, they offer less noise at slow speeds than chain drives.
Synchronous belts with a trapezoidal tooth profile are called timing belts. These belts are used in power transmission applications up to 50 horsepower. Another synchronous belt with a curvilinear tooth profile has a more rounded tooth and has come to be known as the high torque drive, or HTD. HTD sprockets can transmit up to 300 horsepower. With a modified curvilinear tooth profile, a higher tooth angle and slightly shallower tooth, synchronous belts can transmit up to 500 horsepower.

Applications requiring positioning, indexing, constant speeds, and no-slip are ideal for synchronous drives due to their positive tooth engagement. Additionally, wet environments where chains can rust and V-belts might slip, are other ideal applications. The no-slip feature of synchronous belts leads to greater efficiency which reduces power consumption. However, alignment, noise and high shock loads are more critical with synchronous belt drives. Alignment needs to be held to within 1/4° to prevent rapid wear. High horsepower and high-speed synchronous belt drives can experience noise problems due to rapid tooth interaction. Furthermore, synchronous belts are more susceptible to damage from high shock loads, and thus require a greater design service factor. However, properly applied and installed synchronous belt drives will last between 8,000 and 12,000 hrs.

High torque, slow speed applications are ideal for chain drives. Chain drives are compact, economical, and easy to install. They are capable of operating in high temperature environments and provide no slip with no special tensioning. Additionally, they offer the same efficiency advantages as synchronous drives due to their no-slip feature.

Chains are also used in serpentine drive applications and specialty conveyors because a variety of chain attachments and types of chain are available. The versatility, availability, and compactness of chain drives allow them to be used in all industries. While synchronous belts and V-belts are offered only in fixed belt lengths, a chain can be set to any length, providing for unrestricted center distance. Improper lubrication is the primary cause for premature chain failure.

21.2 BELT DRIVES

A belt drive is a low cost means of transmitting rotary motion from one shaft to another. Belt drives are smooth running, operationally quiet, and resistant to start-up or momentary overloads. Recently improved reinforcing materials have made belt drives more practical where formally only chain drives would have been reliably employed. Belt drives are used when large distances between shafts make gears impractical or when operational speed is too high for chain drives. A belt drive is an assembly of belts and pulleys including a means of attaching sheaves or pulleys to their respective shafts to transmit power. Pulleys and belts can be used to increase or reduce speed or torque, or to transfer power from one shaft to another.
As shown in Figure 21.1, belts are either flat or exist in a V-shaped cross section running on grooved pulleys.

- Flat belts: Usually a composite construction with cord reinforcement, suited for high speeds and relatively low power.

- Round belts: Used in agricultural machinery drives and light duty or appliance drives such as vacuum cleaners. Round belts are similar to V-belts and they run in V sheaves.

- V-belts require less tension than do flat belts, because they have more surface area contacting the pulley and therefore more friction. V-belts are comprised of a load carrying cord tensile member located at the pitch line, embedded in a relatively soft matrix which is encased in a wear resistant cover. The wedging action of a V-belt in a pulley groove results in a drive which is more compact than a flat belt drive, but short center V-belt drives are sensitive to shock. The narrow (wedge) V-belt provides more tensile member support than the classical V-belt. They handle an equivalent load, but with narrower face width and smaller diameter.

- Timing or synchronous belts are a specific class of belts that contain toothed members similar to spur gears. Timing belts are positive rather than friction driven since they rely on gear-like teeth on the pulley and belt. The teeth on the belt mesh with the teeth on the pulley providing for a positive, non-slip rotational drive assembly. Because of the toothed nature of the components, specific reduction ratios can be obtained along with increased power torque ratings over non-toothed pulleys and belts.

There are several different belt designs. One is a fabric-wrapped construction. The wrap provides a protective envelope that is designed to prevent damage and prolong life. In a non-wrapped belt, none of the cross section is allocated to a wrap; thus, the total section consists of working tensile material. This allows a higher power rating for belts with non-wrapped construction.

V-belts are applied in a variety of applications but work best in applications greater than 500 RPM. They offer a great benefit in that they will slip upon overload, thus protecting other, more expensive equipment from load surges.

The service life of a properly designed V-drive is approximately 20,000 to 25,000 hours. This life can be dramatically less if the drive is not installed or applied properly. V-belt drives are limited to a maximum misalignment of 1/2°, and standard belts are limited to operating temperatures from -40°F to 130°F. Published belt ratings are often based on a temperature of 85°F. For every 35°F above the baseline temperature, the
life will be cut in half. In addition to ambient temperatures, excessive slip caused by worn sheave grooves, tight bends, and poor ventilation can lead to increased belt operating temperatures.

![Flat Belt](image1.png) ![Round Belt](image2.png)

![V - Belt](image3.png) ![Timing Belt](image4.png)

**Figure 21.1 Belt Types**

Reliable operation of a belt drive depends on substantial tension to create a friction that allows the belts to grip the pulleys allowing them to turn and keeping them from slipping as they turn. Because V-belts operate based on friction and mechanical advantage, proper tensioning of V-belts is the single most important factor necessary for long, satisfactory operation. Too little tension will result in slippage, causing rapid belt and sheave wear. Too much tension can result in excessive stress not only on the belts, but also bearings and shafts. For maximum life, the belt must operate within prescribed tension ranges and in environments compatible with material life.

### 21.3 FAILURE MODES OF BELT DRIVES

The most common failure mode of belts is premature belt failure caused by an improperly tensioned belt. There is a tension at which the belt experiences optimum service life. Anything above or below this tension translates directly into a decrease in belt life.
Misalignment causes belts to experience sidewall stresses and uneven tensile member shock loads. As a result, the belts either suffer a tensile member failure due to sidewall cracking or rollover inside the sheaves. Misalignment can exist as non-parallel shafts, offset pulleys, and shafts or pulleys that have an angular skew at installation.

Belts running on worn pulleys are susceptible to slippage and excessive wear in localized sidewall regions causing a reduction in belt life. Belts run on worn sheaves can roll over in the sheaves, especially if there is any misalignment.

Table 21-1 presents a summary of failure modes for belt drives.

**Table 21-1. Typical Failure Modes of Belt Drives**

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Improper operating tension</td>
<td>Installation error</td>
<td>Belt failure</td>
</tr>
<tr>
<td>Pulley/sheave misalignment</td>
<td>Installation error</td>
<td>Sidewall cracking and belt failure</td>
</tr>
<tr>
<td>Worn pulley/sheave</td>
<td>Incorrect tension</td>
<td>Belt slippage and rapid wear rate</td>
</tr>
<tr>
<td>Temperature extreme</td>
<td>Belt slippage, operating environment</td>
<td>Belt hardening and reduced life</td>
</tr>
<tr>
<td>Chemical contamination</td>
<td>Operating environment</td>
<td>Belt wear and eventual failure</td>
</tr>
<tr>
<td>Foreign objects in the belt drive assembly</td>
<td>Operating environment</td>
<td>Belt wear and eventual failure</td>
</tr>
<tr>
<td>Belt slip</td>
<td>Insufficient tension</td>
<td>Excessive heat and wear generated with reduced belt life</td>
</tr>
<tr>
<td>Belt fatigue</td>
<td>Excessive tension</td>
<td>Broken belt</td>
</tr>
<tr>
<td>Worn belt and pulley/sheave,</td>
<td>Large starting and stopping forces greater than 10% above operating conditions</td>
<td>Premature belt failure</td>
</tr>
<tr>
<td>Normal wear rate</td>
<td>Normal repetitive stressing</td>
<td>Eventual belt failure</td>
</tr>
<tr>
<td>Rapid belt deterioration</td>
<td>Heat build-up due to inadequate ventilation</td>
<td>Variation in drive ratio and reduced belt life</td>
</tr>
<tr>
<td>Improper belt drive operation</td>
<td>Loose pulley/sheave on shaft</td>
<td>Sidewall cracking and belt failure</td>
</tr>
</tbody>
</table>
Belts are usually designed to operate within the temperature range of -20°F to 140°F. Elevated temperatures cause hardening, increased dynamic stiffness and reduced ultimate elongation of the cushion compound, reducing the belts flexibility. The loss in flexibility results in increased stress on the cushion stock. Eventually these stresses become too high for the compound to handle and a cushion crack appears. This crack serves to bring the stresses down to a manageable level. Continued exposure to elevated temperatures, however, will cause a continuous increase in stress level and continued cracking. Eventually, the cushion stock will not be able to support the cord line and the belt will fail to function.

21.4 RELIABILITY PREDICTION OF BELT DRIVES

21.4.1 Belt Misalignment

Parallel misalignment results from pulleys being mounted out of line from each other. Parallel misalignment is generally more of a concern with V-type belts than with synchronous belts because V-type belts run in grooves and are not as free to move sideways on the pulleys. Synchronous belts on the other hand are usually able to self align themselves as they run. However, if there is not enough clearance between the edge of the belt and the pulley flange, the belt can become pinched causing serious performance problems and early belt failure. Therefore, in order to maximize performance and reliability, synchronous drive assemblies should be aligned closely, a maximum allowable misalignment of ¼ degree. Parallel and angular misalignments are shown in Figure 21.2.

Parallel Misalignment          Angular Misalignment

Figure 21.2. Belt Misalignment

21.4.2 Belt Velocity

Belt velocity V is defined as:

\[
V = \frac{DN}{3.82}
\]  

(21-1)
where: \( D \) = pitch diameter, in
\( N \) = rotational speed, RPM

The ratio between the velocity in RPM of the faster shaft and that of the slower shaft is called the speed ratio:

\[
\text{Speed Ratio} = \frac{\text{RPM faster shaft}}{\text{RPM slower shaft}} = \frac{\text{Dia. larger pulley}}{\text{Dia. smaller pulley}}
\]  

Belt speed is an important consideration in estimating the design life of a belt. Higher speeds require increased belt tension to compensate for higher centrifugal force. In general, increasing speed increases belt life because the tight side belt tension decreases with increasing speed. See Equation (21-3). However, if the belt speed is too high, the increase in centrifugal tension can more than offset the decrease in tight side belt tension and belt life can be reduced. In timing belt drives (toothed belt) higher speeds generate dynamic forces resulting in increased tooth stresses and shorter belt life. Belt manufacturers normally provide a horsepower rating of the belt for specified belt speeds. The impact of belt speed on belt loading is explained in the next section.

21.4.3 Belt Loading

The belt on a belt drive assembly must be sufficiently tight to prevent slip causing a belt tension on both sides of the driven pulley or sheave. When the belt drive assembly is stationary with no power being transmitted, the tension on both sides of the pulley is equal. When the drive is rotating and transmitting power there is a tight side tension on the input side of the driven pulley and a slack side tension on the output side of the pulley. Figure 21.3 shows these tensions.
The ratio of the tight side to the slack side tension is called the tension ratio. The higher the tension ratio, the closer the belt is to slipping, i.e., the belt is too loose. The difference in these two tensions is a net tension. The net tension is calculated as follows:

\[
T_w = \frac{33,000 \text{ hp}}{V}
\]  

(21-3)

where:
- \(T_w\) = Tight side tension \((T_T) – \text{ slack side tension } (T_S)\), lbs
- \(hp\) = Power, hp = 33,000 ft-lbs/min
- \(V\) = Belt speed, ft/min

A belt experiences three types of tension as it rotates around a pulley:

- Working tension, \(T_w\) = tight side tension – slack side tension (See equation (21-3))

- Bending tension, \(T_B\) (As the belt bends around the pulley, one part of the pulley is in tension, the other in compression. The bending of the belt around a pulley introduces tension in the tensile member of the belt, the amount of tension dependent on the radius of the bend.

\[
T_B = \frac{C_B}{d}
\]  

(21-4)

where:
- \(T_B\) = Bending tension, lbs
- \(C_B\) = Constant depending on belt size and construction, in-lbs
- \(d\) = Pulley diameter, in

- Centrifugal tension occurs in a belt drive because of the centrifugal force caused by the belt which has weight rotating around the pulley. Centrifugal tension depends on the belt speed.

\[
T_C = MV^2
\]  

(21-5)

where:
- \(T_C\) = Centrifugal tension, lbs
- \(M\) = Constant related to belt weight, lbs/(ft/min)\(^2\)
\[ V = \text{Belt speed, ft/min} \]

There are certain times as the belt moves on the pulley when the three types of tension are cumulative:

\[ T_{\text{PEAK}} = T_T + T_B + T_C \]  \hspace{1cm} (21-6)

These tensions on the belt are shown in Figure 21.4

![Figure 21.4 Belt Drive Tensions](image)

Peak tension is directly related to belt life. The impact of tension (loading) on belt failure rate is shown in Figure 21.5

**21.4.4 Belt Operating Temperature**

Belts can operate successfully in temperatures from about -20 F to 140 F. Outside this temperature range, belt reliability depends on special belt materials. Belt life will normally decrease exponentially with an increase in temperature. Cold temperatures
have a more dramatic effect on belt life. In cold weather, the belt can fail when compounds within the belt structure reach their glass transition temperature causing a catastrophic belt failure. The impact of ambient temperature on the failure rate of the belt is shown in Figure 21.6

![Figure 21.6 Impact of belt loading on Failure Rate](image)

A = 100 ft/min  
B = 200 ft/min  
C = 500 ft/min  
D = 750 ft/min  
E = 1000 ft/min

**Figure 21.5 Impact of belt loading on Failure Rate**

### 21.5 FAILURE RATE MODEL FOR BELT DRIVE

The manufacturer of the belt will usually publish the horsepower rating of the particular belt being used or considered for a belt drive design. In some cases the
torque rating is supplied which can be equated to horsepower using the equations in Section 21.4.3. After the actual horsepower of the belt is computed using the equations in Section 21.4.3, it can be compared to the rated horsepower of the belt. The relationship of this horsepower ratio to estimated failure rate is shown in Figure 21.7.

![Figure 21.6 Impact of Temperature on Belt Failure Rate](image)

Most manufacturers of belts provide a horsepower rating of the belt at specific speeds. The actual horsepower can be estimated using the equations in Section 21.4 or it is usually easier to use the horsepower rating of the prime mover of the drive shaft. It must be remembered that the life of a flat or round belt is very much dependent on the tension of the belt and therefore dependent on maintenance procedures and the tension of the belt at the initial installation and any equipment repair.

The failure rate of the total belt drive can be estimated with the following equation:

\[
\lambda_{BD} = \lambda_{BD,B} \cdot C_{BL} \cdot C_t \cdot C_{PD} \cdot C_{BT} \cdot C_{BV} \cdot C_{SV} + \lambda_p
\]

(21-7)
where:

\[ \lambda_{BD} = \text{Failure rate of belt drive under specific operating conditions, failures/million hours} \]

\[ \lambda_{BD,B} = \text{Base failure rate of belt, 40 failures/million hours} \]

\[ C_{BL} = \text{Multiplying factor for belt loading (See Figure 21.7)} \]

\[ C_t = \text{Multiplying factor for belt operating temperature (See Figure 21.8)} \]

\[ C_{PD} = \text{Multiplying factor for pulley diameter (See Figure 21.9)} \]

\[ C_{BT} = \text{Multiplying factor for belt type (See Table 21-2)} \]

\[ C_{BV} = \text{Multiplying factor for belt drive operating service (See Table 21-3)} \]

\[ C_{SV} = \text{Multiplying factor for belt drive shock environment (See Table 21-4)} \]

\[ \lambda_p = \text{Failure rate for driver and driven pulleys, 0.8 failures/million hours for flat pulleys, 1.5 failures/million hours for grooved pulleys} \]
Belt Load Multiplying Factor, $C_{BL}$

$$C_{BL} = 0.3 + \left( \frac{hp_O}{hp_D} \right)^{4.2}$$

where:

- $C_{BL}$ = Belt load multiplying factor
- $hp_O$ = Operating load, ft-lbs/min
- $hp_D$ = Design or specified load, ft-lbs/min

**Figure 21.7** Belt Load Multiplying Factor, $C_{BL}$
Temperature Multiplying Factor, \( C_t \)

\[ C_t = \frac{I}{2^t} \]

where: \( t = 0 \) for Operating Temperature, \( T_O = -20 \text{ F} \) to \( 40 \text{ F} \)

and: \( t = \frac{40 - T_O}{55} \) for operating temperature, \( T_O > 40 \text{ F} \)

**Figure 21.8** Multiplying Factor for Belt Operating Temperature, \( C_t \)
Driven Pulley Diameter, \( PD \), in

\[
PDC \times PD = \left( \frac{4.5}{PD} \right)^2
\]

where: \( PD \) = Diameter of driven pulley, in

Figure 21.9  Multiplying Factor for Pulley Diameter, \( C_{PD} \)
Table 21-2. Belt Type Multiplying Factor, $C_{BT}$

<table>
<thead>
<tr>
<th>Belt Type</th>
<th>Belt Type Multiplying Factor, $C_{BT}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>SPZ</td>
<td>0.8</td>
</tr>
<tr>
<td>SPA</td>
<td>0.48</td>
</tr>
<tr>
<td>SPB</td>
<td>0.33</td>
</tr>
<tr>
<td>SPC</td>
<td>0.18</td>
</tr>
<tr>
<td>Y</td>
<td>9.09</td>
</tr>
<tr>
<td>Z</td>
<td>4.16</td>
</tr>
<tr>
<td>A</td>
<td>0.93</td>
</tr>
<tr>
<td>B</td>
<td>0.54</td>
</tr>
<tr>
<td>C</td>
<td>0.30</td>
</tr>
<tr>
<td>D</td>
<td>0.14</td>
</tr>
</tbody>
</table>

Table 21-3. Belt Operating Service Factors, $C_{BV}$

Reference 39

<table>
<thead>
<tr>
<th>Driver Pulley</th>
<th>Driven Pulley</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low or normal torque (small AC motors, shunt wound DC motors, small engines)</td>
<td>High or non-uniform torque (single phase AC motors, series wound DC motors, large internal combustion engines)</td>
</tr>
<tr>
<td>Intermittent service</td>
<td>Normal service</td>
</tr>
<tr>
<td>Typical loads</td>
<td>Blowers, small fans, centrifugal pumps, compressors</td>
</tr>
<tr>
<td>1.0</td>
<td>1.1</td>
</tr>
<tr>
<td>1.1</td>
<td>1.2</td>
</tr>
</tbody>
</table>
Table 21-4. Shock Environment Service Factors, $C_{SV}$

<table>
<thead>
<tr>
<th>Driven Machinery</th>
<th>Source of Power</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Low or Normal Torque</td>
<td>High or Non-uniform Torque</td>
<td></td>
</tr>
<tr>
<td>No shock</td>
<td>1.1</td>
<td>1.2</td>
<td></td>
</tr>
<tr>
<td>Light shock</td>
<td>1.2</td>
<td>1.3</td>
<td></td>
</tr>
<tr>
<td>Medium shock</td>
<td>1.3</td>
<td>1.5</td>
<td></td>
</tr>
<tr>
<td>Heavy shock</td>
<td>1.4</td>
<td>1.7</td>
<td></td>
</tr>
</tbody>
</table>

21.6 CHAIN DRIVES

A chain is a reliable machine component, which transmits power by means of tensile forces, and is used primarily for power transmission and conveyance systems. The function and uses of a chain drive are similar to those of a belt drive. The two most common types of chain include steel chain, especially the type called roller chain, which makes up the largest share of chains being produced, and plastic chain. Two basic applications of chain drives include power transmission and conveyer systems. Metal chain drives are normally used for applications below 3000 rpm where accuracy and reliability must be greater than that provided by rubber belts. Chain drives will usually maintain a constant speed under varying load conditions because the metal chain does not slip or stretch and will need only infrequent adjustment.

A chain drive is a combination of chains and sprockets and their means of shaft mounting. A chain can be defined as a series of links, usually comprised of metal, connected and fitted into one another to form what is in effect a flexible rack with a series of integral journal bearings. In its simplest form a chain drive consists of two sprockets of arbitrary size and a chain loop. Sprockets are wheels with external teeth shaped so that they can fit into the links of the drive or driven chain. The center distance from one hinge or joint to the next is known as the pitch of the chain and is the primary identifying dimension. Chain pitch is shown in Figure 21.10.

Power transmission chains can be categorized as roller chain, engineering steel chain, silent chain, detachable chain, and offset sidebar chain. Some of the advantages of chain drives over belt drives include:

- No slippage between chain and sprocket teeth
- Negligible stretch allowing chain drives to carry heavy loads
• Chain drives are less sensitive to dust and humidity than belts and are not adversely affected by sun, oil or grease
• Chain drives can operate at higher temperatures than can belt drives
• Longer operating life expectancy because flexure and friction contact occur between hardened bearing surfaces separated by an oil film
• Longer shelf life because of less susceptibility to storage conditions and non-deterioration of metal

Some of the potential reliability problems associated with chain drives include:
• Chain drives can elongate due to wearing of link and sprocket teeth contact surfaces
• Chain drives require frequent or, in some applications, continual lubrication
• Sprockets often need to be replaced when worn chains are replaced because of wear
• Chain drives are often limited to slower speeds than belt drives

Figure 21.10 Parts of a Chain Drive
21.7 FAILURE MODES OF CHAIN DRIVES

There are several ways a chain can fail:

- In a tensile failure, the chain is overloaded in tension until it is stretched so badly it will not function properly, or it is literally pulled apart.

- If the chain is loaded repeatedly in tension, at a load below the yield strength (the chain is not stretched) microscopic cracks will develop in the link plates. These cracks continue to grow until the chain breaks.

- In a wear failure, material is removed by sliding, or sliding combined with abrasion or corrosion, until the chain will not fit the sprockets. Either the mechanism will not function properly or the remaining material is so thin that the chain breaks.

- Galling is rapid wear caused by metal seizure between the chain pin and bushing. This rapid wear is caused by the combination of excessively high speeds and loads.

Parts of a chain drive are shown in Figure 21.10. Chain drives require tensioning on a regular basis. When the chain has “stretched” by approximately 3 percent the chain typically requires replacement. All the components of the chain (rollers, pins, bushings) have lost their case hardened surface properties at this point and failure is imminent.

Design life of a chain depends on adequate lubrication. Proper lubrication reduces the wear on all moving surfaces of the chain, and helps cushion the chain drive from shock loading. The type of lubrication method varies dependent upon the speed and operating environment. Manual lubrication involves periodic application of oil from an oiler or brush and is used for slow speed and intermittent operation. Oil bath lubrication is best for most applications and involves submerging the lower strand of chain in a chain casing oil sump. Recirculating oil systems or forced lubrication can be used to provide continuous lubrication to the chain. This method is more common in high speed, high horsepower applications.

The key factor causing a chain to jump the sprocket teeth is chain wear elongation. Because of wear elongation, the chain creeps up on the sprocket teeth until it starts jumping sprocket teeth and can no longer engage with the sprocket. When a chain is operating, the outer surface of the pin and inner surface of the bushing rub against one another creating constant wear. Proper lubrication reduces the amount of wear but does not eliminate it.

As the surface of the pin is reduced by wear, the rigidity of the pin decreases and eventually fatigue failure may result. How much wear is acceptable and at what point operation is no longer acceptable depends on the application. When wear elongation is
less than or equal to 1.5 percent for a transmission chain, or less than or equal to 2 percent for a conveyor chain, there is almost no risk of fatigue failure.

Chain sprockets can operate in a variety of environments. Dust and dirt can easily be separated from oil through proper circulation and filtering, allowing chain drives to be operated in rather harsh environments. Furthermore, chain drives are often used in high temperature environments, provided sufficient lubrication is available that won’t break down at elevated temperatures.

In summary, the failure modes of a chain drive will depend on its application and the allowable performance in terms of noise and chain elongation. Table 21-5 provides a summary of possible chain drive failure modes.

21.8 RELIABILITY PREDICTION OF CHAIN DRIVES

For a chain drive to transmit power, the driven sprocket must be rotated against a resisting torque. In contrast to belt drives, one strand of a chain drive must always be slack. Thus power is transmitted solely by the tension side.

Chain drives are less sensitive to dust and humidity than belts and are not adversely affected by sun, oil or grease. They can also operate at much higher temperatures than belts and they do not slip. Chain drives do require frequent lubrication (continual in some applications), they require very close alignment, and since the do not slip they provide no overload protection.

In Figure 21.11, the left sprocket is the driving side (power input) and the right sprocket is the driven side (power output). As counterclockwise rotation power is applied to the driving sprocket while adding resistance to the driven sprocket, then the chain is loaded in tension mainly along the D-A span, and tension is smaller in the other parts. Figure 21.12 shows this relation.
Table 21-5. Typical Failure Modes of Chain Drives

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turned/galled pins</td>
<td>Inadequate lubrication</td>
<td>Damaged chain</td>
</tr>
<tr>
<td>Enlarged holes</td>
<td>Overload</td>
<td>Damaged chain</td>
</tr>
<tr>
<td>Broken pins and/or link plates</td>
<td>Extreme overload</td>
<td>Damaged chain and sprockets</td>
</tr>
<tr>
<td>Worn link plate contours</td>
<td>Chain rubbing on chain guide</td>
<td>Damaged chain</td>
</tr>
<tr>
<td>Cracked link plates</td>
<td>Excessive loading; corrosive environment</td>
<td>- Chain fatigue failure</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Stress corrosion failure</td>
</tr>
<tr>
<td>Broken, cracked or deformed rollers</td>
<td>Excessive speed, chain riding too high on sprocket teeth</td>
<td>Damaged chain</td>
</tr>
<tr>
<td>Chain climbs sprocket teeth</td>
<td>Excessive chain slack; overload</td>
<td>Chain and sprocket wear failure</td>
</tr>
<tr>
<td>Foreign objects in the chain drive assembly</td>
<td>Operating environment</td>
<td>Chain wear and eventual failure</td>
</tr>
<tr>
<td>Excessive noise</td>
<td>Chain interference; loose casing; excessive chain slack; inadequate lubrication</td>
<td>Eventual chain damage</td>
</tr>
<tr>
<td>Bushing/roller fatigue</td>
<td>Chain loading</td>
<td>Chain wear and eventual failure</td>
</tr>
<tr>
<td>Chain elongation</td>
<td>Link-pin joint wear</td>
<td>Chain begins to skip teeth followed by complete failure</td>
</tr>
<tr>
<td>Tensile fatigue loading</td>
<td>Sideplate loading</td>
<td>Sudden catastrophic chain failure</td>
</tr>
</tbody>
</table>
As a chain is subject to increasing stress or load, it becomes longer. This relationship can be graphed (Figure 21.13). The vertical axis shows increasing stress or load, and the horizontal axis shows increasing strain or elongation. In this stress-strain graph, each point represents the following:

- O-A: elastic region
- A-C: plastic deformation
- B: maximum tension point
- C: actual breakage

The chain speed for a given sprocket diameter and sprocket angular velocity is given by:

\[ S = \frac{\pi n D}{12} = \frac{P N n}{12} \]  \hspace{1cm} (21-8)
where:

\[ S = \text{Chain speed, ft/min} \]
\[ n = \text{Sprocket speed, rpm} \]
\[ D = \text{Sprocket pitch diameter, in} \]
\[ P = \text{Chain pitch, in} \]
\[ N = \text{Number of teeth of sprocket} \]

For a chain drive with two sprockets, the speed ratio is:

\[
\frac{n_2}{n_1} = \frac{N_1}{N_2}
\]

(21-9)

And the chain tension is:

\[
T = \frac{33,000 \cdot HP}{S}
\]

(21-10)

where:

\[ T = \text{chain tension, lbs} \]
\[ S = \text{Chain speed, ft/min} \]
\[ HP = \text{Horsepower to be transmitted, hp} \]

21.9 FAILURE RATE MODEL FOR CHAIN DRIVE

A graph showing the area of reliable operation of a chain drive and the impact of chain speed on horsepower and resulting failure mechanisms is shown in Figure 21.14.

The location of Line O-A in Figure 21.14 depends on the chain’s allowable tension, including the fatigue strength of the connecting or offset links and the centrifugal force in high-speed rotation. Line B-C depends on the breakage limit of the bushing and roller.
In this kind of breakage of the bushing and roller, there is no fatigue limit as there is with the link plates. The location of line D-E depends on the bearing function of the pin and the bushing. The range defined within these three lines (O-A, B-C, and D-E) is the usable range. When the chain is used at low speeds, it is limited by line O-A, the fatigue limit.

Horsepower ratings are based upon the number of teeth and the rotating speed of the smaller sprocket, either drive or driven. The pin-bushing area, as it affects allowable working load, is the important factor for medium and higher speeds. For very low speeds, the limiting factor is the ultimate tensile strength of the chain. Figures 21.15 and 21.16 show the effect of RPM on horsepower rating of the chain.

In normal operation, the position at which the chain and the sprockets engage will fluctuate and the chain will vibrate along with this fluctuation. Even with the same chain, if the number of teeth in the sprockets is increased (change to larger diameter), vibration will be reduced. A decrease in the number of teeth in the sprockets will cause an increase in vibration. This is because there is a pitch length in chains, and they can only bend at the pitch point.

Considering the failure modes and stress considerations of chain drives, the following equation can be developed for estimating the failure rate of chain drives:

\[
\lambda_{CD} = \lambda_{CD,B} \cdot C_{CV} \cdot C_{CS} \cdot C_{CT} \cdot C_{CI} \cdot C_{ST} + \lambda_{CS}
\]  \hspace{1cm} (21-11)

where:

\[
\lambda_{CD} = \text{Failure rate of chain drive under specific operating conditions, failures/million hours}
\]

\[
\lambda_{CD,B} = \text{Base failure rate of chain, 15 failures/million hours}
\]

\[
C_{CV} = \text{Multiplying factor for operating service (See Table 21-7)}
\]

\[
C_{CS} = \text{Multiplying factor for chain speed (See Table 21-8)}
\]

\[
C_{CT} = \text{Multiplying factor for chain operating temperature (See Table 21-9)}
\]

\[
C_{CI} = \text{Multiplying factor for lubrication method (See Table 21-10)}
\]

\[
C_{ST} = \text{Multiplying factor for sprocket design (See Figure 21-17)}
\]
\[ \lambda_{CS} = \text{Failure rate for driver and driven sprockets, } 0.8 \text{ failures/million hours} \]

Figure 21.14  Typical Chain Drive Failure Modes
It is important to remember that when determining the horsepower ratings from manufacture’s catalogs that the total horsepower rating of a multiple strand is not a simple multiplication of a single chain horsepower rating times the number of strands. Use the applicable factor from Table 21-6 to determine the total horsepower rating of multiple strand chains.

Figure 21.15 Horsepower Ratings for Typical Single-Strand, Roller-Chain Drive (ANSI No. 60, 3/4 inch pitch)
Table 21-6. Multiple Strand Factor

<table>
<thead>
<tr>
<th>Number of Roller Chain Strands</th>
<th>Multiple Strand Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.0</td>
</tr>
<tr>
<td>2</td>
<td>1.7</td>
</tr>
<tr>
<td>3</td>
<td>2.5</td>
</tr>
<tr>
<td>4</td>
<td>3.3</td>
</tr>
<tr>
<td>5</td>
<td>3.9</td>
</tr>
<tr>
<td>6</td>
<td>4.6</td>
</tr>
</tbody>
</table>

Figure 21.16 Horsepower Ratings Per Inch Width for Typical Silent-Chain Drive (¼ inch pitch)
Table 21-7. Chain Drive Operating Service Factors, $C_{cv}$

<table>
<thead>
<tr>
<th>Driver Sprocket</th>
<th>Driven Sprocket</th>
<th>Type of Driven Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth running (electric motors, turbines)</td>
<td>Slight shock (electric motors with frequent starts, internal combustion engine with hydraulic coupling)</td>
<td>Moderate shocks (internal combustion engines with mechanical coupling)</td>
</tr>
<tr>
<td>1.0</td>
<td>1.1</td>
<td>1.3</td>
</tr>
<tr>
<td>Smooth (conveyers with small load fluctuations, centrifugal blowers)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.4</td>
<td>1.5</td>
<td>1.7</td>
</tr>
<tr>
<td>Some impact (conveyers with some load fluctuations, centrifugal compressors, marine engines)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.8</td>
<td>1.9</td>
<td>2.1</td>
</tr>
<tr>
<td>Large impact (machines with reverse or large impact loads)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 21-8  Chain Speed Multiplying Factor, $C_{cs}$

<table>
<thead>
<tr>
<th>Chain Speed</th>
<th>Speed Coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt; 50 ft/min</td>
<td>1.0</td>
</tr>
<tr>
<td>50 – 100 ft/min</td>
<td>1.2</td>
</tr>
<tr>
<td>&gt; 100 – 160 ft/min</td>
<td>1.4</td>
</tr>
</tbody>
</table>
### Table 21-9 Chain Temperature Multiplying Factor, $C_{CT}$

<table>
<thead>
<tr>
<th>Temperature</th>
<th>$C_{CT}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\leq 170 , ^{\circ}C$</td>
<td>1.0</td>
</tr>
<tr>
<td>170 – 200 $, ^{\circ}C$</td>
<td>1.5</td>
</tr>
<tr>
<td>&gt; 200 – 260 $, ^{\circ}C$</td>
<td>2.0</td>
</tr>
</tbody>
</table>

### Table 21-10 Chain Lubrication Multiplying Factor, $C_{CI}$

<table>
<thead>
<tr>
<th>Type of Lubrication</th>
<th>Multiplying Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manual Operation</td>
<td>1.5</td>
</tr>
<tr>
<td>Drip Lubrication</td>
<td>1.0</td>
</tr>
<tr>
<td>Bath Lubrication</td>
<td>0.8</td>
</tr>
<tr>
<td>Stream Lubrication</td>
<td>0.7</td>
</tr>
</tbody>
</table>
Figure 21.17 Multiplying Factor for Sprocket Design

\[ C_{ST} = \frac{19}{ST} \]

where:

\( ST = \) Number of teeth on smaller sprocket
21.10 REFERENCES


88. Reliability Analysis Center, “Nonelectronic Parts Reliability Data”, NPRD-95


91. Renold Transmission Chain Selection Procedure

106. Design and Analysis of Machine Elements, Douglas Wright, Department of Mechanical and Materials Engineering, The University of Western Australia


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22.1 INTRODUCTION

Fluid conductors are the means by which the entire fluid system is connected. It is therefore very important to assure the reliability of the various connections in the system. Many times in evaluating a fluid system for reliability, the major components such as pumps, valves and actuators are analyzed while overlooking the fluid-carrying link between the components. These fluid conductors are usually very reliable from a design standpoint but their reliability can be very sensitive to the operating environment.

Conductors of a fluid power system are basically of three types: pipe, tubing and hose. A pipe is a rigid conductor not intended to be bent or shaped into a configuration. Tubing is a semi-rigid fluid conductor which is usually bent into a desired shape for the particular application. Hose is a flexible fluid conductor which can be adapted to components that move during operation.

There are many factors to consider when evaluating the reliability of a fluid conductor system. For example, newer designs are combining tubing and hoses as hybrid assemblies. These assemblies provide the strength and heat dissipation
characteristics of metal tubing with the flexibility and vibration dampening characteristics of hose.

The failure modes presented in this Chapter should be considered for the particular application of the fluid conductor because the failure rate of a fluid conductor is probably more sensitive to the operating environment of the system in which it is installed than to the design parameters. Each application must be evaluated individually because of the many installation, usage and maintenance variables that affect the failure rate.

22.2 PIPE

Most failures of fluid conductor systems occur at or within the interconnection points such as fittings and flanges. Reliability of the system therefore depends on the proper selection of interconnecting components that are compatible with each other and the operating environment. Considerations include strength, ductility, hardness, and corrosion resistance. A pipe that can deflect more than 2% in diameter without cracking is considered a flexible pipe, one that cannot deflect to this degree is deemed rigid.

Piping components are designed for an internal pressure to compensate for the most severe condition of fluid pressure and temperature expected in normal operation. In addition to normal fluid operating pressures, potential back pressures, pressure surges, temperature fluctuations and performance variations of pumps, valves and other components must also be considered in the evaluation of fluid conductor systems. These conditions are met using the greatest required pipe thickness and the highest flange rating. The system must also be evaluated for the maximum external differential pressure conditions.

Older fluid conductor systems are comprised of threaded pipe and many hydraulic systems had NPT ports. This type of connection is the least reliable for high-pressure fluids as the thread itself provides a leak path. Because pipe threads are deformed when tightened, any subsequent movement, either loosening or tightening, increases the potential for leaks. Threaded connections have more recently been replaced by more reliable soft seal connections.

System temperatures need to be evaluated when estimating reliability. A single over-temperature event of sufficient magnitude can permanently damage all the seals in an entire high-pressure fluid system resulting in numerous leaks. Also, prolonged operation at above-normal temperatures can produce the same results.

The life of a pipe system depends not only on the material, but the installation and the surrounding environment. Typical pipe system materials include high density polyethylene (HDPE), cross linked polyethylene (PEX), acrylonitrile butadiene styrene (ABS), polyvinyl chloride (PVC), stainless steel, carbon steel, copper and ductile iron. One material does not exhibit a significant difference in lifespan from another. When properly designed and installed, pipe systems of any of these materials can be
sufficiently durable to exhibit extremely low failure rates. Plastic pipe can exhibit some long term failure modes due to chemical attack, deterioration from ultraviolet rays, change in dimension (creep) and environmental stress cracking. Failure data indicates no difference in the failure rate between pressure and gravity flow systems assuming proper pipe material selection and layout and proper tightness of pressure system connections to avoid leaks.

### 22.2.1 Failure Modes of Pipe Assembly

As mentioned previously, most failures of a pipe assembly occur at or within the interconnection points. The life of a pipe system depends not only on the material, but the installation and the surrounding environment. The following failure modes need to be considered when evaluating a pipe assembly for reliability:

- Burst failure caused by internal pressure
- Buckling caused by external pressure
- Bending failure
- Stress related failure from applied loads
- Excessive leakage at the interconnection points

A common failure mode in pipe systems is caused by a sudden reduction in liquid flow in a pipe. When a valve is abruptly closed, dynamic energy is converted to elastic energy creating a pressure wave called water hammer that can cause pipe failure.

Table 22-1 provides a summary of potential failure modes of a pipe assembly.

### 22.2.2 Reliability Prediction of Pipe Assembly

The failure modes presented above should be considered for the particular application of the fluid conductor. The failure rate of a fluid conductor is extremely sensitive to the operating environment of the system in which it is installed as compared to the design of the pipe. Each application must be evaluated individually because of the many installation, usage and maintenance variables that affect the failure rate.

The internal pressure in piping normally produces stresses in the pipe wall because the pressure forces are offset by pipe wall tension. The longitudinal stress from pressure is calculated by:

\[
S_L = \frac{P_D D}{4t}
\]

(22-1)

where:  
\( S_L \) = longitudinal stress, psi  
\( P_D \) = internal design pressure, psi
\[ D = \text{outside pipe diameter, in} \]
\[ t = \text{pipe wall thickness, in} \]

Table 22-1. Typical Failure Modes of Pipe Assemblies

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
</table>
| Damaged connector  | - corrosion
                  - improper torque on fitting
                  - gasket failure                                          | Gradual increase in system leakage                  |
| Burst failure      | - rapidly applied load
                  - pressure transients                                       | Catastrophic pipe assembly failure                  |
| Buckling failure   | - insufficient piping supports                      | Immediate leakage above system requirements         |
| Bending failure    | - bend radius less than allowable                   | Immediate leakage above system requirements         |
| Crack in rigid pipe| - external stress                                    | System leakage                                      |
| Leakage            | - chemical incompatibility with fluid
                  - chemical attack/improper thread sealant
                  - ultraviolet deterioration                          | Gradual increase in system leakage                  |
| Fatigue failure    | - water hammer from upstream component              | System fluid leakage                                 |

The burst pressure of the pipe is determined as follows:

\[ P = \frac{2 t S}{D} \]  

(22-2)

where:
\[ P = \text{burst pressure, psi} \]
\[ t = \text{pipe wall thickness, in} \]
\[ S = \text{tensile strength of pipe material, psi} \]
\( D = \) outside pipe diameter, in

And the working pressure is equal to:

\[ WP = \frac{P}{sf} \]  \hspace{1cm} (22-3)

where:

- \( WP \) = working pressure
- \( P \) = burst pressure
- \( sf \) = safety factor (normally equal to approximately 3.0)

Since the failure rate of a piping assembly usually depends primarily on the connection joints, the basic failure rate of a piping assembly can be estimated at 0.47 failures/million operating hours per connection and the failure rate of the pipe assembly can be estimated with the following equation:

\[ \lambda_P = \lambda_{P,B} \cdot C_E \]  \hspace{1cm} (22-4)

where:

- \( \lambda_P \) = failure rate of pipe assembly, failures per million hours
- \( \lambda_{P,B} \) = base failure rate of pipe assembly, 0.47 failures/million hours
- \( C_E \) = environmental factor, see Table 22-2

<table>
<thead>
<tr>
<th>Operating Environment</th>
<th>Multiplying Factor, ( C_E )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal duty, non-flex pipe, ambient conditions, no vibration or shock</td>
<td>1.0</td>
</tr>
<tr>
<td>Heavy duty, flexible pipe under random pulsations</td>
<td>1.2</td>
</tr>
<tr>
<td>Severe duty, vibration and shock environment</td>
<td>1.4</td>
</tr>
</tbody>
</table>

See Chapter 3 of this Handbook to determine the failure rate of any seals being used in the pipe connectors.
22.3 TUBING

Components on many types of hydraulic equipment are connected by rigid tubing. Because it is rigid, tubing can transmit vibration from one component to another throughout the equipment. Therefore, many hydraulic designs utilize bent tubes and hoses providing the weight and bend advantages of bent tube with the flexibility and vibration dampening characteristics of hose.

Although tubing is a better heat dissipater than hose, in some applications the use of hose can actually result in less heat buildup because of improved laminar flow through the more gradual bends created between hose connections.

All metal-to-metal connections, such as compression and flared type, are sensitive to excessive torque. Thus, the failure rate of line connections is dependent on assembly methods and will significantly affect the infant failure region of the failure rate bathtub curve.

When the conductor assembly is used to provide actuator movement, it will be subject to vibration changing the torque on plumbing connections and causing metal fatigue. Reliability is thus affected by the routing and supporting of the fluid conductor assembly. Tube bending and fabrication require proper training and experience to acquire reliable connections. Misalignment causes strain on the tubing, which can lead to leakage or line failure once in service. Improper deburring and flaring can eventually lead to stress cracks after the lines have been installed.

22.3.1 Failure Modes of Tubing Assembly

Plastic tubing will exhibit different failure modes than that for metal tubing. One common failure mode is caused by the change of properties of the plastic over time and/or temperature. The strength and stiffness of many common plastics change dramatically over a relatively small change in temperature. Some plastics such as PVC can become brittle and will shatter when exposed to lower temperatures. The stiffness of many plastics changes when exposed to heat. Likewise, the amount of load or stress that some plastics can withstand will change over time (creep). Plastics can also fail when combining stress and chemicals (environmental stress cracking).

Leakage is the most common failure mode in fluid systems. Connections that incorporate an elastomeric seal such as BSPP and SAE-4 bolt flange offer the highest seal reliability. NPT is the least reliable type of connector for high pressure hydraulic systems because the thread itself provides a leak path. The threads are deformed when tightened and as a result, any subsequent loosening or tightening of the connection increases the potential for leaks.

A common cause of leakage from 37 degree flare joints is incorrect torque. Insufficient torque results in inadequate seat contact, while excessive torque can result in damage to the tube and connector through cold working.
Table 22-3 provides a summary of potential failure modes for tubing assemblies.

**Table 22-3. Typical Failure Modes of Tubing Conductor Assemblies**

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress cracks</td>
<td>- excessive strain from misalignment</td>
<td>System leakage</td>
</tr>
<tr>
<td></td>
<td>- improper deburring and flaring</td>
<td></td>
</tr>
<tr>
<td>Metal fatigue</td>
<td>- vibration</td>
<td>System leakage</td>
</tr>
<tr>
<td>Compressed air line leak</td>
<td>- line misalignment</td>
<td>Reduced compressor life</td>
</tr>
<tr>
<td></td>
<td>- vibration</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- excessive torque loading on compression fittings</td>
<td></td>
</tr>
<tr>
<td>Damaged connector</td>
<td>- improper torque on fitting</td>
<td>System leakage</td>
</tr>
<tr>
<td></td>
<td>- external impact</td>
<td></td>
</tr>
<tr>
<td>Tubing burst</td>
<td>- pressure transients</td>
<td>Immediate tubing assembly failure</td>
</tr>
<tr>
<td></td>
<td>- suddenly applied load</td>
<td></td>
</tr>
<tr>
<td>Fluid leakage</td>
<td>- chemical incompatibility with fluid</td>
<td>Gradual increase in system leakage</td>
</tr>
<tr>
<td></td>
<td>- improper thread sealant</td>
<td></td>
</tr>
<tr>
<td>Tube buckling failure</td>
<td>- tubing support failure</td>
<td>Immediate leakage above system</td>
</tr>
<tr>
<td></td>
<td></td>
<td>requirements</td>
</tr>
</tbody>
</table>

**22.3.2 Reliability Prediction of Tubing Assembly**

The equations for determining the burst and working pressures of a tubing assembly are the same as those derived in Section 22.2.2.

Since the failure rate of a tubing assembly usually depends primarily on the connection joints, the basic failure rate of a tubing assembly can be estimated at 1.33 failures/million operating hours per connection and the failure rate of the tubing assembly can be estimated with the following equation:

\[
\dot{\lambda}_T = \dot{\lambda}_{T,B} \cdot C_F
\]  

(22-5)
where: \( \lambda_T \) = Failure rate of tube assembly, failures per million hours
\( \lambda_{T,B} \) = Base failure rate of tube assembly, 1.33 failures/million hours
\( C_E \) = Environmental factor, see Table 22-4

See Chapter 3 of this Handbook to determine the failure rate of any seals being used in the tubing connections.

**Table 22-4. Tubing Assembly Environmental Factor, \( C_E \)**

<table>
<thead>
<tr>
<th>Operating Environment</th>
<th>Multiplying Factor, ( C_E )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal duty, ambient conditions, no vibration or shock</td>
<td>1.0</td>
</tr>
<tr>
<td>Heavy duty, random fluid pulsations</td>
<td>1.2</td>
</tr>
<tr>
<td>Severe duty, vibration and shock environment</td>
<td>1.4</td>
</tr>
</tbody>
</table>

### 22.4 HOSE

A hose assembly is comprised of three basic elements including the inner tube to convey the fluid, reinforcement to withstand the fluid pressure, and outer cover to protect the hose from abrasion and other environmental conditions. The failure rate of a hose assembly depends on its size, temperature, application, media, and pressure. However, even knowing all the operating parameters of a system, it is still difficult to predict the expected service life of a hose. Sometimes the best method of obtaining a predicted service life is reliance on an adequate maintenance program that includes fluid cleanliness and visual inspections for abrasion, heat damage, etc.

- **Pressure** – Pressure spikes that exceed the maximum rated working pressure can cause damage and early failure in a hose assembly, the pressure impulses causing the hose to expand and contract.

- **Temperature** – A single over-temperature event of sufficient magnitude or prolonged operation at above –normal temperature can permanently damage the seals in the hose assembly resulting in leakage. At higher temperatures plasticizers tend to leach out of elastomers faster the rate dependent on the temperature and duration. Excessively low temperatures can cause the hose to harden, take on a permanent set, and initiate hose cracking. Proximity to other components producing
high temperatures can create shortened hose life.

- **Application** – An individual hose assembly is designed for a specific range of pressures, flows and temperatures and deviating from these design parameters in actual usage will obviously shorten the design life. Excessive flow velocity, for example, will damage the inner tube, especially at hose bends, and cause premature failure. Secondary failures can also be created in other system components caused by the increased temperature. Improper hose length can also cause failures from hose stretching in a hose assembly that is too short or excessive bending in a hose that is too long for the application. In general, mobile applications face harsher conditions than non-mobile permanent installations due to abrasion.

- **Media** – A fluid incompatible with a hose will shorten the design life of the hose. Consideration must be given to the chemical composition of the environment surrounding the fluid conductor assembly as well as the media being conducted including abrasive particles and corrosive properties when estimating the failure rate.

### 22.4.1 Failure Modes of Hose Assembly

Hose assemblies have a finite life, the main factor contributing to failure being service conditions. The following factors will affect the service life of metal hose:

- Pitting corrosion
- High fluid velocity combined with chemical abrasives
- Stress corrosion
- Vibration
- Torsion fatigue
- Tight radius bending and constant motion

Operating pressure – The maximum operating pressure within the hose should not exceed the recommended working pressure as specified by the manufacturer. Burst pressure should not be used as the operating pressure. Exposing the hose to pressures higher than the working pressure or exposing the hose to a surge pressure above the working pressure of the hose will shorten hose life.

Operating temperatures – High heat conditions may have an adverse affect on hose life due to the degradation of the rubber and the affect on fitting retention. Continuous use of the hose at or above the maximum allowable operating temperature will cause deterioration of the tube, cover and reinforcement thus reducing hose life. Fluid and ambient temperatures, both static and transient must not exceed the limitations of the hose.
External forces – Flexing the hose to less than its minimum bend radius, twisting or kinking the hose will reduce hose life. Evaluating a hose for reliability must therefore include an examination of fittings and adapters designed to prevent the impact of external forces. Excessive abrasion can damage the hose cover, accelerating hose failure.

Fluid and environmental compatibility – The expected life of a hose assembly depends on the chemical resistance of the tube, cover, O-ring fitting and other hose components and compatibility with the fluid being used and the environment. Ultraviolet light, ozone, salt water and various chemicals can shorten hose life.

Hose configuration – The size of the hose assembly components must be adequate to keep pressure losses to a minimum and avoid damage to the hose due to heat generation or excessive turbulence.

Table 22-5 provides a summary of failure modes for hose assemblies.

**Table 22-5. Typical Failure Modes of Hose Assemblies**

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Metal hose fatigue</td>
<td>- high flow velocity</td>
<td>- continual increase in size of crack until complete fracture</td>
</tr>
<tr>
<td>failure</td>
<td>- flexing of corrugations</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Excessive shear stress</td>
<td>- twisting the hose during installation</td>
<td>- circumferential cracks</td>
</tr>
<tr>
<td>in metal hose</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Irregular cracks in</td>
<td>- vibration</td>
<td>- eventual failure - leakage</td>
</tr>
<tr>
<td>metal hose</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inner tube deterioration</td>
<td>- elevated temperature</td>
<td>- eventual hose failure - contaminants entering system</td>
</tr>
<tr>
<td>Excessive fluid</td>
<td>- high fluid temperature caused by excessive flow velocity</td>
<td>- premature hose failure</td>
</tr>
<tr>
<td>temperature</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fractured hose</td>
<td>- excessively small bend radius</td>
<td>- reduced ability to withstand internal pressure</td>
</tr>
<tr>
<td></td>
<td>- hose bend immediately behind the coupling</td>
<td></td>
</tr>
<tr>
<td>Hose leakage</td>
<td>- continuous exposure to high temperature</td>
<td>- loss of hose flexibility</td>
</tr>
<tr>
<td></td>
<td>- chemical deterioration</td>
<td></td>
</tr>
<tr>
<td>Inner tube failure</td>
<td>- inadequate compatibility with fluid</td>
<td>- eventual hose failure</td>
</tr>
</tbody>
</table>
22.4.2 Reliability Prediction of Hose Assembly

The failure rate of hoses is very hard to estimate because of the varied operating conditions. Studies by fluid power part manufacturers indicate that the three most common causes of hydraulic hose failure are abuse, misapplication and improper assembly. Hydraulic hose manufacturers for example estimate that 80% of hose failures are attributable to external physical damage through pulling, kinking, crushing, or abrasion of the hose. Abrasion caused by hoses rubbing against each other or surrounding surfaces is a very common type of hose damage.

Hoses used on diesel engines historically have higher failure rates than those used on gasoline engines. Chemical deterioration causes hose failure and temperature accelerates chemical reaction. Diesel engines run hotter than gasoline engines.

22.4.2.1 Metallic Hoses

There are several factors associated with flexing that affect the service life of corrugated metal hose. Service life may be affected by factors external to the metal hose assembly such as the chemical composition of the environment surrounding the hose assembly as well as the media being transferred.

Turbulent flow of abrasive chemical media over the alloy surface of the hose may cause accelerated corrosion or erosion-corrosion. Liquids or gases that have suspended solid particles will wear or remove the oxide protective film of the hose and leave the alloy exposed and more susceptible to corrosion.

Fatigue is another failure mode to be considered. The flexing of the corrugations during hose operation can cause a failure of a progressive nature. Stress generated by flexure, pulsation, torsion, vibration and flow induced vibration are some other causes for fatigue failure. Applications where the flow of a liquid or gas is above manufacturer specifications and a liner is not incorporated into the hose assembly can result in premature fatigue failure. The high flow velocity causes the corrugations to vibrate at a high frequency and, if the vibration is near the natural frequency of the hose, failure can occur very quickly.

22.4.2.2 Non-metallic Hoses

The most common causes of a non-metallic hose assembly failure include:
- flexing the hose to less than the specified minimum bend radius
- twisting, pulling, kinking, crushing, or abraiding the hose
- operating the fluid system above maximum or below minimum temperature
- exposing the hose to surges in pressure above the maximum operating pressure
- intermixing hose, fittings and other parts that are not compatible

All hoses are rated with a maximum working temperature. Exposure to continuous high temperatures can lead to the hose losing its flexibility. Exceeding these temperatures can reduce hose life by as much as 80%. When hoses are exposed to high external and internal temperatures simultaneously, there will be a significant reduction in hose service life.

Bending a hydraulic hose in more than one plane results in the twisting of its wire reinforcement. A twist of five degrees can reduce the service life of a high-pressure hydraulic hose by as much as 70% and a seven degree twist can result in a 90% reduction in service life. Multi-plane bending is usually the result of inadequate clamping where the hose is subjected to mechanical motion.

Operating conditions have a direct effect on the service life of a hose assembly. Temperature extremes can accelerate the aging of the hose’s rubber tube and cover. Frequent and extreme pressure fluctuations will accelerate hose material fatigue.

\[
\lambda_H = \lambda_{H,B} \cdot C_E \cdot C_T
\]

where:
- \( \lambda_H \) = Failure rate of hose assembly, failures per million hours
- \( \lambda_{H,B} \) = Base failure rate of hose assembly, 1.85 failures/million hours
- \( C_E \) = Environmental factor, see Table 22-6
- \( C_T \) = Temperature factor, see Figure 22.1

<table>
<thead>
<tr>
<th>Environment</th>
<th>Environmental Factor, ( C_E )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal duty, ambient conditions, no vibration or shock</td>
<td>1.0</td>
</tr>
<tr>
<td>Heavy duty, random fluid pulsations</td>
<td>1.2</td>
</tr>
<tr>
<td>Severe duty, vibration and shock environment</td>
<td>1.5</td>
</tr>
</tbody>
</table>
\[ C_T = \frac{1}{2^t} \]

Where:

\[ t = \frac{(T_R - T_O)}{18} \text{ for } (T_R - T_O) \leq 40^\circ F \]

and:

\[ C_T = 0.21 \text{ for } (T_R - T_O) > 40^\circ F \]

\[ T_R = \text{Rated Temperature of Hose, } ^\circ F \]
\[ T_O = \text{Operating Temperature of Hose, } ^\circ F \]

**Figure 22.1 Temperature Multiplying Factor, \( C_T \)**
22.5 REFERENCES

88. Reliability Analysis Center, “Nonelectronic Parts Reliability Data”, NPRD-95

92. PST =>Solutions! , Volume 3, October 1996
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<th>Title</th>
<th>Page</th>
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<td>2</td>
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<td>3</td>
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<td>23.2.2</td>
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<td>3</td>
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<tr>
<td>23.3</td>
<td>BOLT</td>
<td>3</td>
</tr>
<tr>
<td>23.3.1</td>
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<td>3</td>
</tr>
<tr>
<td>23.3.2</td>
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<td>4</td>
</tr>
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<td>4</td>
</tr>
<tr>
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23.1 INTRODUCTION

The failure rate of a total mechanical or hydraulic system depends upon the active components of the system such as pumps, valves, gears and springs. The system will normally contain passive or non-moving parts as well, such as pins, couplings, nuts and bolts. The reliability of these passive or non-moving parts can be very hard to predict partly because their individual failure rates are small and little failure rate data have been compiled. A structural analysis of each part is often not warranted; yet, their cumulative failure rate data must be considered in the total reliability analysis. This chapter provides some guidance on estimating miscellaneous part failure rates. It also provides the appropriate section in this Handbook for specific mechanical parts where additional guidance on evaluating the part for reliability can be obtained.

The design evaluation procedures in other chapters of this Handbook require a detailed understanding of the design and operating environment of the component being considered. In contrast, the items in this Chapter may require an estimate of failure rate based on personal experience with the component and its operating environment. It is important to evaluate the source of any published failure rates before applying them in a reliability analysis with possible modification of the failure rate value for the component based on the intended application.

23.2 AXLE

An axle is a non-rotating member which carries no torque and is used to support rotating wheels and pulleys. The failure rate of an axle is dependent on material properties, weight of the wheel or pulley and its operating environment including shock and vibration. It is difficult to estimate a failure rate for an axle without performing a detailed stress analysis, since an axle is normally designed for infinite life and the failure rate almost totally dependent on operating conditions.

In the process of evaluating an axle for reliability, it is important to consider the total axle system. A bent axle, for example, will cause subsequent failure of bearings, bushings, wheels and other components that are mounted on the axle.
23.2.1 Axle Failure Modes

The following table includes some typical failure modes to consider when evaluating an axle for reliability.

Table 23.1. Typical Failure Modes for an Axle

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fatigue failure at stress raiser</td>
<td>Shock, vibration</td>
<td>Crack growth, eventual breakage</td>
</tr>
<tr>
<td>Bent axle</td>
<td>Bending stresses</td>
<td>Axle mounted component failure</td>
</tr>
<tr>
<td>Unstable rotation</td>
<td>Vibration</td>
<td>Axle fatigue and breakage</td>
</tr>
<tr>
<td>Uneven rotation</td>
<td>Loose mounting assembly</td>
<td>Axle mounted component failure</td>
</tr>
</tbody>
</table>

23.2.2 Axle Failure Rate

A stress–strength analysis is required to accurately estimate the failure rate of an axle for the intended operating environment. For estimating purposes, a typical failure rate for a shaft will be 0.01 failures/million hours. Consideration of this failure rate must include the evaluation of the parts mounted on the shaft that may be sensitive to axle failure mechanisms. The procedures in Chapter 20 can be used to determine the failure rate for small axles.

23.3 BOLT

Structurally, a bolt acts as a pin to keep two or more parts from slipping relative to each other, or to clamp two or more pieces together. In either case the bolt must be tightened properly if it is to perform its intended function. Thus, the failure rate of a bolted joint is very much dependent on the assembly or rework process.

23.3.1 Bolt Failure Modes

The possible failure modes of a bolted joint depend on the application. In fluid systems, for example, a failed bolted joint may result in leakage; the definition of leakage, in turn, dependent on the type of fluid being contained and the criticality of the leak. For machinery that is subject to shock and vibration, self-loosening, corrosion and
fatigue are failure modes of concern. Table 23.2 lists some of the potential failure modes of a bolted joint.

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Loose bolt</td>
<td>Insufficient clamp force or preload, excessive shock and/or vibration</td>
<td>Joint movement, fluid leakage, eventual fatigue failure</td>
</tr>
<tr>
<td>Plastic deformation</td>
<td>Applied force overload, excessive bearing pressure</td>
<td>Tensile failure</td>
</tr>
<tr>
<td>Sheared bolt</td>
<td>Repeated loading, vibration</td>
<td>Fatigue failure</td>
</tr>
<tr>
<td>Stripped threads</td>
<td>Excessive preload</td>
<td>Eventual bolt failure</td>
</tr>
</tbody>
</table>

### 23.3.2 Bolt Failure Rate

Approximately 75 different factors affect the tension created in a single bolt when a torque is applied. Still more variables must be considered when a group of bolts in an assembly is tightened. Fortunately, the design process of bolted joints is well established for normal machine designs and the failure rate quite low. A typical failure rate of a bolt based on published failure rate data is 0.12 failures per million hours. This failure rate can usually be assumed for the bolted system including the washer(s), nut and threadlocking devices. Table 23.3 provides recommended multiplying factors to adjust the base failure to consider the operating environment.

Just as failure modes are dependent on the application of the equipment, so must the failure rate be adjusted per the individual machine design and the number of bolts being used. For a complex machine design where weight or potential deformation becomes a factor, a finite element or stress analysis is usually performed. Detailed procedures for evaluating a bolt design for reliability are contained in Chapter 16.

### 23.4 BUSHING

Mechanical bushings consist of a cylinder with a hole, the bushing fitting into a bore and providing strength, wear resistance, lubrication or precision size and shape for components that are assembled with pins, shafts or bolts. Many bushings are press fit into structures and can be replaced after wear without requiring replacement of the entire structure.
Table 23.3  Service Factors for a Bolt

<table>
<thead>
<tr>
<th>LOADING</th>
<th>NORMAL VIBRATION</th>
<th>CONTINUOUSLY HIGH VIBRATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>Light loading (office type equipment)</td>
<td>1.00</td>
<td>1.50</td>
</tr>
<tr>
<td>Medium loading (compressors, large motors)</td>
<td>1.25</td>
<td>1.88</td>
</tr>
<tr>
<td>Heavy loading (Production equipment, trucks, aircraft, shipboard)</td>
<td>1.67</td>
<td>2.50</td>
</tr>
<tr>
<td>Very heavy loading (construction equipment, tanks)</td>
<td>2.50</td>
<td>3.75</td>
</tr>
</tbody>
</table>

Sleeve bushings or journal bearings are used to protect steel shafts from wear due to a combination of pressure and rotating motion. Slide bushings are used where accurate linear movement of a large object is required. A ball bushing permits rotation, sliding motion or both. A bushing blank is used as an insert in a hole to reduce the inside diameter of the hole and protect the surrounding body structure from damage resulting from vibrational and loading stress.

Chapter 7 contains the procedures for evaluating a bearing for reliability including the failure modes to be considered. These same procedures can be used for determining the failure rate of a bushing.

23.4.1 Bushing Failure Modes

Failure modes of a bushing depend on the application of the bushing. Mechanical bushings may be a fixed or removable cylinder metal lining used to constrain, guide, or reduce friction, or they may be a threaded adapter to permit joining of pipes with different diameters. In some applications leakage can be a potential failure mode. Some leakage can be expected since by design there is a gap between the bushing and shaft. Bushings are also used in chain drives. Chapter 21 contains some failure mode information on these types of bushings. Table 23.4 provides some typical failure modes for bushings.

23.4.2 Bushing Failure Rate

The failure rate of a bushing depends on many design factors including the yield strength of the material, hardness, elastic modulus, thermal conductivity, and galling resistance. Application variables include load, shaft speed, surface contaminants, lubrication and clearance. In highly loaded, slowly rotating systems, the maximum
bearing pressure is considered a key design criterion. Maximum pressure is the highest compressive stress a bushing can sustain without failure due to cracking or spalling. Normally, this maximum pressure should be no greater than twice the yield strength of the bushing material. For less highly loaded but more rapidly rotating systems, the pressure-velocity (PV) limit is a factor determining failure rate. In this case pressure is the total weight supported by the shaft converted to a compressive load, and velocity is the rotational speed converted to a velocity.

Table 23.4. Typical Failure Modes for Bushing

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Loose on shaft</td>
<td>Vibration</td>
<td>Loose coupling</td>
</tr>
<tr>
<td>Corrosion</td>
<td>Dissimilar materials/fluid</td>
<td>Shaft failure</td>
</tr>
<tr>
<td>Structural failure</td>
<td>Misalignment, excessive load</td>
<td>Bushing failure</td>
</tr>
<tr>
<td>Damaged set screw</td>
<td>Over tightening</td>
<td>Bushing failure</td>
</tr>
<tr>
<td>Loss of radial or axial movement</td>
<td>Loss of lubrication</td>
<td>Damaged grease seal</td>
</tr>
</tbody>
</table>

Typical published failure data for a bushing would indicate a failure rate of 0.72 failures/million hours. Obviously, this rate needs to be adjusted depending on personal experience with the application. In some cases the bushing is an integral part of the shaft. Chapter 20 provides guidance on determining a failure rate for this application. Chapter 21 contains information on the use of bushings in a chain linkage.

23.5 CAM MECHANISM

A cam is a projecting part of a rotating wheel or shaft that strikes a lever at one or more points on its circular path. The cam can be a simple tooth to deliver pulses of power or an eccentric disc or other shape to produce a smooth oscillating motion in the follower. A cam follower is the lever or output link making contact with the cam. The follower can be in the form of a roller, flat face or spherical.

Cam-followers are fitted between the camshaft lobe and the driven mechanism such as the end of a valve stem or push rod which it operates. Cam-followers reduce wear on the camshaft and ensure that the driven mechanism operates smoothly. Follower size is determined by the total useful force requirement.

After the cam-follower displacement curve is established, the unit is sized for long service life with minimum weight and size. Cam size with respect to pressure angle and
radius of curvature is an important design consideration. The pressure angle at any point on the profile of a cam is the angle between the direction where the follower wants to go at that point and where the cam wants to push it. It is the angle between the tangent to the path of follower motion and the line perpendicular to the tangent of the cam profile at the point of cam-roller contact. If the pressure angle exceeds 30 degrees for a radial cam for example, the force exerted on the cam is too great reducing life and accuracy.

23.5.1 Cam Mechanism Failure Modes

The primary failure mode of a cam–follower is wear. A cam does not necessarily operate at a constant drive speed and high speed cam-follower mechanisms often exhibit substantial drive speed fluctuations because of changes in the angular velocity of the driving mechanism due to load variations, and backlash between the driving mechanism and the cam.

Chapter 18 of this Handbook contains some additional guidance for evaluating the design of slider-crank mechanisms and the failure modes to be considered. The following table includes some typical failure modes to consider when evaluating a cam mechanism for reliability.

Table 23.5. Typical Failure Modes for a Cam Mechanism

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cam follower error</td>
<td>Excessive pressure angle, high torque during follower acceleration</td>
<td>Cam mechanical error</td>
</tr>
<tr>
<td>Drive speed fluctuation</td>
<td>Load variation and/or backlash</td>
<td>Rapid wear and mechanical error</td>
</tr>
<tr>
<td>Loose shaft/cam contact</td>
<td>Loose shaft pin</td>
<td>Loss of mechanical motion</td>
</tr>
<tr>
<td>Worn cam- follower contact</td>
<td>Excessive load, binding</td>
<td>Cam mechanical error and loss of follower contact</td>
</tr>
<tr>
<td>Cam/follower material fracture</td>
<td>Excessive load</td>
<td>Loss of follower contact</td>
</tr>
</tbody>
</table>
23.5.2 **Cam Mechanism Failure Rate**

The failure rate of a cam mechanism (approximately 6.1 failures per million hours) is dependent on its operating speed, friction and wear rate between the driving mechanism, cam, and follower, and the material properties of the individual parts. Equations contained in Chapter 9 can be used to estimate the wear rate of a cam mechanism. Chapter 18 of this Handbook contains some additional guidance for evaluating the design of slider-crank mechanisms. These procedures can be used to estimate the failure rate of a cam-follower mechanism.

23.6 **FITTING**

Fittings are designed to seal fluid within a fluid system. A multitude of options exist for fittings in fluid systems. One type of fitting is an all-metal fitting which relies on metal-to-metal contact. Another design employs O-rings to contain pressurized fluid. In either design, tightening threads between the mating halves of the fitting or between the fitting and component port, forces the two mating surfaces together to form a seal. Quick disconnect couplings can significantly improve the serviceability of fluid systems by saving maintenance time. Couplings incorporating valves can automatically shut off fluid flow and maintain system pressure.

It is important to consider chemical compatibility of the coupling materials with respect to fluid media and the strength of coupling materials with respect to functional requirements of the application. Plastic couplings are lightweight and used for many low pressure applications. Metal couplings are typically chosen when high fluid pressure, vibration and other more severe operating conditions call for greater strength and durability.

23.6.1 **Fitting Failure Modes**

Leakage is the primary failure mode of fluid systems. Compression fittings that incorporate an elastomeric seal offer the best seal reliability. NPT is the least reliable type of connector for high pressure hydraulic systems because the thread itself provides a leak path. The threads become deformed when tightened and subsequent loosening or tightening of the connection increases the probability of leakage. Flared fittings are inexpensive and simple in design. The metal-to-metal seal of the flare is subject to seepage at high fluid pressures. Some flare joint fittings employ a conical washer between the nose and flare to reduce the probability of leakage. A common cause of leakage from flare joints is incorrect torque. Insufficient torque results in inadequate seat contact and excessive torque can cause cold working damage to the tube and connector. Torque can also be affected by vibration, vibration also causing metal fatigue of the connector.
The reliability of a fitting employing an elastomeric seal depends on fluid compatibility and fluid temperature. Fluid operating temperatures above 85°C can cause damage to many seals. Table 23.6 provides some typical failure modes to be considered in evaluating a fitting for reliability. See Chapter 3.2 for a listing of failure modes for the seal itself.

### Table 23.6. Typical Failure Modes of a Fitting

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid seepage</td>
<td>- Damaged sleeve or ferrule</td>
<td>Eventual fitting failure</td>
</tr>
<tr>
<td></td>
<td>- Damaged O-ring</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- Incorrect assembly</td>
<td></td>
</tr>
<tr>
<td>Fractured plastic housing</td>
<td>- Shock, vibration</td>
<td>Fitting failure</td>
</tr>
<tr>
<td>Leakage</td>
<td>- Fluid incompatibility</td>
<td>- Generation of contaminants in system</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Eventual fitting and component failure</td>
</tr>
<tr>
<td>Damaged flare joint, fluid</td>
<td>- Incorrect tightening during assembly</td>
<td>Fitting failure</td>
</tr>
<tr>
<td>leakage</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Damaged seal</td>
<td>- Fluid temperature above acceptable limit</td>
<td>System leakage</td>
</tr>
</tbody>
</table>

#### 23.6.2 Fitting Failure Rate

The failure rate of a fitting can be estimated as 1.3 failures per million hours for a threaded fitting and 2.4 failures per million hours for a quick disconnect fitting based on published failure rate data. However, failure modes as discussed in the previous section need to be addressed in estimating the failure rate. NPT ports, for example, will have a higher failure rate for high-pressure hydraulic systems because the thread itself provides a leakage path. The evaluation procedures in Chapter 3 can be used to estimate the reliability of an individual O-ring seal. If the fitting contains a shut-off valve, refer to Chapter 6 to estimate its reliability.

#### 23.7 FLYWHEEL

Flywheels store kinetic energy as a mechanical battery and are used to smooth the variations in shaft speed that are caused by loads or power sources that vary in a cyclic fashion.
Overall performance of the flywheel depends on a sufficient moment of inertia, matching the power source to the load, and resulting performance requirements. One of the main considerations in flywheel design is balancing. By design, flywheels are devices with large inertia and they must, therefore, be balanced to remove eccentric loading and reduce the loading on bearings and other components.

The spinning of a flywheel creates stress at the inner hub connection which can lead to fracture. Estimating the failure rate of a flywheel must therefore consider the flywheel velocity. Rotating parts such as a flywheel can be simplified to a rotating ring to determine stress levels.

23.7.1 Flywheel Failure Modes

Flywheels develop large stresses at their inter hub connection due to dynamic forces caused by spinning. These stresses can lead to failure. Table 23.7 includes some failure modes to consider when evaluating flywheel reliability.

Table 23.7. Typical Failure Modes for a Flywheel

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flywheel loosened from shaft</td>
<td>Broken shaft/wheel connection</td>
<td>Uncontrolled release of energy</td>
</tr>
<tr>
<td>Flywheel fracture</td>
<td>Centrifugal forces</td>
<td>Complete loss of output energy</td>
</tr>
<tr>
<td>System vibration</td>
<td>Unbalanced flywheel</td>
<td>Shaft/bearing damage</td>
</tr>
</tbody>
</table>

23.7.2 Flywheel Failure Rate

The failure rate of a flywheel depends on a design balance between rotational speed, material density and tensile strength. Flywheel performance depends on an optimum energy-to-mass ratio and the flywheel must therefore spin at the maximum possible speed since kinetic energy increases only linearly with mass but as the square of rotational speed. However, a rapidly rotating object is subject to centrifugal force that can create sudden fatigue failure. While dense material can store more energy than low density material, it is also subject to higher centrifugal force and thus fails at lower rotational speed than low density material. Therefore, the tensile strength of the flywheel material becomes an important reliability consideration.
A base failure rate of 0.2 failures per million hours can be assumed for a flywheel. Table 23.8 provides some service factors to modify the expected failure rate of a flywheel for the expected operating environment.

<table>
<thead>
<tr>
<th>Driven Machinery</th>
<th>Normal Torque, Low Velocity</th>
<th>Normal Torque, High Velocity</th>
<th>High Torque, Low Velocity</th>
<th>High Torque, High Velocity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uniform</td>
<td>1.1</td>
<td>1.3</td>
<td>1.2</td>
<td>1.4</td>
</tr>
<tr>
<td>Light shock</td>
<td>1.2</td>
<td>1.4</td>
<td>1.3</td>
<td>1.5</td>
</tr>
<tr>
<td>Medium shock</td>
<td>1.3</td>
<td>1.5</td>
<td>1.4</td>
<td>1.6</td>
</tr>
<tr>
<td>Heavy shock</td>
<td>1.4</td>
<td>1.6</td>
<td>1.5</td>
<td>1.7</td>
</tr>
</tbody>
</table>

23.8 HINGE

A hinge is a joint that holds two parts together so that one can swing relative to the other. Most hinges are separate parts which are added to a completed assembly for further installation. Living hinges, on the other hand, are molded into a part during the forming process. A Butt Hinge is composed of two plates attached to abutting surfaces of a door and door jamb and joined by a pin. A Gravity Hinge closes automatically as a result of the weight of a door to which it attached. A Strap Hinge is a surface mounted hinge with long flaps of metal on each side, by which it is secured to a door and adjacent post or wall. A Pintle Hinge pivots about an upright pin or bolt. A spring loaded hinge can automatically open or close a door, eliminating the need for a latch to secure the door in its resting position.

Rotation of a hinge is permitted along only one axis, and during bending the neutral axis does not change length. Therefore, in the elastic case, the hinge can be considered as a beam undergoing pure bending, and the neutral axis forms a circular arc. Built-in plastic hinges generally undergo some plastic deformation. The hinge must stretch to compensate, the hinge undergoing tension as well as bending.

23.8.1 Hinge Failure Modes

Most hinge failures are caused by excessive loading of the part they are supporting. Typical failure modes for a hinge are listed in Table 23.9. In the event of a spring
loaded hinge, see Chapter 4, Section 4.2 for specific failure modes of a mechanical spring.

### 23.8.2 Hinge Failure Rate

The failure rate of a living hinge depends on whether strain in the hinge is purely elastic or whether plastic strain is also present. Plastic strain is further divided into pure bending strain and bending plus tensile strain. The design reliability of a hinge depends on the type of stress it is expected to carry in service.

The failure rate of both a metal hinge and a living hinge can be expected to be 0.5 failures per million cycles for normal applications. Chapter 18 provides some equations and procedures to evaluate a hinge for reliability considering the operating environment in determining the failure rate. Procedures for determining the failure rate of a hinge spring are contained in Chapter 4.

#### Table 23.9. Typical Failure Modes for a Hinge

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bent pin</td>
<td>Excessive loading</td>
<td>Hinge Failure</td>
</tr>
<tr>
<td>Weak spring</td>
<td>Shock loads</td>
<td>Hinge failure</td>
</tr>
<tr>
<td>Damaged plating</td>
<td>Corrosion</td>
<td>Eventual hinge failure</td>
</tr>
<tr>
<td>Noisy hinge</td>
<td>Contaminants, bent pin, loss of lubrication</td>
<td>Eventual hinge failure</td>
</tr>
<tr>
<td>Jammed hinge</td>
<td>Excessive loading, damaged pin</td>
<td>Hinge failure</td>
</tr>
</tbody>
</table>

### 23.9 KEYS AND PINS

Keys and pins are used on shafts to secure rotating elements such as gears and pulleys. Keys are used to enable the transmission of torque from shaft to the gear, pulley or other shaft supported element. Pins are used for axial positioning and for transfer of torque and/or thrust. Typical types of pins include cotter pins, dowel pins, hitch pins, taper pins and roll pins. The Woodruff key is often used to position a wheel against a shaft shoulder. Here the key-slot need not be machined into the shoulder stress-concentration region.
23.9.1 Key and Pin Failure Modes

Typical failure modes to be considered in the evaluation of reliability for keys and pins are included in Table 23.10.

23.9.2 Key and Pin Failure Rate

Pins are subject to loading in shear. Fatigue failure begins with a small crack. Stress concentration factors for keyways depend on the fillet radius at the bottom and ends of the keyway. Published failure rates for keys and pins are in the range of 0.35 failures/million hours.

Table 23.10. Typical Failure Modes for Keys and Pins

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sheared pin or key</td>
<td>Loose part on shaft</td>
<td>Damage to higher indenture level component</td>
</tr>
<tr>
<td>Loose pin or key</td>
<td>Enlarged holes</td>
<td>Loose part on shaft</td>
</tr>
<tr>
<td>Damaged pin</td>
<td>Shock/ vibration</td>
<td>Misalignment of part</td>
</tr>
<tr>
<td>Damaged keyway</td>
<td>Excessive axial loading</td>
<td>Axial movement of mounted part</td>
</tr>
</tbody>
</table>

23.10 PILLOW BLOCK

Pillow blocks are comprised of a bearing mounted in a housing and typically used to provide load support for a rotating shaft. In many belt drive systems the drive mechanism that couples the motor to a fan or blower is retained by pillow block bearings. Chapter 7 provides guidance on evaluating the bearing for reliability. As with most bearings, pillow block bearings are specified in terms of $B_{10}$ life and the procedures in Chapter 7 can be used to modify the $B_{10}$ life for the intended pillow block application.

23.10.1 Pillow Block Failure Modes

Pillow blocks are used to support a shaft, the mounting surface being on a parallel plane with the axis of the shaft. Elongated bolt holes in the base or feet of the unit allow for some adjustment of the pillow block. Therefore correct assembly of the pillow block is critical to achieving the expected life expectancy of the pillow block. A pillow block...
mounted at an angle with respect to the shaft can cause scoring of the shaft, heat build up and early seal leakage. The housing material for a pillow block bearing is typically made of cast iron or pressed steel and failures of the housing itself are almost nonexistent. Typical failure modes are listed in Table 23.11.

See Chapter 7, Section 7.4 for specific bearing failure modes. See Chapter 3, Section 3.2 for specific seal failure modes. See Section 23.13 later in this Chapter on set screws for specific set screw failure modes.

### 23.10.2 Pillow Block Failure Rate

Although the pillow block is comprised of a standard bearing and a housing, the effect of a misaligned pillow block on the shaft must be considered in evaluating a pillow block for reliability. Typically a pillow block will experience a failure rate of 5.0 failures per million revolutions. The $B_{10}$ life of the pillow block may need to be adjusted after considering the potential failure modes.

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Worn bearing race</td>
<td>Excessive side loading, misalignment,</td>
<td>Shaft vibration</td>
</tr>
<tr>
<td>Bearing seizure</td>
<td>Loss of lubricant, excessive rotational velocity, high temperature</td>
<td>Shaft seizure</td>
</tr>
<tr>
<td>Loss of lubricant</td>
<td>Seal leakage</td>
<td>Bearing wear</td>
</tr>
<tr>
<td>Damaged housing</td>
<td>Shock, vibration</td>
<td>Damaged pillow block</td>
</tr>
<tr>
<td>Loose shaft coupling</td>
<td>Loose set screw</td>
<td>Early shaft wear</td>
</tr>
</tbody>
</table>

### 23.11 POWER SCREWS

A power screw is a mechanical device for translating rotational motion to linear motion. Typical applications include the screw for jacks, presses and vises.

The wear life of a power screw is difficult to predict because of the number of variables to consider including load, speed, screw material, surface finish, lubrication, duty cycle, operating temperature, and environmental factors such as vibration and the existence of abrasive and corrosive contaminants. Power screws are subject to both adhesive and abrasive wear. These wear mechanisms are described in Chapter 9.
Also, Chapter 16, Section 16.3 provides the equations and procedures for evaluating the stress-strength aspects of a power screw.

Ball screws are a type of power screw. Ball screws allow balls to roll between the screw shaft and nut to achieve high efficiency. Manufacturers usually report the rated load that a given ball screw can exert for 1 million inches of cumulative travel. This is a similar rating method as for the $B_{10}$ life of ball bearings and the same relationship can be used to estimate the life of the particular power screw:

$$\frac{\lambda_{PS}}{\lambda_{PS,B}} = \left(\frac{L_A}{L_S}\right)^3 \tag{23-1}$$

where:

$\lambda_{PS} = \text{Failure rate of power screw, failures/million hours}$

$\lambda_{PS,B} = \text{Base failure rate of power screw from published life, failures/million hours}$

$L_A = \text{Equivalent radial load, lbs}$

$L_S = \text{Basic dynamic load rating, lbs}$

This concept of estimating the life of a part with respect to loading is explained in Chapter 7.

23.11.1 Power Screw Failure Modes

The nuts of a power screw system used for moving loads such as in the lead screw of a lathe are usually made of a softer material than that of the screw. This process eliminates the potential problem of material bonding and galling under load pressure if the same materials were to be used. Also, the nut is cheaper than the drive screw thus reducing maintenance costs. The failure rate of the nut can therefore be expected to have a higher failure rate than the screw. For applications that position and support heavy loads such as scaffolding, like materials are sometimes used.

Another failure mode to consider is wear. Plating of a power screw is an important design feature because of the stiction involved under load. The first level of wear on a plated part will be the plating itself. Once the plating has worn away, bare metal is exposed to the environment. For moving load applications, proper lubrication is important to achieve maximum service life. Typical failure modes for a lead screw are listed in Table 23.12.
23.11.2 Power Screw Failure Rate

The number of variables as described previously affecting the failure rate of a drive screw make it extremely difficult to predict. Without some personal experience with the design of the power screw in its intended operating environment, it is normally best to evaluate the individual parts of the power screw using the procedures in Chapters 9 and 16.

Table 23.12. Typical Failure Modes for a Power Screw

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Screw bonded to nut</td>
<td>Loss of lubrication</td>
<td>Load can not be moved</td>
</tr>
<tr>
<td>Screw bonded to nut</td>
<td>Excessive load pressure</td>
<td>Load can not be moved</td>
</tr>
<tr>
<td>Scored screw shaft</td>
<td>Misalignment</td>
<td>Eventual power screw failure</td>
</tr>
<tr>
<td>Fractured nut</td>
<td>Excessive PV</td>
<td>Failed power screw operation</td>
</tr>
<tr>
<td>Worn screw threads</td>
<td>Foreign contaminants, loading</td>
<td>Unstable power screw operation</td>
</tr>
</tbody>
</table>

23.12 RIVET

There are two common types of rivets, lap-joint and butt-joint. In a lap-joint rivet the plates to be joined overlap each other and are held together by one or more rows of rivets. In the butt-joint, the plates to be joined are in the same plane and are joined together with a cover plate which is riveted to both plates by one or more rows of rivets. Factors to be considered in evaluating a riveted joint for reliability include type of joint, spacing of rivets, type and size of rivets, hole-size and rivet material.

Both lap-joint and butt-joint rivet assemblies are subject to shear, tension, and crushing.

The shearing stress in the rivet will be:

\[
S_S = \frac{P}{A_s}
\]  

(23-2)

where:

\(S_S\) = Shearing stress, lb/in²

\(P\) = Tensile load on the joint, lbs
\[ A_S = \text{Shear area, in}^2 \]

The controlling tensile stress in the plate occurs near the rivet hole:

\[ S_T = \frac{P}{(w-d)t} \quad (23-3) \]

where:
- \( S_T = \text{Tensile stress, lb/in}^2 \)
- \( w = \text{Width of the plate, in} \)
- \( d = \text{Hole diameter, in} \)
- \( t = \text{Plate thickness, in} \)

The crushing stress due to load transfer at the contact between plate and rivet is:

\[ S_C = \frac{P}{td} \quad (23-4) \]

where:
- \( S_C = \text{Crushing stress level at the area of the rivet, lb/in}^2 \)

The total design will include a sufficient number of rivets to ensure that the failure of any one rivet because of improper installation or damage will not result in immediate failure of the structure.

**23.12.1 Rivet Failure Modes**

The primary failure modes in a conventional riveted joint assembly are rivet shear and sheet cracking due to load bearing in the sheet. In lower stress aluminum structures, failure is more frequent in the sheet, the failure mode being tension cracks starting at stress concentrations on hole-surfaces. In higher stress applications, the failure mode may be either shear of the rivets or sheet cracking.

Rivets may fail by:
- Shearing through a cross section of the rivet
- Crushing

Plates may fail by:
- Tearing along a single line from center of rivet hole to edge of plate
• Shearing along two parallel lines extending from opposite sides of the rivet hole to the edge of the plate
• Tearing between adjacent rivets
• Tensile failure of the plate
• Crushing

23.12.2 Rivet Failure Rate

Riveted structures designed in accordance with manufacturer’s specifications and assembled without additional stress raisers can be expected to have a fatigue failure rate of about 0.08 failures per million hours. Both the method of making the hole for the rivet and the presence of a countersink affect the fatigue strength. For example, for single-row joints, fatigue strength increases with decreasing pitch; for three-row joints, fatigue strength increases with increasing spacing; and for multi-row joints, fatigue strength increases with increasing numbers of rows. In evaluating rivet joint designs for reliability it is important to recognize the fact that slightly out-of-round holes can increase the failure rate by a factor of 10. Assurance of rivet assembly reliability is normally assured by trial and error experimentation on sample products.

23.13 SETSCREW

Setscrews are special fasteners designed to hold gears and pulleys on shafts. Unlike bolts and tap screws that depend on tension to develop a clamping force, the setscrew depends on compression to develop a clamping force. The resistance to axial or rotary motion of the collar or hub relative to the shaft is called the holding power. This holding power is a frictional resistance between the contact surfaces of the collar and shaft plus the penetration of the setscrew into the shaft. The primary failure mode of the setscrew-shaft combination is a loose set screw and loss of holding power. Setscrews may be a drive type or point style. Of particular importance to the failure rate of the setscrew is the size of the setscrew and the holding power provided by the clamping action. Holding power is generally specified as the tangential force in pounds. The size of the shaft to be used with a particular size of setscrew will determine the holding power required. Cup-point and cone-point setscrews for example, penetrate the shaft deeper than oval-point or flat-point setscrews.

Setscrew selection is usually based on the setscrew diameter being equal to approximately one-half the shaft diameter. Manufacturer’s data provide more exact design criteria. The shape of the screw head affects the seating torque that can be attained because it determines how much torque can be transmitted to the screw. For example, less torque can be transmitted through a slotted setscrew than a socket head setscrew, the holding power of the slotted screw is about 45% less.
When a setscrew is used in combination with a key, the screw diameter should be equal to the width of the key. In this combination, the setscrew holds the parts in an axial direction only. Torsional load on the parts is then carried by the key. The key should be tight fitting so that no motion is transmitted to the screw. Under high reversing or alternating loads, a poorly fitted key will cause the screw to back out and lose its clamping force.

23.13.1 Setscrew Failure Modes

An important consideration in evaluating a setscrew-shaft combination is the holding power provided by the clamping action. Holding power is generally specified as the tangential force in pounds, the resistance to axial or rotary motion of the collar or hub relative to the shaft. Total reliability of the setscrew-shaft combination is also dependent on the number of setscrews being used. Two setscrews give more holding power than one, but not necessarily twice as much. Holding power is approximately doubled when the second setscrew is installed in an axial line with the first but is only about 30% greater when the screws are diametrically opposed. Normally a displacement of 60 degrees is used when two setscrews need to be installed on the same circumferential line providing a holding power of 1.75 times the holding power of one setscrew.

Another failure mode to be considered in the setscrew-shaft combination is corrosion created by dissimilar materials. A thin film of rust-preventive oil is normally used to protect unplated setscrews, or cadmium or zinc plated setscrews can be used. A summary of failure modes for setscrews is shown in the following table.

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial motion of collar or</td>
<td>Shock and vibration resulting in loose setscrew</td>
<td>Loss of shaft motion</td>
</tr>
<tr>
<td>hub</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rotary motion of collar or</td>
<td>Shock and vibration resulting in loose setscrew</td>
<td>Loss of shaft motion</td>
</tr>
<tr>
<td>hub</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Damaged setscrew</td>
<td>Incorrect torque at installation</td>
<td>Loss of shaft motion</td>
</tr>
<tr>
<td>Damaged setscrew</td>
<td>Severe operating environment resulting in corroded setscrew</td>
<td>Loss of shaft motion</td>
</tr>
</tbody>
</table>
Table 23.14. Service Factor Multiplying Factors for a Setscrew

<table>
<thead>
<tr>
<th>OPERATING ENVIRONMENT</th>
<th>LOW HUMIDITY</th>
<th>HIGH HUMIDITY</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal duty</td>
<td>1.0</td>
<td>1.2</td>
</tr>
<tr>
<td>Random shock and vibration</td>
<td>1.2</td>
<td>1.4</td>
</tr>
<tr>
<td>Continuous shock and vibration</td>
<td>1.5</td>
<td>1.8</td>
</tr>
</tbody>
</table>

23.13.2 Setscrew Failure Rate

Setscrews are subject to axial and rotational loading in shear. Published failure rates for setscrews are in the range of 0.35 failures/million hours. Table 23.14 provides some recommended multiplying factors to adjust the base failure rate for the intended operating environment.

23.14 WIRE ROPE

A typical wire rope contains wire twisted in one direction to form the strands, and the strands twisted in the opposite direction to form the rope. In the completed rope the visible wires are approximately parallel to the axis of the rope. In another wire rope design the wires in the strand and the strands in the rope are twisted in the same direction so that the outer wires run diagonally across the axis of the rope. This wire rope design provides more resistance to abrasive wear and failure to fatigue but is harder to handle and more likely to kink and untwist.

When a loaded rope is bent over a sheave, the rope stretches like a string, rubs against the sheave, and causes wear of both the rope and the sheave. The amount of wear that occurs depends upon the pressure of the rope in the sheave groove. This pressure is called the bearing pressure given by:

\[ p = \frac{2F}{dD} \]  \hspace{1cm} (23-5)

where:

\[ p = \text{Bearing pressure, pounds} \]

\[ F = \text{Tensile force on the rope, lbs/in}^2 \]
The optimum wire rope for a specific application is a difficult challenge because a high rating in one design parameter generally means a lower rating in another parameter. These design parameters include:

- **Strength** – resistance to breaking - the total load to be considered includes those loads created by abrupt starts and stops and shock loads in addition to the dead weight.

- **Resistance to bending fatigue** - the sharper the bend, the higher is the probability of fatigue failure. Accelerating the rate of travel also increases the fatigue factor. Close-coupled reverse bending will further increase the probability of fatigue failure. The greater the number of wires in each strand, the greater the resistance of rope to bending fatigue.

- **Resistance to vibrational fatigue** - vibration in the system is transmitted through the wire rope in the form of shock waves. This energy is absorbed by the various components and can eventually lead to wire failure in the rope.

- **Resistance to abrasion** - abrasion can occur on drums, sheaves or whenever rope rubs against itself or other material. Abrasion also occurs internally whenever rope rubs against itself or other material. Abrasion also occurs internally whenever wire rope is loaded or bent and it weakens the rope by wearing away metal from inside and outside wires. Excessive wear can be caused by faulty sheave alignment, incorrect groove diameters, or improper drum winding.

- **Resistance to crushing** - crushing is the effect of external pressure on a rope, which damages it by distorting the cross-section shape of the rope, its strands and/or the core. When a rope is damaged by crushing, the wire, strands and core are prevented from moving and adjusting as required during operation. Crushing resistance is the ability to resist and absorb external forces.

### 23.14.1 Wire Rope Failure Modes

The main issue with the reliability of a wire rope is safety, since a failure of a wire rope can result in catastrophic consequences. The rated capacity of a wire rope is load which a new wire rope may handle under given operating conditions and at an assumed design factor. A design factor of 5 is typical for a wire rope (operating loads not to exceed 20% of the catalog breaking strength). Operating loads need to be reduced when life or valuable property are at risk. A design factor of 10 is usually chosen when
wire rope is used to carry personnel (operating loads not to exceed 10 \% of catalog breaking strength). The wire rope design must also be reviewed to assure that wire rope attachments have the same working load limit as the wire rope being used. Typical failure modes of a wire rope assembly are listed in Table 23.15.

**23.14.2 Wire Rope Failure Rate**

Wire rope manufacturers provide a fatigue diagram for their products similar to an S-N diagram. Assuming the ultimate tensile strength of the wire material is selected for the application including published safety factors, the wire rope can be expected to have a failure rate of approximately 17.5 failures per million operations.

**Table 23.15. Typical Failure Modes for a Wire Rope**

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>FAILURE CAUSE</th>
<th>FAILURE EFFECT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Breakage</td>
<td>Abrupt starts, sudden stops exceeding dead weight load</td>
<td>Loss of load</td>
</tr>
<tr>
<td>Fatigue failure</td>
<td>Sharp bending around sheave, vibrational fatigue, installation error, loss of lubricant</td>
<td>Loss of load</td>
</tr>
<tr>
<td>Wire crushing</td>
<td>External forces</td>
<td>Strands and core do not adjust resulting in erratic movement</td>
</tr>
<tr>
<td>Damaged rope</td>
<td>Faulty sheave alignment</td>
<td>Excessive wear, early failure</td>
</tr>
</tbody>
</table>

**23.15 REFERENCES**


100. Fittings and Flanges, Hydraulics & Pneumatics magazine,

24.1 INTRODUCTION

The Failure Mode, Effects, and Criticality Analysis (FMECA) is an essential design function that begins in the concept stage and continues through product development and testing. The FMECA is used to assure the reliability and maintainability of product design and adequate planning for logistics support requirements. To be effective, the FMECA must be iterative to correspond with the nature of the design process itself. The extent of effort and sophistication of approach used in the FMECA will be dependent upon the nature and requirements of the individual program. This makes it necessary to tailor the FMECA to each individual development program. Tailoring
requires extensive planning so that regardless of the degree of sophistication, the FMECA will contribute meaningfully to program decisions regarding the feasibility and adequacy of a design approach.

The usefulness of the FMECA as a design tool and in the decision making process is dependent upon the effectiveness with which problem information is communicated for early design attention. Probably the greatest criticism of the FMECA has been its limited use in improving designs. The chief causes for this have been untimeliness and the isolated performance of the FMECA without adequate inputs to the design process. Too often the FMECA is completed as design requirement as opposed to a design tool with results of the FMECA buried in a file cabinet. Timeliness is another important factor in differentiating between effective and ineffective implementation of the FMECA. While the objective of the FMECA is to identify all modes of failure within a system design, its first purpose is the early identification of all catastrophic and critical failure possibilities so they can be eliminated or minimized through design correction at the earliest possible time. Therefore, the FMECA should be initiated as soon as preliminary design information is available at the higher system levels and extended to the lower levels as more information becomes available on the items in question.

Although the FMECA is an essential reliability task, it also provides information for other purposes. The FMECA is used for assurance of subsystem design isolation, failure detection, maintainability, safety, survivability, and adequate logistics support. This coincident use must be a consideration in planning the FMECA effort to prevent the proliferation of requirements and duplication of efforts within the same development program.

The FMECA is initiated early in the design phase to aid in the evaluation of the design and to provide a basis for establishing corrective action priorities. The FMECA is an analysis procedure which includes all probable failures in a system within specified ground rules, determines by failure mode analysis the effect of each failure on system operation, identifies single failure points, and ranks each failure according to a severity classification of failure effect. This procedure is the result with steps which, when combined, provide the FMECA. These two steps are:

a. Failure mode and effects analysis (FMEA).

b. Criticality analysis (CA).

24.2 FMECA PLANNING

Planning the FMECA task involves system description development, worksheet design, procedure implementation, FMECA revisions to reflect design changes, and results analysis to provide design guidance. Establishing ground rules for conducting the analyses, identifying the lowest indenture level of analysis, identifying a coding system, and defining system failures all need to be considered in FMECA planning.
24.2.1 Worksheet Formats

FMEA and CA worksheets that organize and document the FMECA, need to be designed to optimize understanding of failure modes and effects at all indenture levels of the system. The initial indenture level of analysis is identified (item name) on each worksheet, and each successive indenture level is documented on a separate worksheet or group of worksheets. Samples of worksheets are included in Section 24.8. However, it is important that these examples be modified for the system being analyzed.

24.2.2 Ground Rules and Assumptions

Ground rules identify the FMECA approach to be used including hardware, functional or combination, the lowest indenture level to be analyzed, and general statements of what constitutes a failure of the equipment in terms of performance criteria and allowable limits. Every effort should be made to identify and record all ground rules and analysis assumptions prior to initiation of the analysis. These ground rules provide a consistent approach when completing the worksheets. It must be recognized however that ground rules and analysis assumptions may need to be added for any item if requirements change.

24.2.3 Indenture Level

The indenture level applies to the system hardware or functional level at which failures are postulated. The lowest indenture level of analysis is usually established at the lowest replaceable assembly of the system such as an actuator, valve or pump. The objective is to identify the failure modes of all parts that can contribute to an equipment failure such as an O-ring or spring. Failure modes at this level are normally included at the replaceable part level. Examples are included in the worksheets in Section 24.8.

24.2.4 Coding System

A coding system is established for consistent identification of system functions and equipment and for tracking failure modes. If a functional block diagram is available for the system, the coding system of that diagram should be used for the worksheets. Any coding system based upon the hardware breakdown structure, work unit code numbering system, or similar system can be used that is consistent with the reliability and functional block diagram numbering system to provide complete visibility of each failure mode. The same coding system must be used for the FMEA and CA worksheets to permit tracking of the failure modes.

24.2.5 Failure Definition

General statements of what constitutes an equipment failure of the item in terms of performance parameters and allowable limits for each specified output needs to be established.
Failure definitions need to consider the equipment operating environment as well as design parameter limits.

24.3 GENERAL PROCEDURE

The FMECA needs to be performed to systematically examine the system to the lowest established indenture level. The analysis identifies potential failure modes at each indenture level. In the early stages of design when system definitions and functional descriptions are not available to the established lowest indenture level, the initial analysis is performed to the lowest possible indenture level to provide optimum results. When system definitions and functional definitions become available, the analysis is extended to the established indenture level.

24.3.1 System Definition

System definition requires a review of all descriptive information available on the system to be analyzed. The following is representative of the information and data required for system definition and analysis.

- **Technical Specifications and Development Plans** - Technical specifications and development plans generally describe what constitutes and contributes to the various types of system failure. These will state the system objectives and specify the design and test requirements for operation, reliability, and maintainability. Detailed information in the plans will provide operational and functional block diagrams showing the gross functions the system must perform for successful operation. Time diagrams and charts used to describe system functional sequence will aid in determining the time-stress as well as feasibility of various means of failure detection and correction in the operating system. Acceptable performance limits under specified operating and environmental conditions will be given for the system and equipments.

  Information for developing mission and environmental profiles will describe the mission performance requirements in terms of functions describing the tasks to be performed and related to the anticipated environments for each mission phase and operating mode. Function-time relationships from which the time-stress relationship of the environmental conditions can be developed need to be established. A definition of the operational and environmental stresses the system is expected to undergo, as well as failure definitions, must be developed for an effective FMECA.

- **Trade-off Study Reports** - These reports, if available, help to identify areas of marginal and state-of-the-art design and explain any design compromises and operating restraints. This information will aid in determining the possible and most probable failure modes and related causes in the system.

- **Design Data and Drawings** - Design data and drawings identify each item and the item configuration that perform each of the system functions. System design data and drawings will usually describe internal and interface functions of the system beginning at
system level and progressing to the lowest indenture level of the system. Design data will usually include either functional block diagrams or schematics that will facilitate construction of reliability block diagrams.

- **Reliability Data** - The determination of the possible and probable failure modes requires an analysis of reliability data on the item selected to perform each of the system internal functions. It is always desirable to use reliability data resulting from reliability tests run on the specific equipment to be used with the tests performed under the identical conditions of use. When such test data are not available, procedures in this Handbook can be used to estimate the reliability of the part for the intended operating environment. Reliability data from MIL-HDBK-217 for electronic parts or from operational experience and tests performed under similar use conditions on items similar to those in the systems can be used. Source of the failure rate data is recorded on the Criticality Analysis worksheet.

### 24.3.2 FMECA Process

The FMECA is most effective if initiated as an integral part of early design process of system functional assemblies and updated to reflect design changes. This process will prove beneficial at design review meetings. The FMEA will also be useful in defining special test considerations, quality inspection points, preventive maintenance actions, operational constraints, useful life, and other pertinent information and activities necessary to minimize failure risk. The following discrete steps are used in performing an FMEA:

a. Define the system to be analyzed. Complete system definition includes identification of internal and interface functions, expected performance at all indenture levels, system restraints, and failure definitions. Functional narratives of the system should include descriptions of each mission in terms of functions which identify tasks to be performed for each mission, mission phase, and operational mode. Narratives should describe the environmental profiles, expected mission times and equipment utilization, and the functions and outputs of each item.

b. Construct block diagrams. Functional and reliability block diagrams which illustrate the operation, interrelationships, and interdependencies of functional entities should be obtained or constructed for each item configuration involved in the system's use. All potential item and interface failure modes are identified and their effect on the immediate function or item and on the system defined. Each failure mode in terms of the worst potential consequences which may result are identified and a severity classification category assigned.

c. Failure detection methods and compensating provisions for each failure mode are identified. Corrective design or other actions required to eliminate the failure or control the risk can be identified for design review and corrective action.
d. Identify effects of corrective actions or other system attributes, such as requirements for logistics support. Document the analysis and summarize the problems which could not be corrected by design and identify the special controls which are necessary to reduce failure risk.

24.3.3 Severity Classification

Severity classifications are assigned to provide a qualitative measure of the worst potential consequences resulting from item failure. A severity classification is assigned to each identified failure mode and each item analyzed in accordance with the severity classification categories defined as follows:

Category I - Catastrophic - A failure which may cause death or total system loss *

Category II - Critical - A failure which may cause severe injury, extensive property damage, or major system damage that results in loss of mission *

Category III - Marginal - A failure which may cause minor injury, minor property damage, or minor system damage that results in mission delay or degradation.

Category IV - Minor - A failure not serious enough to cause injury, property damage or mission delay or degradation, but which will necessitate repairs at a later time.

* In some cases the loss of mission may be sufficiently severe so that it may be classified as a Category I failure.

24.4 FMEA PROCEDURE

24.4.1 Analysis Purpose and Approach

The purpose of the FMEA is to study the results or effects of item failure on system operation and to classify each potential failure according to its severity. Variations in design complexity and available data will generally dictate the analysis approach to be used. There are two primary approaches for accomplishing an FMEA. One is the hardware approach which lists individual hardware items and analyzes their possible failure modes. The other is the functional approach which recognizes that every item is designed to perform a number of functions that can be classified as outputs. The outputs are listed and their failure modes analyzed. For complex systems, a combination of the functional and hardware approaches may be considered. The FMEA may be performed as a hardware analysis, a functional analysis, or a combination analysis and may be initiated at either the highest indenture level and proceed through decreasing indenture levels (top-down approach) or at the part or assembly level and proceed through increasing indenture levels (bottom-up approach) until the FMEA for the system is complete.
The functional approach is normally used when hardware items cannot be uniquely identified or when system complexity requires analysis from the initial indenture level downward through succeeding indenture levels. The functional approach is normally utilized in an initial indenture level down fashion (top-down approach); however, it can be initiated at any level of indenture and progress in either direction. Each identified failure mode is assigned a severity classification that is utilized during design to establish priorities for corrective actions.

The hardware (bottom-up) approach is normally used when hardware items can be uniquely identified from schematics, drawings, and other engineering and design data. The hardware approach is normally utilized in a part level up fashion (bottom-up approach); however, it can be initiated at any level of indenture and progress in either direction. Each identified failure-mode shall be assigned a severity classification which will be utilized during design to establish priorities for corrective actions.

The procedures contained in this Handbook assume a bottom-up hardware approach. This approach combined with the procedures in the Handbook for estimating the failure rates of hardware items in their intended operating environments provides the optimum method of performing a FMEA. After the failure modes are identified for each hardware item, the probability of occurrence is determined and then severity classifications are assigned to each failure mode and each item to provide a basis for establishing corrective action priorities. First priority is given to the elimination of identified Category I (catastrophic) and Category II (critical) failure modes. The combination of failure mode probability of occurrence and severity category permits a criticality rating of failure modes requiring additional investigation as to corrective action requirements.

24.4.2 System Definition

The first step in performing the FMEA is to define the system to be analyzed. Functional narratives are developed for each mission, mission phase, and operational mode and include statements of primary and secondary mission objectives. The narratives include system and part descriptions for each mission phase and operational mode, expected mission times and equipment utilization, functions and output of each item, and conditions which constitute system and part failure. Developing a system definition is extremely beneficial when completing the FMEA worksheets maintaining continuity at the lower indenture levels. The system definition is also important during design reviews when failure mode criticality is evaluated with respect to equipment performance and mission requirements. The following paragraphs describe those subjects that comprise a system definition.

- **Mission Functions and Operational Modes** - The system definition includes descriptions of each mission in terms of functions which identify the task to be performed and the functional mode of operation for performing the specific function. Mission functions and operational modes are identified starting at the highest system level and progressing to the lowest indenture level to be analyzed.
• **Environmental Profiles** - Environmental profiles include the anticipated environmental conditions for each mission and mission phase. When a system is to be utilized in more than one environment each different environmental profile needs to be described. The intended use, through time, of the system and its equipments are developed from the mission time statements for each environmental profile. The use time-environment phasing is used in determining the time-stress relationships and the feasibility of failure detection methods and compensating provisions in the operating system.

• **Mission Time** - A quantitative statement of system function time requirements is developed and included in the system definition. Function-time requirements are developed for items which operate in different operational modes during different mission phases and for items which function only if required.

• **Block Diagrams** - Block diagrams which illustrate the operation, interrelationships, and interdependencies of functional entities of a system are advisable to provide the ability for tracing failure mode effects through all levels of indenture. Both functional and reliability block diagrams are required to show the functional flow sequence and the series dependence or independence of functions and operations. Block diagrams can be constructed in conjunction with or after defining the system and display the system as a breakdown of its major functions. More than one block diagram will usually be required to display alternative modes of operation, depending upon the definition established for the system. All inputs and outputs of the item as a whole are shown on the diagram.

Each block on the diagram should be designated by a consistent and logical item number that reflects the functional system breakdown order. A uniform numbering system developed in functional system breakdown order is required to provide traceability and tracking through all levels of indenture.

A functional block diagram illustrates the operation and interrelationships between functional entities of a system as defined in engineering data and schematics. A functional system block diagram will provide a functional flow sequence for the each indenture level of analysis and present hardware indenture used for FMEA worksheets. A reliability block diagram defines the series dependence or independence of all functions of a system or functional group for each life-cycle event. The reliability block diagram provides identification of function interdependencies for the system.

24.4.3 **FMEA worksheet**

The documentation of the FMEA is the next step and is accomplished by completing the columns of the established FMEA worksheet. Guidelines for designing the FMEA worksheet are included in Section 25.2 above, FMECA Planning. An example of an
FMEA worksheet format is shown in Figure 25.1. The following paragraphs contain a description of FMEA worksheet entries.

- **Part Number** – A part number for the component being analyzed such as a valve or actuator being analyzed. The serial number of the component, schematic diagram symbol or drawing number can be used to properly identify the component.

- **Worksheet Code** – A sequential code number used to track worksheet entries, provide a reference between FMEA and CA worksheets and summary reports.

- **Equipment Functional Description** - A uniform identification code with general requirements is used to provide consistent identification of system functions of the equipment and provide complete visibility of each failure mode and its relationship to the system function identified in the applicable block diagram. The name or nomenclature of the item or system function being analyzed for failure mode and effects is listed.

- **Function** - A concise statement of the function performed by the hardware item is listed which includes both the inherent function of the part and its relationship to interfacing items.

- **Failure Modes and Causes** - All predictable failure modes for each indenture level analyzed are identified and described. Potential failure modes are determined by examination of item outputs and functional outputs identified in applicable block diagrams and schematics. Failure modes of individual item function are postulated on the basis of narratives and the failure definitions included in the ground rules. The most probable causes associated with the postulated failure mode are identified and described. Failure Mode Tables in the various chapters of this handbook provide listings of possible failure modes for the various mechanical components.

    Since the failure mode may have more than one cause, all probable independent causes for each failure mode are identified in the indenture levels per the indenture level analysis. To assist in assuring that a complete analysis is performed, each failure mode and output function is examined in relation to the following typical failure conditions:
    
    a. Premature operation.
    b. Failure to operate at a prescribed time.
    c. Intermittent operation.
    d. Failure to cease operation at a prescribed time.
    e. Loss of output or failure during operation.
    f. Degraded output or operational capability.
    g. Other unique failure conditions, as applicable, based upon given characteristics and operational requirements or constraints.

- **Mission Phase or Operational Mode** - A concise statement of the mission phase and operational mode in which the failure occurs is entered on the worksheet. Where a
specific event or time can be defined from the system definition and mission profiles, the most definitive timing information should also be entered for the assumed time of failure occurrence.

- **Failure Effects** - The consequences of each assumed failure mode on item operation, function, or status is identified, evaluated, and recorded on the worksheet. Failure effects focus on the specific block diagram element which is affected by the failure under consideration. The failure under consideration may impact several indenture levels in addition to the indenture level under analysis; therefore local, next higher level, and end effects need to be evaluated. Failure effects also consider the mission objectives, maintenance requirements and personnel and system safety.

  Local effects concentrate specifically on the impact an assumed failure mode has on the operation and function of the item in the indenture level under consideration. The consequences of each postulated failure affecting the item is described along with any second-order effects which result. The purpose of defining local effects is to provide a basis for evaluating compensating provisions and for recommending corrective action. It is possible for the local effect to be the failure mode itself.

  Next higher level effects concentrate on the impact an assumed failure has on the operation and function of the items in the next higher indenture level above the indenture level under consideration. The consequences of each postulated failure affecting the next higher indenture level are described.

  End effects evaluate and define the total effect an assumed failure has on the operation, function, or status of the uppermost system. The end effect described may be the result of a double failure. For example, failure of a safety device may result in a catastrophic end effect only in the event that both the prime function goes beyond limit for which the safety device is set and the safety device falls. Those end effects resulting from a double failure need to be indicated on the FMEA worksheets and can be detailed in the Remarks Column.

- **Failure Detection Method** - A description of the methods by which occurrence of the failure mode is detected by the operator is recorded on the worksheet. The failure detection means, such as visual or audible warning devices, automatic sensing devices, sensing instrumentation or other unique indications are identified.

  A description of indications which are evident to an operator that a system has malfunctioned or failed, other than the identified warning devices, are recorded. Proper correlation of a system malfunction or failure may require identification of normal indications as well as abnormal indications. If no indication exists, identify if the undetected failure will jeopardize the mission objectives or personnel safety. If the undetected failure allows the system to remain in a safe state, a second failure situation should be explored to determine whether or not an indication will be evident to all operator Indications to the operator should be described as follows:
a. Normal. An indication that is evident to an operator when the system or equipment is operating normally.
b. Abnormal. An indication that is evident to an operator when the system has malfunctioned or failed.
c. Incorrect. An erroneous indication to an operator due to the malfunction or failure of an indicator such as an instrument, sensing device or visual or audible warning device.

A description of the most direct procedure that allows an operator to isolate the malfunction or failure is recorded. An operator will know only the initial symptoms until further specific action is taken such as performing a more detailed built-in-test (BIT) or operational procedure. The failure being considered in the analysis may be of lesser importance or likelihood than another failure that could produce the same symptoms and this must be considered. Fault isolation procedures require a specific action or series of actions by an operator, followed by a check or cross reference either to instruments, control devices, circuit breakers, or combinations thereof. This procedure is followed until a satisfactory course of action is determined.

- **Compensating Provisions** - The compensating provisions, either design provisions or operator actions, which circumvent or mitigate the effect of the failure are identified and evaluated. This step is required to record the true behavior of the item in the presence of an internal malfunction or failure.

  Compensating provisions which are features of the design at any indenture level that will nullify the effects of a malfunction or failure, control, or deactivate system items to halt generation or propagation of failure effects, or activate backup or standby items or systems are described.

  Design compensating provisions include:
  a. Redundant items that allow continued and safe operation.
  b. Safety or relief devices such as monitoring or alarm provisions which permit effective operation or limits damage.
  c. Alternative modes of operation such as backup or standby items or systems.

  Compensating provisions which require operator action to circumvent or mitigate the effect of the postulated failure need to be described. The compensating provision that best satisfies the indication(s) observed by an operator when the failure occurs is determined. This may require an investigation to determine the most correct operator action(s). The consequences of any probable incorrect action(s) by the operator in response to an abnormal indication should be considered and the effects recorded.

- **Severity Classification** - A severity classification category is assigned to each failure mode and item according to the failure effect. The effect on the functional condition of the item under analysis caused by the loss or degradation of output is
identified so the failure mode effects will be properly categorized. For lower levels of indenture where effects on higher indenture levels are unknown, a failure’s effect on the indenture level under analysis is described by the severity classification categories as follows:

Category I - Catastrophic - A failure which may cause death or total system loss *

Category II - Critical - A failure which may cause severe injury, extensive property damage, or major system damage that results in loss of mission *

Category III - Marginal - A failure which may cause minor injury, minor property damage, or minor system damage that results in mission delay or degradation.

Category IV - Minor - A failure not serious enough to cause injury, property damage or mission delay or degradation, but which will necessitate repairs at a later time.

* In some cases the loss of mission may be sufficiently severe so that it may be classified as a Category I failure.

**Remarks** - Any pertinent remarks pertaining to and clarifying any other column in the worksheet line are noted in the Remarks column. This entry also may include a notation of unusual conditions, failure effects of redundant items, recognition of particularly critical design features or any other remarks that amplify the line entry.

24.5 CRITICALITY ANALYSIS PROCEDURE

24.5.1 Analysis Purpose and Approach

The purpose of the Criticality Analysis (CA) is to rank each potential failure mode identified according to the combined influence of severity classification and its probability of occurrence based upon the best available data. As outlined in Section 25.2, a qualitative or quantitative approach to the Criticality Analysis can be used. The qualitative approach is appropriate when specific failure rate data are not available. As part of this Handbook, the quantitative approach is assumed. As parts configuration data and failure rate data become available, criticality numbers should be calculated and incorporated in the analysis. The failure probability levels, when used, should be modified as the system is better defined. There are failure rate data sources other than this Handbook such as MIL-HDBK-217 for electronic parts, OREDA and NPRD. Section 25.7, References, provides information on these sources.

24.5.2 CA Worksheet

Calculation of a criticality number for each failure mode and component part is accomplished by completing the columns of the planned CA worksheet. An example of
a CA worksheet format is shown in Figure 24.3. For tracking purposes, the following information is transferred from the FMEA worksheet to the CA worksheet:

- Part Number
- Worksheet Code
- Equipment Functional Description
- Function
- Failure modes and causes
- Mission phase or operational mode
- Severity Classification

Other CA worksheet entries are as follows:

- **Failure Rate Data Source** – The source of the failure rate data such as this Handbook, reliability testing data, MIL-HDBK-217, OREDA or NPRD is recorded on the worksheet. See Section 24.7 for identification of failure rate data sources.

- **Base Failure Rate** – The Base Failure Rate from the failure rate data source is recorded on the worksheet. The base failure rate is typically the failure rate found in the failure rate data source that must then be adjusted for the intended operating environment.

- **Adjustment Factors** - The base failure rate from the failure rate data source may need to be adjusted for the operating and environmental conditions of the equipment being analyzed. Adjustment factors may be the multiplying factors as contained in the various chapters of this Handbook, they may be the analyst’s best judgment or it may be that no adjustment factor is needed for example if testing data for the particular part is being used. Explanations can be included in the Remarks column.

- **Part Failure Rate** - The part failure rate ($\lambda_p$) as calculated using the procedures in this Handbook or from the appropriate failure rate data source is entered on the worksheet. Where appropriate, application factors, environmental factors, and other factors as may be required are applied to the base failure rates ($\lambda_b$) obtained from handbooks or other reference material to adjust for differences in operating stresses.

- **Failure Effect Probability** - $\beta$ values are the conditional probability that the failure effect will result in the identified criticality classification, given that the failure mode occurs. The $\beta$ values represent the analyst’s judgment as to the conditional probability the loss will occur. Typical $\beta$ values are as follows:
  - Actual loss = 1.00
  - Probable loss = >0.10 to <1.00
  - Possible loss = >0.0 to 0.10
  - No effect = 0.0

- **Failure Mode Ratio** - The fraction of the part failure rate ($\alpha$) related to the particular failure mode under consideration is evaluated by the analyst and recorded. The failure
mode ratio is the probability expressed as a decimal fraction that the part or item will fail in the identified mode. If all potential failure modes of a particular part or item are listed, the sum of the $\alpha$ values for that part or item will equal one. Individual failure rate multipliers may be derived from failure rate source data or from test and operational data. If failure mode data are not available, the $\alpha$ values will represent the analyst’s judgment based upon analysis of the item’s functions.

- **Duty Cycle** – In many cases the component being analyzed is only used part of the time with respect to the total equipment operating time. Examples include a valve or an actuator that is used only for a particular mode of equipment operation.

- **Detectability** - The probability that the failure will be detected by the operator in event of component failure.

- **Operating Time** - The operating time in hours or the number of operating cycles of the component per mission is derived from the system definition and listed on the worksheet.

- **Failure Mode Criticality Number** - The value of the failure mode criticality number ($C_m$) is calculated and listed on the worksheet. $C_m$ is the portion of the criticality number for the item due to one of its failure modes under a particular severity classification. For a particular severity classification and operational phase, the $C_m$ for a failure mode may be calculated with the following formula:

  $$C_m = \lambda_p \beta \alpha d t$$

  where:

  - $C_m$ = Criticality number for the failure mode
  - $\lambda_p$ = Part Failure Rate
  - $\beta$ = Conditional probability that failure mode results in listed end effect
  - $\alpha$ = Failure Mode Ratio
  - $d$ = Probability that the failure mode will be detected by the operator or other means such as an alarm or monitor
  - $t$ = Duration of applicable operating hours or mission phase (hours or cycles)

- **Item Criticality Number** - The second criticality number calculation is for the item under analysis. Criticality numbers ($C_r$) for the items of the system shall be calculated and listed on the worksheet. A criticality number for an item is the number of system failures of a specific type expected due to the item’s failure modes. The specific type of system failure is expressed by the severity classification for the item’s failure modes.

  For a particular severity classification and mission phase, the $C_r$ for an item is the sum of the failure mode criticality numbers.
\[ C_r = \sum_{n=1}^{j} (\lambda_n \beta \alpha d t) n \]

where:
- \( C_r \) = Criticality number for the item
- \( n \) = The failure modes in the item that fall under a particular criticality classification
- \( j \) = Last failure mode in the item under the criticality classification

- **Remarks** - Any pertinent remarks pertaining to and clarifying any other column in the worksheet line are noted in the Remarks column. Examples include a description of the failure rate data source, adjustment factors and \( \alpha \) or \( \beta \) factors.

### 24.6 CRITICALITY MATRIX

The criticality matrix provides a means of identifying and comparing each failure mode to all other failure modes with respect to severity. The matrix is constructed by inserting item or failure mode identification numbers in matrix locations representing the severity classification category and the probability of occurrence. The resulting matrix display shows the distribution of criticality of item failure modes and provides a tool for assigning corrective action priorities. A sample Criticality Matrix is shown in Figure 25.3. The frequency of occurrence can be displayed as a probability of occurrence, as a severity level or failure rate. The numbers within the blocks of Figure 25.3 represent the identification numbers of individual failure modes. The resulting display indicates the relative criticality of the failure modes.

### 24.7 REFERENCES

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment".

88. Reliability Analysis Center, “Nonelectronic Parts Reliability Data”, NPRD-95


### 24.8 WORKSHEET EXAMPLES

The following pages contain examples worksheets used for the FMEA and CA. It is important to note that these examples need to be modified for the system being analyzed. The preceding Sections of this Chapter need to be reviewed thoroughly before deciding on the Worksheet formats.
<table>
<thead>
<tr>
<th>Part Number (S/N)</th>
<th>Worksheet Code</th>
<th>Equipment Functional Description</th>
<th>Function ID No.</th>
<th>Function</th>
<th>Mode ID No.</th>
<th>Failure Modes</th>
<th>Cause ID No.</th>
<th>Failure Causes</th>
<th>Mission Phase or Operational Mode</th>
<th>Failure Effects</th>
<th>Failure Detection Method</th>
</tr>
</thead>
<tbody>
<tr>
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</table>

**Figure 24.1 (Page 1 of 2)**
## FMEA
Data Collection Worksheet

<table>
<thead>
<tr>
<th>Compensating Provisions</th>
<th>Severity Class</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Provisions in the design or operator actions which circumvent or mitigate the effect of failure such as redundancy, alarm provisions or alternate modes of operation</td>
<td>Severity Class of I, II, III, IV</td>
<td>Remarks clarifying or amplifying worksheet entries such as design features, safety provisions, etc.</td>
</tr>
<tr>
<td></td>
<td>I</td>
<td>Category I is catastrophic (a failure which may cause death or total property damage)</td>
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<td>II</td>
<td>Category II is a critical failure, one that may cause severe injury, extensive property damage or loss of mission</td>
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<td>III</td>
<td>Category III is a marginal failure effect, one that may cause minor injury, minor property damage or mission degradation</td>
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<tr>
<td></td>
<td>IV</td>
<td>Category IV is a minor failure effect, a failure mode not serious enough to cause injury, property damage, but may necessitate repairs</td>
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</tbody>
</table>

Figure 24.1 (Page 2 of 2)
## Criticality Analysis

### Data Collection Worksheet

<table>
<thead>
<tr>
<th>Part Number (S/N)</th>
<th>Worksheet Code</th>
<th>Equipment Functional Description</th>
<th>Function ID No.</th>
<th>Function</th>
<th>Mode ID No.</th>
<th>Failure Modes</th>
<th>Cause ID No.</th>
<th>Failure Causes</th>
<th>Operational Mode</th>
<th>Failure Rate Data Source</th>
<th>Base Failure Rate ( (A_b \times 10^6) )</th>
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<tbody>
<tr>
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<td>1</td>
<td>Only one failure mode should be entered for each row because of the individual effects, a and b values and compensating provisions</td>
<td>1</td>
<td>Description of all probable independent causes of this specific failure mode</td>
<td>The mode in which the equipment is operating when the failure mode occurs or effects equipment operation</td>
<td>RAC, OREDA, NPRD, NSWC Handbook</td>
<td>Failure rate from source</td>
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Figure 24.2 (Page 1 of 2)
## Criticality Analysis

**Data Collection Worksheet**

<table>
<thead>
<tr>
<th>Adjustment Factors (fr Factors)</th>
<th>Part Failure Rate (A_p \times 10^6)</th>
<th>Failure Effect Probability (p)</th>
<th>Failure Mode Ratio (e)</th>
<th>Duty Cycle (DC)</th>
<th>Detectability (d)</th>
<th>Operating Time (t)</th>
<th>Failure Mode Criticality Number (C_n)</th>
<th>Item Criticality Number (C_{ni})</th>
<th>Severity Class</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Any correction factors required to convert acquired failure rate to application failure rate</td>
<td>Base failure rate multiplied by (\alpha) factor</td>
<td>Conditional probability that identified failure mode will result in end effect (0.0 to 1.0)</td>
<td>Ratio of part failure rate related to identified failure mode</td>
<td>A value between 0.0 and 1.0 indicating the time ratio the component is functioning when the equipment is in operation</td>
<td>Probability that failure mode will be detected prior to failure</td>
<td>The operating time in hours or operating cycles between overhauls</td>
<td>(\lambda_p \times \beta \times \alpha \times d \times t)</td>
<td>Criticality number for the part</td>
<td>The Severity Category from FMEA worksheet</td>
<td>Remarks to clarify or amplify entries on the worksheet such as explanations of failure rate sources, adjustment factors or calculations,</td>
</tr>
<tr>
<td>Category I is catastrophic (a failure which may cause death or total property damage)</td>
<td>Category II is a critical failure, one that may cause severe injury, extensive property damage or loss of mission</td>
<td>Category III is a marginal failure effect, one that may cause minor injury, minor property damage or mission degradation</td>
<td>Category IV is a minor failure effect, a failure mode not serious enough to cause injury, property damage, but may necessitate repairs</td>
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</tbody>
</table>

**Figure 24.2 (Page 2 of 2)**
Figure 24.3 Sample Criticality Matrix


28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment".


63. Thomas Couplings Applications Manual
64. Bolam, J.R., "Coupling Alignment: The Reverse Indicator Method Simplified", P/PM Technology, July/Aug 90
66. Robertson, R., and Smith, B., "Why Flexible Couplings Fail", Plant Engineering and Maintenance, Jun 89

71. Dennis N. Pratt, "Investigation of Spline Coupling Wear", Report No. SY-51R-87, December 1987, Naval Air Warfare Center, Patuxent River, Maryland


73. Dennis Pratt, "Results of Air Compressor Reliability Investigation", Report No. TM 88-38 SY, January 1989, Naval Air Warfare Center, Patuxent River, Maryland

74. D. Pratt, "Results of Gear Pump Reliability Investigation", Report No. TM 89-24 SY, February 1990, Naval Air Warfare Center, Patuxent River, Maryland

75. D. Pratt, "Results of Centrifugal Pump Reliability Investigation", Report No. TM 89-69 SY, February 1990, Naval Air Warfare Center, Patuxent River, Maryland

76. D. Pratt, "Results of Pneumatic Impact Wrench Reliability Investigation", Report No. TM 90-88 SY, December 1990, Naval Air Warfare Center, Patuxent River, Maryland


80. D. Pratt, "Results of Air Force MC-2A Air Compressor Unit Reliability Investigation", Report No. TM 92-89 SY, March 1993, Naval Air Warfare Center, Patuxent River, Maryland

81. D. Pratt, "Results of Dayton 5A701 Linear Actuator Reliability Investigation", Report No. TM 93-89 SY, February, 1994, Naval Air Warfare Center, Patuxent River, Maryland

82. TD 84-3 R&M Data for Industrial Plants, A.P. Harris and Associates, 1984


86. Allan Budris, Eugene Subini, R. Barry Erickson, ”Pump Reliability – Correct Hydraulic Selection Minimizes Unscheduled Maintenance”, PumpLines, Fall 2001


88. Reliability Analysis Center, “Nonelectronic Parts Reliability Data”, NPRD-95


91. Renold Transmission Chain Selection Procedure

92. PST =>Solutions! ,Volume 3, October 1996


100. Fittings and Flanges, Hydraulics & Pneumatics Magazine,


106. Design and Analysis of Machine Elements, Douglas Wright, Department of Mechanical and Materials Engineering, The University of Western Australia


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The index provides a rapid way of locating reliability information for a specific subject within the Handbook. If an entire chapter is devoted to that subject, the Chapter number only is referenced. If the item is part of one or more Chapters, the location(s) within the Chapter are listed. If you can not locate a particular part in the Handbook, please contact the individual identified in the Preface.

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