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MILITARY HANDBOOK

RELIABILITY/DESIGN THERMAL APPLICATIONS

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Reliability/Design, Thermal Applications

1. This handbook was developed by the Department of Defense in accordance with established procedure.

2. This publication was approved on 19 January 1978 for printing and inclusion in the military handbook series.

3. This document provides basic and fundamental information on the thermal design of military electronic equipment. It will provide information and guidance to personnel concerned with such design. The handbook is not intended to be referenced in purchase specifications except for informational purposes, nor shall it supersede any specification requirements.

4. This handbook will be reviewed periodically to insure its completeness and currency. Beneficial comments (recommendations, additions, deletions) and any pertinent data which may be of use in improving this document should be addressed to: Naval Electronic Systems Command (Code 5043), Department of the Navy, Washington, D.C. 20360, by using the self-addressed Standardization Document Improvement Proposal (DD Form 1426) appearing at the end of this document, or by letter.

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1. SCOPE

1.1 Purpose. This handbook has been prepared specifically to guide engineers in the thermal design of electronic equipment with improved reliability. The primary purposes are: to permit engineers and designers, who are not heat transfer experts, to design electronic equipment with adequate thermal performance with a minimum of effort; to assist heat transfer experts, who are not electronic experts; to aid engineers in better understanding the thermal sections of Department of Defense specifications and standards for equipment; and to assist Navy personnel in evaluating thermal design during the various stages of equipment procurement and development.

1.2 Scope. This handbook recommends and presents electronic parts stress analysis methods which lead to the selection of maximum safe temperatures for parts so that the ensuing thermal design is consistent with the required equipment reliability. These maximum parts temperature must be properly selected since they are the goals of the thermal design, a fact which is often overlooked. Many thermal designs are inadequate because improper maximum parts temperatures were selected as design goals. Consequently, the necessary parts stress analysis procedures have been emphasized. Specific step by step thermal design procedures are given in chapter 4.

Proper operation at the desired performance and reliability levels can only be achieved if the electronic, thermal, and mechanical designs are all well executed and carefully integrated. Such a result can be accomplished best by the equipment designers, who must control all pertinent factors. It must be emphasized that the thermal design is fully as important as the circuit design.

Poor communications, lack of funds, and obsolete, incomplete, and unreliable data have hampered the development of proper thermal design. Methods of predicting the thermal performance of electronic equipment are becoming commonly known, but most organizations that have produced successful designs have achieved their goals by techniques peculiar to a specific equipment design.

This handbook supersedes certain Navy thermal design manuals previously published, such as NAVSHIPS 900, 192 (Design Manual of Natural Methods of Cooling Electronic Equipment), NAVSHIPS 900, 194 (Design Manual of Methods of Forced Air Cooling Electronic Equipment), and NAVSHIPS 900, 195 (Design Manual of Methods of Liquid Cooling Electronic Equipment).

This handbook covers the thermal design of shipboard, ground based, airborne (avionic), and space electronics. Simplified methods of design calculation, including nomographs and curves, are included so that engineers without heat transfer background can design acceptable equipment. Alternatively, the proper mathematical expressions are included along with comments on computer analysis methods so that experts in heat transfer can utilize these sophisticated techniques. When specific recommendations are given, care must be taken that the adoption of such recommendations does not conflict with the requirements of the contracting activity.

Throughout this handbook, equations intended specifically for use in design are labeled "D.E." (Design Equation).

We wish to express our appreciation for the productive cooperation we have received during this program from government agencies and industry. Source materials are listed in the References together with reference numbers at pertinent locations in the discussion. This handbook is not to be construed as an endorsement of any commercial products or techniques mentioned herein.

1.3 Need for adequate thermal performance. High temperature is a particularly insidious enemy of most electronic parts because it causes slow progressive deterioration rather than catastrophic failure. The mean time to failure of each part is a statistical function of its stress level and the entire complex of thermal history and chemical structure. Some authorities consider that inadequate cooling is presently the primary cause of poor reliability in military electronic equipment.

It is the failure of individual parts that leads to equipment failure. Electronic parts are prone to premature failure due to overstress (i.e., thermal, electrical or mechanical stress). Electrical and thermal stresses are closely interrelated and a reduction in electrical power dissipation correspondingly tends to ease the thermal stress. Examination of MIL-HDBK-217 (Reliability Prediction of Electronic Equipment) shows that failure rates of typical parts vary significantly with temperature. The following table presents a few extreme examples to indicate the effects:

TABLE I. Failure Rates

Part Description	λ_b Failures Per Million Hours. Base Failure Rate		ΔT°c	Ratio of High to Low Failure Rate
	High Temperature	Low Temperature		
PNP Silicon Transistors	.063 at 130°c and 0.3 stress	.0096 at 25°c and 0.3 stress	105	7:1
NPN Silicon Transistors	.033 at 130°c and 0.3 stress	.0064 at 25°c and 0.3 stress	105	5:1
Glass Capacitors	.047 at 120°c and 0.5 stress	.001 at 25°c and 0.5 stress	95	47:1
Transformers and Coils MIL-T-27 Class Q	.0267 at 85°c	.0008 at 25°c	60	33:1
Resistors Carbon Comp	.0065 at 100°c and 0.5 stress	.0003 at 25°c and 0.5 stress	75	22:1

This information shows that the certain components (capacitors, resistors, coils, and transformers) are actually more temperature sensitive than transistors. Decreases in failure rates as great as those shown above are not always attainable, but very significant reductions can be and have been achieved by reduction of thermal stress (temperature). Cooling systems must be designed to control parts temperatures to the desired levels under all anticipated thermal environments.

Significant improvements in reliability and availability have been achieved by modifying the cooling systems in various existing shipboard and airborne electronic equipments. In several instances the reliability gains were as great as 500 percent in MTBF improvement. Even so the improved cooling systems were not of optimum thermal design; rather they could be better described as salvage jobs. Further, these equipments were used equipments composed of parts with a previous history of severe thermal stress. If optimum thermal designs had been applied with new parts, it was estimated that the reliability gains would have been increased by a factor of three. (Reference 1)

It has been proven that the additional cost of designing and implementing adequate thermal performance into equipment is very worthwhile. In several validated cost analyses, the average costs of thermally improving equipments were paid for by maintenance cost savings alone during the initial six months of equipment operation. The important and priceless gains in availability were excluded in the analysis and in this sense were free. Had these thermal improvements been incorporated into the equipments during their R&D phase, the costs could have been amortized, on the average, during the first three to four months of operation. Thus, on a life cycle cost basis the Navy can achieve significant economies through additional investments in adequate thermal design. (Reference 2)

Adequate thermal design results in the minimization of parts temperature excursions when power dissipation or environmental temperatures vary. Temperature cycling in excess of ± 15°C has been found to significantly reduce parts life and reliability. Failure rate increases as large as 8.1 have been observed when the temperature cycling exceeded ± 20°C. (Reference 3)

2. REFERENCED DOCUMENTS

2.1 Issues of documents. The following documents form a part of this handbook to the extent specified herein.

SPECIFICATIONS

MILITARY

MIL-E-16400 - Electronic, Interior Communication and Navigation Equipment, Naval Ship and Shore: General Specification for

MIL-M-28787 - Standard Electronic Module Program: General Specification for

STANDARDS

MILITARY

MIL-STD-1378 - Requirements for Employing Standard Electronic Modules

MIL-STD-1389 - Design Requirements for Standard Electronic Modules

HANDBOOK

MILITARY

MIL-HDBK-217 - Reliability Stress and Failure Rate Data for Electronic Equipment

3. DEFINITIONS

3.1 The following terminology, which is utilized throughout this handbook is recommended:

Part - A small element which is used to form electronic equipment. Normally, it would not be further disassembled into its constituents, for example, a resistor, a transformer, a screw, a capacitor, an integrated circuit or a transistor.

Assembly - A group of parts which usually performs one or more detailed electronic functions and can be readily removed without special tools from electronic equipment, for example, a packaged plug-in audio amplifier.

Chassis - A single continuous supporting frame or other structure.

Unit - A mechanical group of parts, or assemblies provided within a single cabinet or other enclosure.

Set - A group of assemblies or units which performs an overall series of complete electronic functions, for example, a radar set.

System - A group of sets, specially integrated, but which may be in different locations, for example, a guidance system.

Point surface temperature - The average temperature at a specified location on a surface.

Hot spot - A region or area of high temperature.

Internal temperature - The temperature of a gas, solid, or liquid at a specified location within an enclosure.

Heat dissipation - The difference between the electrical input and output of an electronic device expressed in watts.

Unit heat dissipation - The heat dissipation per unit surface area expressed in watts per square inch.

Heat concentration - Heat dissipation per unit volume expressed in watts per cubic inch or cubic foot dependent upon the relative size of the device under consideration.

Ultimate sink - A body of matter to which thermal energy, in its path from a heat source, is ultimately delivered.

Thermal environment - The condition of (1) fluid type, temperature, pressure and velocity; (2) surface temperatures, configurations, and emissivities; and, (3) all conductive thermal paths surrounding an electronic device.

Custodians:

Army - EL
Air Force - 11

Review activities:

Army - AR
Navy - SH, AS
Air Force - 17, 85, 19, 15, 13

User activities:

Navy - MC, YD

Preparing activity:

Navy - EC
(Project RELI-0005)

4. APPROACHES TO THERMAL DESIGN

4.1 Fundamental aspects and procedures. The useful power output of electronic equipment is usually very small compared with the input power required to operate the circuitry. The excess power is, of course, dissipated in the form of heat. As long as the circuit component parts are large, few, and well separated, and air is permitted to circulate freely, the dissipation of heat causes few problems. Reduction in physical size brings the smaller component parts closer together with consequent concentration of heat within a smaller space. Current electronic design practice places great emphasis on small size with very high component part density. The resulting concentration of heat requires proper application of thermal design principles to permit effective heat removal so that the equipment will operate within the required temperature limitations.

Modern electronic equipment must incorporate a cooling system to provide a path of low thermal resistance (high thermal conductance) from heat-dissipating component parts to an ultimate heat sink of reasonably low temperature.

Several important factors must be thoroughly understood before undertaking the thermal design of electronic equipment. They include the following:

4.1.1 The quantity of dissipated heat controls the temperature rise and hence, the operating temperature in any given configuration.

4.1.2 The division of the total heat into the three transfer modes, e.g., conduction, convection, and radiation, is an inverse proportion to the thermal resistances of the three modes.

4.1.3 Under steady-state conditions, thermal equilibrium or heat balance exists.

4.1.4 The natural law of conservation of energy applies and all dissipated heat is rejected. A low-thermal-resistance path to a heat sink produces a large temperature gradient.

4.1.5 The important parameters in thermal design are power dissipation, thermal resistance, and temperature.

4.1.6 Temperature is a measure of the quality of effectiveness of the thermal design.

4.1.7 The cooling system used should be the simplest and most economical one that fits the particular electrical and mechanical designs, the environmental conditions, and the specifications.

4.1.8 Factors to be considered in thermal design (with relative emphasis dependent upon conditions of usage) include: size and weight, heat dissipation, economy, temperature limitations of component parts consistent with the required failure rates, circuit configuration, thermal environment, heat concentration, and the conditions of the ultimate sink.

4.1.9 Thermal design should start simultaneously with the electrical and mechanical designs.

4.1.10 Thermal design must not adversely affect electronic performance.

4.1.11 In cases where optimum thermal and electronic designs are not compatible with each other, a compromise may be necessary.

4.1.12 Relatively large tolerances are inherent in thermal design.

4.1.13 Mathematical analysis of the cooling system should be made early in the design process.

4.2 Fundamental thermal principles.

4.2.1 Heat (thermal energy) is transferred from one region to another by virtue of temperature difference. The two fundamental axioms are that heat flows from a high temperature region to one of lower temperature and that the heat emitted by the high temperature region must be exactly equal to that absorbed by the low temperature region.

4.2.2 When heat is transferred at a steady rate and the temperature at any given point is constant, the steady state is said to exist. On the other hand, if the heat flow is a function of time, the flow is said to be in the transient state.

4.2.3 In general, there are three modes or methods of heat transfer; conduction, convection, and radiation. They may occur singly or in combination. While evaporation and condensation may be classified under convection, for the purposes of this handbook they are treated separately, since mass transfer as well as heat transfer occurs.

4.2.4 Heat conduction is considered to be caused through molecular oscillations in solids and elastic impact in liquids and gasses. When heat is transferred by conduction the heat flow relationships are analogous to Ohm's law for electric current flow; that is, the rate of heat transferred is analogous to current flow, the temperature differences are analogous to voltage drops and the thermal resistance to heat transfer is analogous to electrical resistance.

4.2.5 The process of heat transfer from the surface of a solid to moving masses of fluids, either gaseous or liquid, is known as convection. This mode of heat transfer is brought about through circulation of the fluid.

4.2.6 All bodies continuously emit thermal radiation in the form of electromagnetic waves ranging in wavelength from the long infrared to the short ultraviolet. Radiation emitted from a body can travel undiminished through a vacuum or through gasses with relatively little absorption. When radiation is intercepted by a second body, part may be absorbed as thermal energy, part may be reflected from the surface, and part may be transmitted, still in electromagnetic wave form, through the body as in the case of glass.

4.2.7 It must be emphasized that adequate thermal design including effective cooling of parts is not a cure for electrical overstress. Voltage in excess of rated values impairs the reliability of dielectrics. Power dissipation in excess of rated values raises material temperatures within parts and lowers reliability. Thermal and electrical design must be closely coordinated during new equipment development to achieve the required reliability.

4.3 Design approaches. There are two approaches to a satisfactory electronic thermal design. The first is to use component parts capable of withstanding the temperatures which occur. In this "brute force" method, the temperatures of the various component parts are permitted to rise haphazardly until thermal equilibrium is reached for each component part. This method has serious shortcomings and is not recommended.

The second and recommended method is the controlled heat-removal method. Effective heat transfer techniques establish low-thermal-resistance paths from the various component parts to a suitable heat sink. Small temperature gradients, and consequently low operating temperatures, protect temperature-sensitive parts and circuits. Inherent in the controlled-heat-removal method is the directing of heat flow along specified paths from the various sources to a low-temperature sink. This prevents the heat from being indiscriminately scattered and transferred into adjacent parts.

4.4 Basic thermal design procedures. The purpose of thermal design is to control the temperatures of the electronic parts in an equipment so that they will not exceed specific maximum safe temperatures and to minimize the parts temperature variations under all environmental conditions in which the equipment will operate. The maximum safe temperatures must be calculated based on a parts stress analysis and must be consistent with the required equipment reliability and the failure rate assigned to each part.

It is usually necessary to maximize the heat transferred by a single mode only in order to obtain adequately low thermal resistances within an equipment. Even though a complete cooling system may include several modes of heat transfer, each particular heat path will usually emphasize a single mode. The utilization of a single heat transfer by other modes can often be ignored. For example, with metallic conduction as the predominant mode, the conductive thermal resistance can be so low that the heat transferred by radiation and convection is almost negligible. That is, in the electrothermal analogue, the shunt thermal resistances due to radiation and convection are so large that they are insignificant for design purposes.

It is recommended that this single mode heat transfer concept be used whenever possible in thermal design.

4.5 Step-by-step procedure. The following three chapters (5, 6, and 7) describe the important design procedures necessary to thermal design. In summary, the steps to be followed are:

4.5.1 Study in detail the applicable specifications, standards, and other documents pertinent to the equipment to be designed so that the thermal requirements are completely understood. Establish the maximum and minimum environmental temperature ranges of the equipment heat rejecting surfaces, heat sinks or coolants.

4.5.2 Ascertain the available cooling techniques and constraints. Which cooling techniques are permissible? What coolants are available? What are their temperatures, pressures, flow rates, etc.?

4.5.3 Perform a stress analysis of the individual electronic parts of the equipment and determine the maximum allowable temperature for each part consistent with the equipment reliability and the failure rate assigned to that part. Determine the power dissipated by each heat producing part.

4.5.4 Develop an electrical analog network of the complete heat flow circuit from all the electronic parts to the coolant or the heat sink surfaces. Assign the maximum allowable temperatures to the parts and the maximum environmental temperatures to the equipment heat sinks or coolants.

4.5.5 Calculate the unit heat concentrations (watts/cu.in.), based on preliminary packaging considerations.

4.5.6 Where necessary, calculate the thermal resistances internal to parts and establish maximum parts surface temperatures.

4.5.7 Determine the overall thermal resistance values required in the circuit from parts surfaces to sinks or coolants.

4.5.8 Subdivide the thermal resistances arbitrarily (based on the unit heat concentration and information discussed later) and tentatively assign thermal values to groups of resistors in the analogue network. Subdivision is usually based on packaging considerations, for example, groups of parts in a given module, drawer or section of a chassis or on a single card. Next, combine certain resistors in parallel using a single heat flow path from that group of parts to the sink or to an intermediate sink.

4.5.9 Evaluate the various values of tentatively assigned thermal resistance to ascertain if the values are realistic and practicable in terms of the available and permissible cooling techniques. (The ranges of thermal resistances attainable with the various cooling techniques are later given in chapter 7.)

4.5.10 Select the cooling technique or mode of heat transfer applicable to each thermal resistance in the circuit.

4.5.11 Readjust values and converge until satisfactory and practical values of thermal resistance are selected.

4.5.12 Perform a preliminary thermal design for each thermal resistance in the circuit using the design procedures given in chapters 8, 9, 10, 11, 12, 13, and 18 pertinent to the particular cooling techniques selected.

4.5.13 Evaluate the penalties related to the selected cooling techniques. Study alternative techniques and optimize based on constraints peculiar to the requirements of the equipment being designed.

4.5.14 Finalize the thermal design using procedures given in the design chapters.

4.5.15 Thermally evaluate the first model of the equipment and validate the thermal performance using procedures given in chapter 14.

4.6 Stress effects and conservatism

4.6.1 Parts stress analysis procedures (later discussed in this manual) are strongly recommended to determine that maximum thermal stress levels of parts are consistent with the required equipment reliability. Parts stress analysis is not derating. Rather, it involves the use of the part at its optimum rating for a particular application in terms of all stresses. The nominal rating of a part can be misleading, since it usually represents the most optimistic and ideal application of that part and this situation almost never occurs in military electronic equipment.

4.6.2 Conservatism must be the aim of every design for reliable equipment. Consequently, an adequate safety factor should be designed into the equipment. Structural designers, for example, stress the steel in bridges to one quarter to one fifth of its ultimate tensile strength. But, electronic engineers frequently operate parts at their maximum ratings. With even a small safety factor, the reliability of electronic equipment will be increased. Further, it must be remembered that the ratings of electronic parts are a compromise based upon economics, life, and performance. Sometimes the safety factor in part ratings is extremely small.

4.7 Thermal reliability requirements. The specifications pertinent to an equipment to be designed usually lists the maximum temperature of the environment in which the equipment must function and meet the required reliability. Alternatively, a given supply of coolant is specified at a particular temperature (or range of temperatures), flow rate and pressure. In the former situation, the equipment must reject the dissipated heat to the environment and, in the later case, the heat must be transferred to the coolant.

The goal of the thermal design is to provide heat flow paths of adequately low thermal resistance so that the parts temperatures do not exceed the maximum values determined by the parts stress analysis under the maximum conditions of environmental temperature or minimum coolant effectiveness.

Future specifications will likely mandate parts thermal stress analysis. This is particularly desirable because the maximum parts temperatures which are the thermal design goals then will be properly determined. "How hot is too hot" must be determined with the best possible accuracy. A thermal design based on incorrect goals will be inadequate.

4.8 Methods of thermally rating parts and equipment

4.8.1 Many electronic parts developed in recent years are rated in terms of external surface temperatures at given locations and the internal thermal resistances from these surfaces to the most temperature-sensitive internal elements are given. This rating procedure is recommended. It is specific and precise and determines the thermal state of the internal elements. Other electronic parts, developed in previous years are in general, rated individually for certain performance at specified ambient temperatures. The ambient temperature for a part is the temperature of the surrounding medium in which a part must operate. Ambient temperatures are often indeterminate with densely packaged parts. Ambient temperature ratings define the convective heat transfer local to a part and do not completely define the thermal conditions, since heat transfer by radiation and conduction are not specified. With densely packaged equipment, the local air temperature (ambient) surrounding a part is not necessarily related to the parts temperature due to heat radiation or conduction effects to or from other nearby parts. These effects are frequently significant and can lead to the overheating of parts even though the ambient temperature rating appears not to be exceeded. Ambient temperature rating sufficed for the type of equipment used two decades ago because the widely separated convectively cooled heat sources operated at relatively low temperatures.

4.8.2 Modern electronic equipment is rated and specified in terms of ambient temperatures. This is adequate only for equipment which rejects heat into the surrounding fluid by convection. Thermal environment ratings are preferred because they include all convective, conductive, and radiative heat transfer and define the thermal condition of the surrounding fluid and mounting hardware. Thermal environment is defined by the following conditions: (1) cooling fluid type, temperature, pressure, and velocity; (2) equipment surface temperature, configurations, and emissivities; and, (3) all conductive thermal paths surrounding the electronic part or equipment. Item (1) includes ambient temperature. In addition, particularly with airborne equipment, it is desirable to specify the thermal characteristics of the coolant which is provided specifically for cooling the equipment.

4.9 Methods for minimizing heat dissipation. The electrical efficiency of many electronic circuits is very low, ranging from almost zero up to just a few percent. Careful preliminary or "bread-board" design will in some instances increase the efficiency and help alleviate heat removal problems by reducing the dissipated power. Each circuit should be analyzed during its early development and redesigned, if necessary, to obtain the highest practical efficiency prior to initiating the thermal design. It is necessary to examine each electronic stage individually for unnecessary power dissipation during both standby and full-output conditions. Analysis will provide an indication of where the greatest amount of power is being wasted and which of several efficiency improvement procedures should be used.

4.9.1 Semiconductor devices. The power dissipated by semiconductor devices can in certain instances be significantly reduced through the application of the so-called transconductance efficiency factor. Reference 39 describes this procedure in detail. It is analogous to the starved tube circuits which were developed in the late 1940's. Basically, it has been found that large reductions of the voltages to bipolar transistors result in equivalent or enhanced performance and reliability and provide major reductions in power dissipation.

In some applications it has been found that a maximum transconductance efficiency is the proper goal and in others a comparatively low efficiency leads to the best results. High transconductance efficiency can produce other advantages such as reduction of noise through minimization of current. The point is made in Reference 39 that most transistor RF and IF amplifiers in receivers operate at reduced gain due to difficulties in impedance matching and would likely oscillate if full gain potential were attained. Consequently, operation at reduced voltage and power dissipation (as much as 75%) will provide the same voltage gains in a practical circuit even though the potential gain may be somewhat reduced. This procedure is recommended particularly for discrete parts circuits. Integrated circuits are in part utilizing this approach, for the very reason that dissipation must be minimized.

4.9.2 Electron tubes. Small signal circuits using electron tubes when operated at reduced voltages, such as 100 volts or lower, on the anodes and screen grids will provide enhanced or equivalent performance to that obtained at much higher voltages. The power dissipations will, of course, be significantly reduced. An outstanding example is the unusually low tube failure rates achieved by the hybrid auto radios of the late 1950's that used 12 volts on the electron tube anodes and screen grids. Heater voltage reduction is not recommended.

4.10 Units and conversion factors. Heat transfer and the underlying disciplines of thermodynamics and fluid mechanics are very complicated subjects dealing basically with the movement of energy in space and time. Many of the equations have been derived by dimensional analysis, and are written as functional relations among dimensionless groups, the Reynolds, Nusselt, Grashof, and Prandtl Numbers, for example.

A dimension in the sense used here is a certain kind of quantity, such as mass, force, power, length, temperature, etc. A unit is the size or amount in terms of which any particular dimension is measured. Quantities are expressed in many different units, depending on the source of the data. For example, grams, ounces, and pounds are units of mass; dynes and newtons are units of force; meters, feet, and miles are units of length.

There are three requirements for accuracy of an equation. First, it must of course truly express the functional relation among physical quantities. Second, it must balance dimensionally so that each side of it expresses the same relation among dimensions. Third, it must balance quantitatively so that each side is expressed in the same unit. If the third condition is satisfied the second condition is also satisfied. However, the second condition is useful as a quick check of the truth of an equation.

It is possible and convenient to express quantities in terms of a set of fundamental dimensions. Various sets may be chosen. The most commonly used set is that of Mass M, length L, time t, and temperature T. Appendix A, Table of Symbols, gives the dimensional expression for each quantity used in this manual.

The Hagen-Poiseuille equation for pressure drop due to laminar flow in a pipe is used to illustrate methods of checking an equation:

$$p = \frac{32\mu \ VL}{D^2}$$

where μ = dynamic viscosity
V = average fluid velocity
L = pipe length
D = pipe diameter
32 = a dimensionless constant

Referring to the Table of Symbols the corresponding dimensional equation is:

$$M/Lt^2 \stackrel{\frown}{=} (M/Lt)\ (L/t)\ (L)\ (1/L^2)$$

where the symbol $\stackrel{\frown}{=}$ is used instead of =, to signify dimensional and not numerical equality. Since this reduces to 0 $\stackrel{\frown}{=}$ 0, the equation is dimensionally correct.

The equation must also be true quantitatively. Referring to Appendix A, Table of Units and Conversion Factors, the equation of units is:

$$\frac{\text{lbf}}{\text{sq.in.}} = 32\mu\ \frac{\text{in.}}{\text{sec.}}\ \ \frac{\text{in.}}{\text{sq.in.}}$$

where lbf = pounds of force
V = in in./sec.
L and D are in inches

It is clear that a correct result is obtained if viscosity is known in lbf - sec./sq.in. This is a logical unit since the concept of viscosity involves a shear stress. Some tables of physical properties do use this combination of units for viscosity, but other units are also found. Two frequently used measures of viscosity are centipoises, and pounds of mass per hour-foot. In thus, becomes necessary to convert from one system of units to another.

Conversion factors for units are very important, and particularly annoying, in heat transfer calculations. The Table of Units and Conversion Factors, Appendix A, gives most of the conversions which are necessary. One of the most comprehensive and convenient conversion tables is that in the Handbook of Chemistry and Physics, 49th edition or later, published by the Chemical Rubber Publishing Company in Cleveland, Ohio.

A serious conversion problem arises from the custom of using the word pound as the name of units for both mass and force. It therefore, is necessary to distinguish between pounds of mass (lbm) and pounds of force (lbf). The lbm is a legally defined standard unit for quantity of matter. Any unit of force must be defined by Newton's Second Law, force = mass x acceleration. The poundal is a force unit defined as that which accelerates one pound of mass at the rate of one foot per second. Unfortunately, this unit is seldom used in American engineering practice, which prefers the pound of force as a unit. The lbf is defined as that force which accelerates one lbm at the standard terrestrial gravitation rate g=32.17 ft. per second squared (to four decimal places). Note carefully, however, that lbm and lbf are different kinds of quantities and there is no equality between them.

The correct equality is written:

$$1 \text{ lbf} = 1 \text{ lbm} \times 32.17 \text{ ft./sec.}^2$$
$$= 32.17 \text{ lbm-ft./sec.}^2$$
$$1 \text{ lbm} = 0.0312 \text{ lbf-sec.}^2\text{/ft.}$$

When an equation is checked dimensionally, an unbalance of L/t^2 sometimes appears. This usually means that g must be used as a conversion factor.

It must also be remembered that 32.2 ft. per second squared is 32.2 x $(3600)^2$ or 416 million ft. per hour squared! Many a ridiculous answer is obtained by expressing g in the wrong units.

Confusion also sometimes occurs with regard to mass and weight density. One pound of mass weighs one pound, as weighed on an equal arm balance. A spring balance is in error by the difference between local gravity and that at the place where it was calibrated. The widest variation on earth is less than one part in 2,000.

This discussion will be clarified by the following example: Calculate the pressure drop in 100 feet of 6 in. diamter pipe for water at 50°F at an average velocity of 50 ft./sec.

$$p = \frac{32\mu\ (50)\ (100)}{(0.5)\ (0.5)}$$
$$= 64 \times 10^4\ \mu\ \frac{(\text{ft}^2\quad)}{(\text{sec.}\ \text{ft.}^2)}$$

at 50°F, μ for water is 2.73 x 10^{-5} lbf-sec./ft.^{2}

$$p = (64 \times 10^4)\ (2.73 \times 10^{-5})$$
$$= 17.5 \text{ lbf/ft.}^2$$
$$= 0.121 \text{ psi}$$

Now suppose that in the available table the viscosity is given as 3.17 lbm/hr.-ft. Converting to lbm/sec.-ft.

$$\mu = \frac{3.17}{(3600)} = 8.81 \times 10^{-4} \text{ lbm/sec.-ft.}$$
$$p = (64 \times 10^4)\ (8.81 \times 10^{-4})$$
$$= 565 \text{ lbm/ft.-sec.}^2$$

This equation is correct but the units in which the pressure is expressed are so unusual as to be practically meaningless. However, by using the conversion factor lbm/lbf the familiar units are obtained.

$$p = (565)\ (0.0312) \frac{\text{lbf}}{\text{ft.-sec.}^2}\ \frac{\text{sec.}^2}{\text{ft.}}$$
$$= 17.5 \text{ lbf/ft.}^2$$
$$= 0.121 \text{ psi}$$

This result agrees with the previous one.

In this handbook the fundamental units in which all quantities are defined are: the mass pound, inch, second, and degree celsius. Certain other units are also used because they are convenient, or because they are in common usage. Among these units are: watts for power and heat flow rate; pounds of force per square inch (psi) for pressure; cubic feet per minute (cfm) for gas flow rate, and gallons per minute (gpm) for liquid flow rate. The table of Units and Conversion Factors includes these units.

This handbook makes extensive use of electric circuit analogues to represent heat transfer systems, since electronic engineers find this device realistic and understandable. In these thermal circuits, thermal resistances are used rather than conductances. Thermal capacitances are also used to represent the heat capacities of physical parts; these are of importance when non-steady state conditions prevail.

The unit of thermal resistance is degrees celsius per watt, (°C/w). The unit of thermal capacitance is joules per degree celsius (w-sec./°C).

5. DETERMINATION OF THERMAL REQUIREMENTS

5.1 General considerations. The environmental and other aspects to be considered during the thermal design of Naval electronics equipments cannot always be clearly defined in the specifications. The following includes real world considerations which must be resolved in order to provide satisfactory equipment.

5.1.1 Provision must be included in the thermal design for adequate cooling during maintenance and test periods. For example, adequate cooling must be provided when drawers or chassis are withdrawn from cabinets, when chassis are bench tested, when the sides of cabinets are removed, or when avionics equipment is serviced in a hanger or on a flight deck. The lack of or inadequacy of cooling during maintenance periods under conditions such as the aforementioned has led to serious premature failures due to thermal overstress, even though the time periods were relatively short. Thus, the thermal design and the design for maintainability must be integrated.

5.1.2 The thermal design must include provisions for at least marginal cooling under certain emergency or battle conditions. Vital equipments must function even though portions of the cooling system may be damaged or inoperative. For example, in closed loop forced air to fresh water shipboard cooling systems, when the water system is inoperative, provision must be maae for the cirulcation of room or cabin air through the equipment, using the existing blowers or fans in conjunction with emergency air inlet and exhaust openings in the cabinet.

5.1.3 The thermal design must also include provision for rapid and ready maintenance of the cooling system.

5.1.3.1 Easy access to air and other coolant filters is extremely important. Equipment has overheated too often as a result of dirty air filters and blower scrolls. For example, on a certain ship, it was found that 65% of the filters were plugged solid and unchanged for several years because maintenance was difficult.

5.1.3.2 Heat exchangers, especially forced air to fresh water shipboard exchangers, must be readily accessible for removal and cleaning. The air side and liquid side of heat exchangers can become blocked with foreign material where passages are too small in crossectional area. Even though the liquid or water systems are provided with purification systems such as deionizers and filters, the buildup of fungus, scale, and other deposits accumulates at an astounding rate. The crossectional area of water passages on shipboard heat exchangers should be equivalent to that of a 0.375 inch inside diameter tube. The so-called compact heat exchanger should not be used unless it is truly warranted.

5.1.4 The condensation of moisture within equipment (on, for example, circuit cards and component parts, integrated circuit leads, connectors, and sockets with small pin spacings, etc.) can cause many serious problems including short circuits, electrolysis and subsequent conductor separation, circuit malfunctions and serious corrosion. Operation of equipment at temperatures below the dew point of high humidity air on shipboard is to be avoided. This is one of the major reasons for increasing the temperature of fresh water on shipboard from the 95°F previously specified to 40°C (105°F). The design of cooling systems using chilled water at lower temperatures should be approached with caution.

In aircraft, due to the wide variation in altitude, temperature, and humidity encountered during a mission, direct forced air cooled equipment can become moisture laden. This is a serious problem unique to avionics equipment. An excellent approach, now used in recently developed avionics, is to isolate the cooling air from the electronics; that is to use indirect forced air cooling techniques with internal air heat exchangers or cold plate type heat exchangers.

5.1.5 The installation, location and environment of electronic equipment can seriously influence the thermal performance. Well designed equipments have overheated because cooling air intakes have been located adjacent to the hot air exhausts of other equipments. Second handed cooling air is unsatisfactory for cooling. Equipments have been installed in small poorly ventilated compartments also containing large steam pipes. Equipments have been installed next to ships funnels and stacks.

When a number of relatively small individual equipments are mounted in a large cabinet, supplemental cooling means must be provided, since individual equipments are usually designed for cooling on all external surfaces. The air must circulate freely around all air cooled equipments. Also, the space or room in which a multiplicity of air cooled equipments are to be mounted must be adequately ventilated or air conditioned.

5.1.6 The cooling systems of each equipments must be checked for proper operation prior to installation. Thermal problems have been traced to fan, blower or pump motors running backwards. Wads of paper have been found in air ducts and coolant lines have been found plugged when the equipment was delivered. The thermal design was adequate, but the implementation was unsatisfactory. This points out the need for adequate inspection and test of each cooling system.

5.1.7 The supply of coolant to electronic equipments if for short periods often much less than specified. On ships certain combat situations require shut down of the air conditioning and ventilation systems (see section 5.1.2). In addition, in aircraft, for example, the flow of jet engine bleed air used for cooling can be drastically reduced during a high speed dash and the thermal problem is compounded by the increased aerodynamic heating of the aircraft. In some aircraft, the flow rate and temperature of the available cooling air matches that specified as being available for equipment cooling under ideal flight conditions only.

5.1.8 Solar radiation on electronic equipment, equipment shelters, vans, and aircraft can cause severe thermal problems especially in desert areas and equatorial or semi-equatorial areas such as Southeast Asia. Shiny aluminum aircraft parked in the sun can achieve internal cabin temperatures of the order of 74°C (165°F), which is considerably higher than the 50 or 55°C maximum equipment's design temperature. Upon equipment startup, the cooling air and equipment temperatures are too high for an appreciable period, until the aircraft cooling system or ram air cooling system stabilizes thermally. Similar conditions occur in vans, shelters, and mobile equipment. One solution has been to park the aircraft or van in the shade provided by a simple cloth awning a few feet above the vehicles. Alternatively, with aircraft, the equipment is not energized prior to takeoff.

5.1.9 On ships and aircraft, fuel oil particles which seem to be always present in the air, accumulate together with dust, lint particles, and moisture as a dirty deposit on surfaces of electronic equipment. This has been labeled "FLUG" and is particularly obnoxious in forced air cooled equipment. Blower scrolls, fan blades, filters, and all electronic parts in the air stream become coated in time. The result is that the effective thermal resistance is substantially increased unitl equipment major maintenance is performed (perhaps many months later). In the meanwhile, parts operating temperature are increased as is the probability of equipment failure. Therefore, it is suggested that the thermal design provide for the effect of FLUG and other degradation effects. The thermal design should be such that equipment which has been in service for some years is still adequately cooled.

5.1.10 Thermal transients occur during equipment warmup, as equipment duty cycle and power dissipation change, and as thermal environment and coolants change temperature. Transients tend to be smoothed out since equipment usually has a relatively long thermal time constant. As a rough rule of thumb, equipments which are properly designed thermally will have a warmup time to thermal equilibrium of the order of 5 to 10 minutes. However, parts temperatures will vary, particularly due to transient environmental effects, as in aircraft. Parts temperature cycling in excess of $\pm$ 20°C has been found to have a serious effect on reliability and it is desirable that these thermal excursions be minimized. Temperature control and regulation is thus warranted in special situations.

5.1.11 The thermal design must include consideration of the above factors which should be an item in the requirements section of any equipment contract. For short periods the thermal environment is often more severe than specified and as a minimum the equipment should be designed to meet the required reliability under the maximum specified thermal environment, not some average environment.

5.2 Purpose. The purpose of the following subsections is to outline stress analysis procedures for determining specific maximum parts temperatures that form the goals of thermal design. If the design is such that these maximum temperatures are not exceeded under the maximum thermal environment specified for a given equipment then the design is acceptable. These maximum parts temperatures must be determined in terms of the total stresses that each part is subjected to and must be consistent with the failure rate assigned to each part. If the maximum parts temperatures are incorrect, it follows that any thermal design based on these values is also incorrect.

5.3 Relationship between parts stress and reliability.

5.3.1 System or operational reliability is usually measured by the probability that an equipment will perform its specified function under specified conditions for a specified time. Inherent equipment reliability depends on equipment design. The important factors are selection and application of circuits and component parts, the thermal, electrical and mechanical stress levels of the parts, the cyclic variation of these stresses, the strength of the mechanical structure, manufacturing techniques, maintainability and quality of workmanship including inspection procedures. These effects are amenable to calculations and can, when the necessary data are available, be predicted and/or determined with reasonable accuracy from a broad spectrum of failure rate data, failure reports, and statistical knowledge of production and inspection procedures.

5.3.2 Effects of heat on parts failure rates. The failure rates of parts in general increase with loading or stress level, whether it be thermal, electrical, or mechanical. Stresses below the intensity which causes catastrophic failure result in progressive deterioration of material. The effect of temperature cycling is believed to be extremely significant, although little knowledge of the effects exists.

Thermal aspects of equipment design are discussed in section 2.0.e. of MIL-HDBK-217. Also, MIL-HDBK-217 states that the part failure rates are additive in computing the circuit failure rate and the cumulative effect of many minor thermal improvements will equal that of a few major ones. Thus, inattention to a majority of minor points in thermal design may compromise good treatment of a minority of major points. It can be concluded, that for a given circuit, reliability is almost entirely at the mercy of thermal environment. Fortunately, it is possible to control the temperatures of the parts at sufficiently low levels to achieve reasonable failure rates, whatever the severity of the thermal environment in which the equipment operates.

As previously stated, high part temperatures result in performance degradation and in accelerated failure. While the details of this deterioration vary from one part type to another, it can be concluded that elevated temperatures always threaten equipment reliability. Equipments which function adequately at one temperature level will be less reliable or inoperable as temperatures increase. A quantitative idea of this inter-dependence can be gleaned from the part failure characteristics presented in the MIL-HDBK-217. Naturally, a similar though composite relationship holds for the circuits in which these parts are used. These data make it clear that reliability is not solely a matter of avoiding the excessive temperatures to which the failure rates are asymptotic. They illustrate that reduction in temperature at any range down to about 20°C is reflected by significantly lower failure rates.

In order to achieve present and new reliability goals it is therefore, necessary to reduce parts temperatures to the lowest practically attainable levels.

Thermal failure of electronic parts is caused by deterioration, due to high temperature, of the materials of which the part is made. An old rule of chemistry states that the speed of chemical reactions doubles for every 10°C increase in temperature. Parts failures rates are known to increase exponentially with temperature as evidenced by published data. A thermal failure may occur so rapidly as to be considered catastrophic. However, there is always a slow, progressive deterioration of dielectrics, cathode coatings, transistor junctions and many other materials which accelerates with temperature, leading eventually to failure. These effects are cumulative so that failure rate depends to some extent on the entire thermal history, the temperature-time integral. Thermal failure is, therefore insidious since it is usually impossible to determine the percentage of life remaining in a part. This has a direct bearing on the effects of temperature cycling, which is specified in nearly all specifications for testing electronic parts and equipments, and which occurs during the normal operation of equipment. There are indications that temperature cycling has very adverse effect on reliability but there exist little quantitive data and no adequate theory by which the effect can be accurately estimated.

The true thermal stress is the temperature of the material of which the part is made. Since this is internal to the part, it is generally impossible to measure. The temperature of the accessible outer surface is the most practical index of the thermal condition of the part. Surface or body temperature is a function of the heat dissipation within the part and of its thermal environment, which is defined as a complex of: (1) coolant type, temperature, pressure and velocity, (2) the configurations, emissivities and temperatures of neighboring surfaces, and (3) all conductive heat flow paths surrounding the part. This becomes evident from the Figure 1, which shows an electrical analog of the thermal system of a typical part.

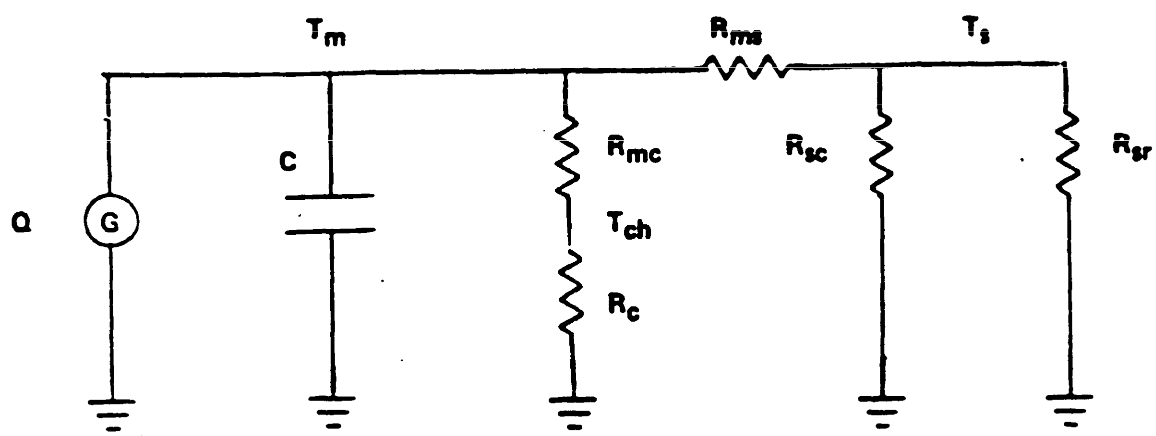


FIGURE 1. Equivalent Thermal Circuit of a Part

- G = constant current generator with Q internal heat dissipation
- C = thermal capacity of the part
- R_{mc} = thermal resistance, material to chassis
- R_c = thermal resistance, chassis to heat sink
- R_{ms} = thermal resistance, heat source to surface
- R_{sc} = thermal resistance due to convective cooling
- R_{sr} = thermal resistance due to radiation
- T_m = temperature of the material
- T_{ch} = temperature of the chassis
- T_s = temperature of the surface
- T_e = environmental temperature

5.4 Cost benefits of thermal reliability. One of the most important questions is that of economics. How much will it cost overall for adequate thermal design during equipment development or thermal improvement of existing equipment and is it economically worthwhile? What are the availability gains?

As a valid example, under Navy sponsorship, a comprehensive and conservative cost analysis study (not cost effectiveness) was made pertinent to three shipboard equipments. (Reference 2) Updated costs were used to include inflationary effects, interest costs, detailed design and development costs for improved thermal systems and maintenance and logistics costs. All cost data were real, since field costs were available for quantities of equipments which had been in service for some years and the thermal modifications had actually been made.

It was determined that the cost of modifying small (100 watt) and medium size (500 watt) equipments could be fully paid for in four to six months of operation, based on maintenance cost savings alone. For a large high power radar, improvements could be paid for in about nine months of service, again based only on maintenance costs savings. The gains in availability and probability of mission success, which are priceless and impossible to assess economically, were excluded and in this sense were free. Further, the added incremental cost of providing an adequate thermal design during the initial design and procurement of these equipments was calculated. It was less than the cost of later modifications.

Thus, it was concluded that the Navy would obtain significant economic benefits in life cycle costs by investing in (1) thermal improvement R&D, (2) equipment procurement specifications that mandated adequate thermal designs, (3) dissemination of thermal design information, (4) development of thermally adequate installation environments, and (5) improved thermal maintenance procedures.

These gains were attained under severe handicaps namely:

a. The equipments modified and evaluated were old used equipments and the parts had a previous history of severe thermal overstress.
b. The thermal design of the improvements was not optimum. Major modifications could not be made.

Therefore, the initial added investment in adequately cooling electronic equipment pays off significantly in improved reliability and reduced life cycle costs of equipment.

5.5 Determination of the maximum parts temperature

5.5.1 A reliability model must be constructed in accordance with the procedures given in the reliability section of the applicable contract specifications. MIL-HDBK-217 presents typical procedures for this purpose. This model must include a part by part breakdown of the functional circuit with detailed failure rates assigned to each part. The summation of the failure rates is the predicted equipment reliability, usually measured as the MTBF.

5.5.2 The voltage, current, power dissipation, and any other factors pertinent to the electrical stress imposed on a part must be determined for each part. From this a stress ratio must also be determined for each part as outlined in, for example, MIL-HDBK-217. Further, for some parts, a K factor, which is related to the parts application, for example, airborne, or shipboard, must be applied to the stress ratio.

5.5.3 Examine the temperature vs failure rate curves in MIL-HDBK-217. FARADA data (Failure Rate Data Handbook - Tri Service and NASA) and RADC Reliability Handbook Vols. I and II should be used as secondary sources when not covered by MIL-HDBK-217. Based on the K factor, the electrical stress and the required failure rate, ascertain the maximum part temperature. The temperature should be given as a surface temperature. If it is given as an ambient temperature, convert to surface temperature. Chapter 6 presents conversion equations for resistors. The RADC Reliability Notebooks include ambient to surface temperature conversion equations for most ambient temperature rated parts. The equations are included with the data for each type of part.

5.5.4 The maximum temperatures must be established for each part in accordance with the above procedure. The designer should examine the temperature vs failure rate data for each part as given in MIL-HDBK-217. The maximum temperatures will often be lower than those with which designers typically work, especially when the electrical stresses are high. However, these are the temperatures which must be maintained as minimum under the maximum specified environmental conditions for the equipment.

6. THERMAL DESIGN REQUIREMENTS

6.1 Electrical analogy of heat flow. In this handbook, the electrical analogy of heat flow is used throughout in the thermal design calculations, including computer solutions. This method has been widely used successfully and is recommended. Further, electronic engineers can readily understand and solve a thermal design problem which is expressed as an electrical circuit network.

6.1.1 Analogies. The analogy is: Heat flux (or power dissipated) is analogous to electric current. Temperature (or temperature difference) is analogous to voltage. Thermal resistance to heat flow is analogous to electrical resistance. (Similarly, thermal conductance is analogous to electrical conductance). The unit of thermal resistance is degrees C per watt.

The equations and details of natural heat transfer are presented in the theory section of chapter 8. In brief, the above analogies are particularly applicable to heat transfer by conduction and are applicable also to other modes of heat transfer when their nonlinearities are considered.

Thermal capacitance is also analogous to electrical capacitance.

In the thermal circuit:

$$q\,dt = mc_p\,dT \qquad (6\text{-}1)$$

$$q = mc_p\frac{dT}{dt} \qquad (6\text{-}2)$$

This is analogous to the electric circuit equation:

$$i = C\,\frac{dE}{dt} \qquad (6\text{-}3)$$

Therefore, thermal capacitance is:

$$c_{th} = mc_p = \rho v c_p \qquad (6\text{-}4)$$

There is no thermal equivalent of inductance since heat energy is not stored by virtue of heat flow rate. Heat has no inertia. The partial differential equations of heat flow are thus diffusion, rather than wave equations. They contain only first derivatives with respect to time.

The heat capacity of component parts and structures is of interest in transient heat transfer analysis only. Thermal capacities can be represented as capacitors connected between temperature nodes and ground in the equivalent thermal circuit.

6.1.2 Electro-thermal analog networks. The concept of an electro-thermal equivalent circuit can be extended as a computational tool for both steady state and transient heat flow problems. Temperature can be considered as a potential that causes heat flow. Constant temperature heat sources are equivalent to ideal voltage sources. Under steady state conditions, the heat dissipation of electronic equipments is constant. These constant heat flow sources are equivalent to ideal current sources. Regions of conductive, convective, and radiative heat flow may all be treated as thermal resistances. Since heat flux is analogous to electric current, masses of material which absorb or reject heat with change of temperature may be treated as thermal capacitances. The ultimate heat sink is equivalent to "ground" or "earth," to which all heat sources and thermal circuits are connected.

All of the mathematical methods of electric circuit analysis and synthesis can be applied to these equivalent thermal circuits. These methods range from simple arithmetical Kirkhoff's Law calculations, through one-dimensional transmission line and multi-dimensional field theory, integral transform and linear graph methods, to difference equations and digital computer programs. If temperature and/or heat flux methods are applied to cooling problems, the methods of control engineering can be used.

As previously mentioned, heat conduction paths are directly analogous to electric current conduction paths. Convection heat flow can be represented by thermal resistance, since:

$$q = Ah_c \Delta T \qquad (6\text{-}5)$$

$$R_c = \frac{\Delta T}{q} = \frac{1}{Ah_c} \qquad (6\text{-}6)$$

The analogy is more difficult to apply to convection than to conduction because the convective heat transfer coefficient is a nonlinear function of ΔT, and varies much more with temperature than does thermal resistivity. However, approximate values of the temperatures are usually known, and a successive approximation method of two or three steps will yield a value of h_c within $\pm$ 5%, which is about as good as the thermal data applicable to most electronic equipment cooling problems.

Radiation introduces a wider uncertainty since radiant heat exchange varies with the difference in the fourth power of the two temperatures.

6.1.3 The ultimate sink. Thermal equivalent circuits are commonly drawn as "open circuits," just as is customary with electronic circuits. Obviously, if current or heat is to flow, the circuit must be closed. Heat sources are equivalent to current generators, connected between the passive circuit and the heat sink, which is equivalent to ground.

The heat generated in electronic equipment is transferred by various modes through various channels and ultimately reaches a heat sink. The cooling process is simply one of transporting thermal energy from a heat source to a heat sink at a lower temperature. The design problem is to provide a thermal path of sufficiently low resistance and a heat sink at a sufficiently low temperature to maintain all nodal temperatures at the required levels.

The term "ultimate sink" signifies a heat sink so large that its temperature does not vary with the quantity of heat energy dumped into it. It is an infinite thermal capacitor. This ultimate sink may be the earth, the atmosphere, a large body of water such as a lake or ocean, or space, depending on the location of the equipment being cooled.

However, from the practical view, the electronic designer will have available and will make use of intermediate sinks. In transient heat flow problems every mass of material present will act as an intermediate sink or a thermal capacitor. Such sinks have finite heat capacity. Thus, in steady state heat transfer, heat must be removed from them at the same rate as it is supplied. These intermediate sinks are generally those local to the heat sources and can include chassis structures, the case or cabinet housing the equipment, a cold plate, a finned heat sink or a coolant such as the local air or a liquid.

6.2 Determination of required thermal resistances

6.2.1 Determine the maximum safe parts temperature consistent with the required reliability for each part as outlined in chapter 5.

6.2.2 Ascertain the maximum environmental temperature specified for the equipment or the coolant.

6.2.3 Determine the allowable temperature rise for each part. (6.2.1 minus 6.2.2). (If the 6.2.2 temperature is greater than that of 6.2.1, then refrigeration is required).

6.2.4 The thermal resistance required to adequately cool each part is equal to the allowable temperature rise divided by the heat dissipation.

The procedures can be best illustrated by the following elementary example:

Example 6-1

Determine the thermal resistances required for adequately cooling a small metal cased subassembly which is to be mounted on a chassis having a local sink temperature of 65°C. The following parts are to be cooled:

(1) Four (4) power transistors each dissipating three (3) watts with maximum safe junction temperatures of 150°C.
(2) Three (3) IC's each dissipating one (1) watt with maximum safe junction temperatures of 140°C.
(3) One (1) resistor dissipating one half (1/2) watt with a maximum safe surface temperature of 95°C.
(4) Two (2) resistors dissipating one quarter (1/4) watt having maximum safe surface temperatures of 105°C.

Then, the total thermal resistances required are:

(1) For the power transistors:
$$\frac{150-65}{3} = \frac{85}{3} = 28.3°\text{C/watt}$$

(2) For the IC's:
$$\frac{140-65}{1} = 75°\text{C/watt}$$

(3) For the resistor:
$$\frac{95-65}{1/2} = \frac{30}{1/2} = 60°\text{C/watt}$$

(4) For the resistors:

$$\frac{105-65}{1/4} = \frac{40}{1/4} = 160°C/w$$

6.2.5 Thermal analog circuit. Prepare a thermal analog circuit of the heat flow paths. The procedure is demonstrated by continuing with the previous example. The thermal analog circuit is shown in Figure 2.

The total thermal resistances in the design must not exceed those given in Figure 2. These are the thermal design requirements.

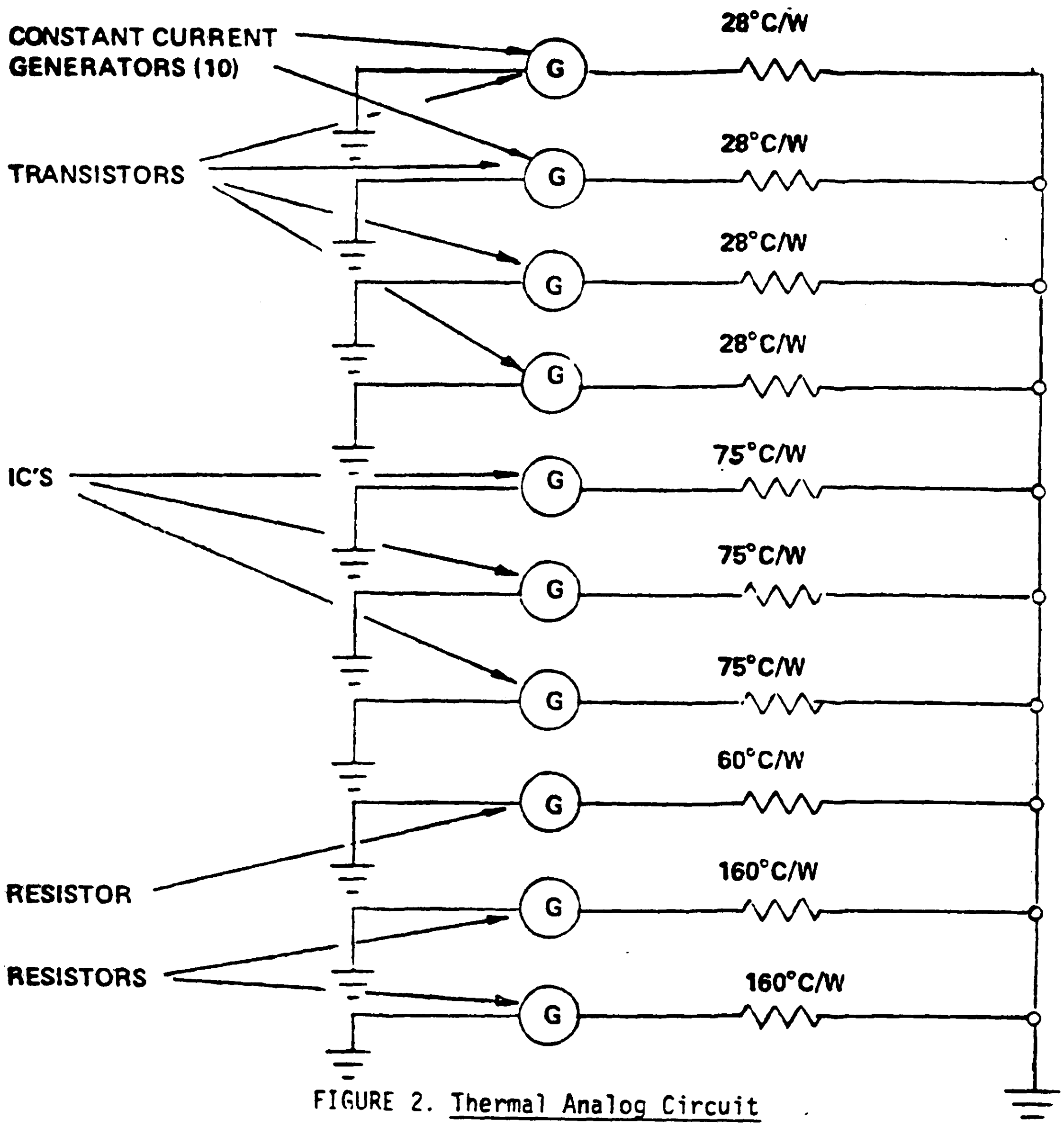


FIGURE 2. Thermal Analog Circuit

6.2.6 <u>Combining thermal resistances</u>. The resistances should be combined where common heat flow paths or ratings exist. The procedure is illustrated by continuing the previous example.

The resistances of Example 6-1 can be combined since the power transistors all operate at the same thermal ratings. Thus, they will be mounted on another metal plate, and the resistors will be supported by their leads. The combined thermal analog circuit is shown in Figure 3.

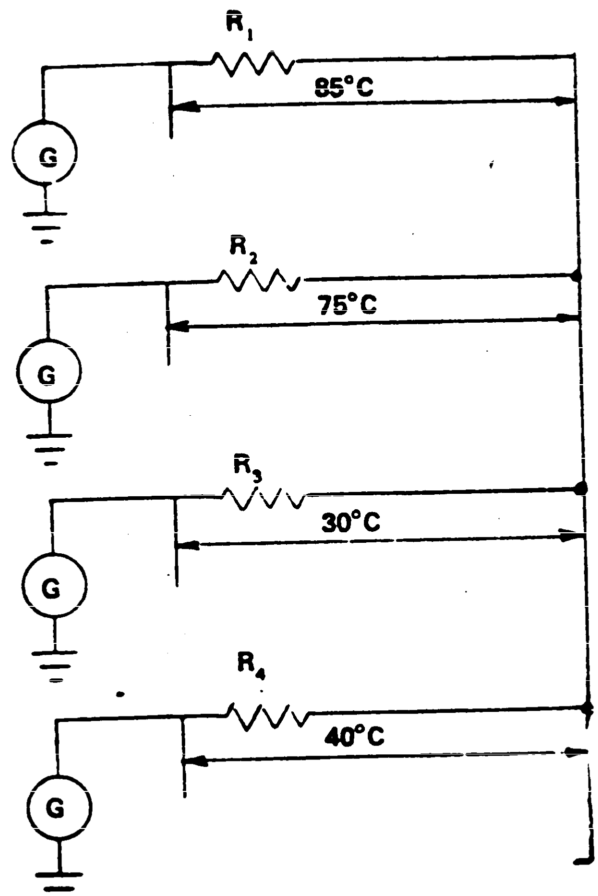


FIGURE 3. Thermal Analog Circuit

The recombined thermal resistances are:

$$R_1 = \frac{28.3}{4} = 7.075°C/w \text{ (Say 7.1)}$$

$$R_2 = \frac{75}{3} = 25°C/w$$

$$R_3 = 60°C/w$$

$$R_4 = \frac{160}{2} = 80°C/w$$

Further combination is possible in this example since common heat flow paths exist. That is because the subassembly will be mounted on a chassis and the heat flow from all parts has a common path through the mounting surface interface with the chassis and through the common metallic hardware.

The thermal analog circuit of Example 6-1 is further combined and the thermal resistances are subdivided to show the detailed thermal resistances as shown in Figure 4.

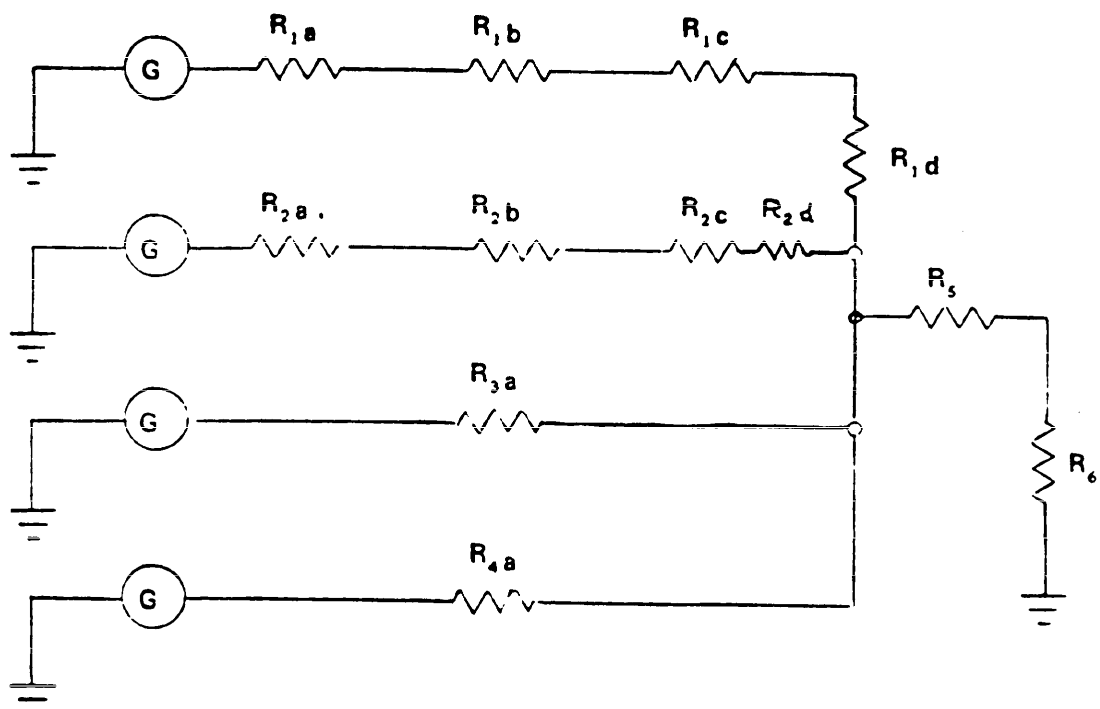


FIGURE 4. Thermal Analog Circuit

Where:

R_{1a} is the internal thermal resistance of the transistors from the collector junctions to the mounting surfaces.

R_{1b} is the interface mounting resistances of the transistors and includes the thermal resistances of the mica insulating washers.

R_{1c} is the thermal resistance of the metal transistor mounting plate.

R_{1d} is the interface thermal resistance between the transistor metal mounting plate and inner surface of the metal case of the subassembly.

R_5 is the thermal resistance through the base of the metal case.

R_6 is the thermal interface resistance between the outside surface of the metal case and the local heat sink at 65°C.

R_{2a} is the combination of internal thermal resistances to the transistors in the IC's.

R_{2b} is the mounting interface thermal resistance of the IC's.

R_{2c} is the thermal resistance of the IC metal mounting plate.

R_{2d} is interface thermal resistance between the IC mounting plate and the inner surface of the metal case.

R_{3a} and R_{4a} are the thermal resistances between the resistors and the inner surface of the case via lead conduction, convection, and radiation.

6.3 Tentative assignment of thermal resistance values

6.3.1 The procedure to be followed in tentatively assigning values to thermal resistances can be demonstrated by continuing design Example 6-1.

Examination of the analog circuit of Figure 4 shows that the total heat flow of (12+3+1/2+1/2)=16 watts is through R_5 and R_6. Obviously, these resistances must be small. Good judgement is required to assign low thermal resistance values which can be practically achieved without penalties such as large heavy metallic heat conductors or large surface areas and high contact pressures at the interfaces.

Examination of chapter 8 (Natural Cooling Methods) design procedures will show that a reasonable value for R_6 is 1.0°C/watt and for R_5 0.1°C/watt is appropriate. The internal thermal resistances of the transistors and IC's are given by their manufacturers as 2°C/watt and 30°C/watt. Thus, R_{1a} is fixed as 2/4=1/2°C/watt and R_{2a} is fixed as 30/3=10°C/watt. Also, the interface thermal resistance for the transistors is determined from the manufacturers data and chapter 8 as 1.2°C/watt for each transistor and thus, R_{1b} is 1.2/3=.4°C/watt. Similarily, typical IC interface resistances are 15°C/watt and R_{2b}=15/3=5°C/watt. These values are essentially fixed since they are a function of the surface condition, area, and pressure at the device interface.

Therefore, for the transistors R_{1a}, R_{1b}, R_5, and R_6 are established and have a total thermal resistance of 0.5+0.4+0.1+1.0=2.0°C/watt. Since a total thermal resistance of 7.1°C/watt is permitted in the total heat flow path, then 5.1°C/watt can be allocated to R_{1c} and R_{1d}. The joint thermal resistance R_{1d} will likely be about three times greater than R_6, because the interface area is smaller and the contact pressure is less in a joint using relatively thin metal. Thus, R_{1d} can be designed to be 3.0°C/watt and R_{1c} designed for 2.1°C/watt based on procedures given in chapter 5.

For the IC's - R_{2a}, R_{2b}, R_5, and R_6 are established to have a total resistance of 16.1°C/watt. The remaining thermal resistance of 25-16.1=8.9°C/w is available for R_{2c} and R_{2d}. This is a feasible value for a heat conducting metal strap and the dimensions can be calculated based on procedures and data given in chapter 8.

The half watt resistor requires a thermal resistance R_{3a} of 30 minus 1.1 or 28.9°C/watt. Calculations from chapter 8 show that the thermal resistances for natural convection plus radiation and lead conduction will provide only a thermal resistance of 55°C/watt. Therefore, a heat conducting clamp having a thermal resistance of 15°C/watt must be used to clamp the resistor to the case near the base. This allows 13.9°C/watt for the clamp interface and the heat flow through the case to R_5. These values are readily obtained using chapter 8 procedures.

The two one quarter watt resistors require a thermal resistance of 80-1.1 or 78.9°C/watt which is readily achieved by natural convection plus radiation and lead conduction per chapter 8 data.

The resultant temperatures can be calculated based on the heat flow, thermal resistance, and Ohm's law. The temperatures can be measured after construction of the first model to validate the thermal design.

6.3.2 Comments on assignment of thermal resistance values. Knowledge and experience will greatly aid in assigning tentative values and converging on the final values of thermal resistance. However, a few trial designs will rapidly indicate the desired direction and indicate "ball park" values. Often the cooling techniques initially selected cannot practically provide the required low thermal resistances. Other cooling techniques must then be considered. Chapter 4 presents information and data on cooling techniques selection based on thermal design requirements, unit heat concentration, and thermal resistances.

6.3.3 The procedure used in the electrical analog circuit for a large complex equipment is the same as that outlined above for the small assembly except that the network will have many thermal resistances. It is generally desirable to lump the thermal resistances internal and external to assemblies, intermediate sinks, chassis, drawers, or modules and evaluate each independently based on tentative resistance values in the common heat flow paths. With parallel heat flow paths the analysis is, of course, simplified. Additional information for the electro-thermal analog is presented in the design chapters, which permit the designer to design for a specific required thermal resistance using a particular cooling technique.

6.4 Computer assisted thermal analysis

6.4.1 General. A number of organizations have developed computer programs for the thermal analysis of electronic equipment. The programs range from small programs for the design of, for example, a heat sink with a single transistor, to large programs for the thermal analysis of the complete electronic system in an aircraft or satellite.

The use of computer assisted thermal analysis requires considerable effort and expense. This includes the development of the program (modeling, programming, and debugging), the calculation and/or measurement of all thermal parameters, validation, and running costs. Consequently, computer assisted analysis of routine thermal designs can be economically justified only by those groups engaged full time in thermal design. Under these conditions, the costs can be shared among the various design problems. However, complete electronic systems packages or those with transients usually require computer assisted analysis.

6.4.2 Description of typical programs. Under DOD sponsorship, several excellent large computer programs have been developed for the thermal analysis of electronic equipment. (Reference 34, 35, and 36) These programs are accurate and will provide excellent results.

The computer programs are based on the same electro-thermal analog procedure as outlined in this chapter. In brief, a complex thermal system is analyzed by designing a schematic of the electrical analog of the system and simulating it on a digital computer. The most laborous and tedious portion of the effort is the requirement for dividing the thermal system into units of homogenous "lumps" which in turn are nodes of an analogous electrical resistor-capacitor network. The success of the analogies depends largely upon the correspondence of this R-C network model to the real system. Numerous tradeoff decisions must be made at this point. Greater accuracy is obtained with finer lumping, (i.e., small and many lumps). However, this also increases the complexity of the network as well as the amount of storage and computation time required of the computer; the later two directly affect the cost of running the analysis on the computer.

The method of describing any thermal system in the generalized program provides for almost all contingencies such as (1) specification of any possible network layout, (2) setting all parameter values, both fixed and variable, (3) provision for functional inputs and active components, (4) provision for control systems as required for cooling and refrigeration equipment, (5) ease of changing some parameter values after a particular test to repeat the test without the necessity for completely redescribing the system, and (6) complete freedom of the investigator in selecting data and form of output.

The determination of values for all parameters is required before any analysis can commence. This also can be a very difficult and tedious task. The programs require large amount of memory reserved for each of several types of values such as voltages, resistances, capacitances, current sources, etc. Approximately 1200 heat sources can be considered simultaneously. Thus, a large amount of memory must be reserved for the many types of parameter values. Reference 35 lists a FORTRAN program for an IBM 7044-7094 computer and Reference 36 is a similar program revised to operate on an IBM 360-65 computer.

NOTE: Other available thermal analysis programs include NATA (Numerical Analysis Thermal Application-IBM, ESC, Owego, N.Y.). (For IBM 7090), (Chrysler Improved Numerical Differencing Analyzer - available from COSMIC), and NASTRAN (NSDRC).

7. THE SELECTION OF OPTIMUM COOLING METHODS

7.1 General. This chapter discusses the relative merits of the various cooling means and methods of determining the most practical cooling mode for a particular application. The figures of merit which have been assigned are the heat concentration in watts/cu. in. and the thermal resistance. In those instances wherein the external surface area of the item under consideration limits the thermal resistance, the unit heat dissipation in watts/sq. in. is also mentioned as a secondary figure of merit. These numbers can also be used as a measure of the magnitude of the design problem.

The selection of the optimum cooling method should be concurrent with the breadboard development of the electronic circuit. If the electronic performance is influenced by the cooling method, the circuit of the prototype model should be modified after the initial breadboard tests.

The selection of cooling methods discussed herein are primarily predicated upon the designer having freedom of cooling technique selection. Often the overall cooling techniques are specified. Even so, the designer usually has a choice of local cooling techniques in the heat flow path terminating in the specified technique. Factors such as the complexity of the equipment, space, power, thermal environment, available sinks, and cost must also be considered. Since the optimum method of heat removal within a subassembly and a unit may differ from that used to transfer the heat to the ultimate sink, each will be separately discussed.

7.2 Heat transfer within a unit

7.2.1 General. The method of heat removal from within an assembly must be such as to provide a low temperature gradient between the heat producing parts and the cooled surface or the local sink. The cooling method must be simple, lightweight, reliable, easily maintained, and economical. Further, it should occupy a minimum of volume, preferably utilizing the voids between densely packaged parts.

7.2.2 Natural methods. Natural cooling means are recommended for use within most miniaturized electronic assemblies. They are frequently the only possible means of heat removal. Hermetic sealing and the dense packaging of parts can prevent the use of other techniques.

Metallic conduction should be considered initially as the primary cooling means. Radiation cooling is not recommended as a primary means, since high temperature differences are required for appreciable heat transfer. Further, the control of the cooling path is lost since the heat will be radiated into nearby assemblies. Convection cooling requires large areas which are seldom available within assemblies. Also convection currents can transfer the heat into other locations which will only require additional cooling.

Plastic embedment may be used at heat concentrations of the order of 0.25 watts per cubic inch at environmental temperatures of the order of 85°C. Metallic conduction can be used for heat concentrations as great as 2 watts per cubic inch (See Figure 5). The maximum unit heat dissipation for free air cooled surfaces is usually 0.25 watts/sq. in. In a few high temperature devices, unit heat dissipations as high as 0.5 watts/sq. in. have been achieved.

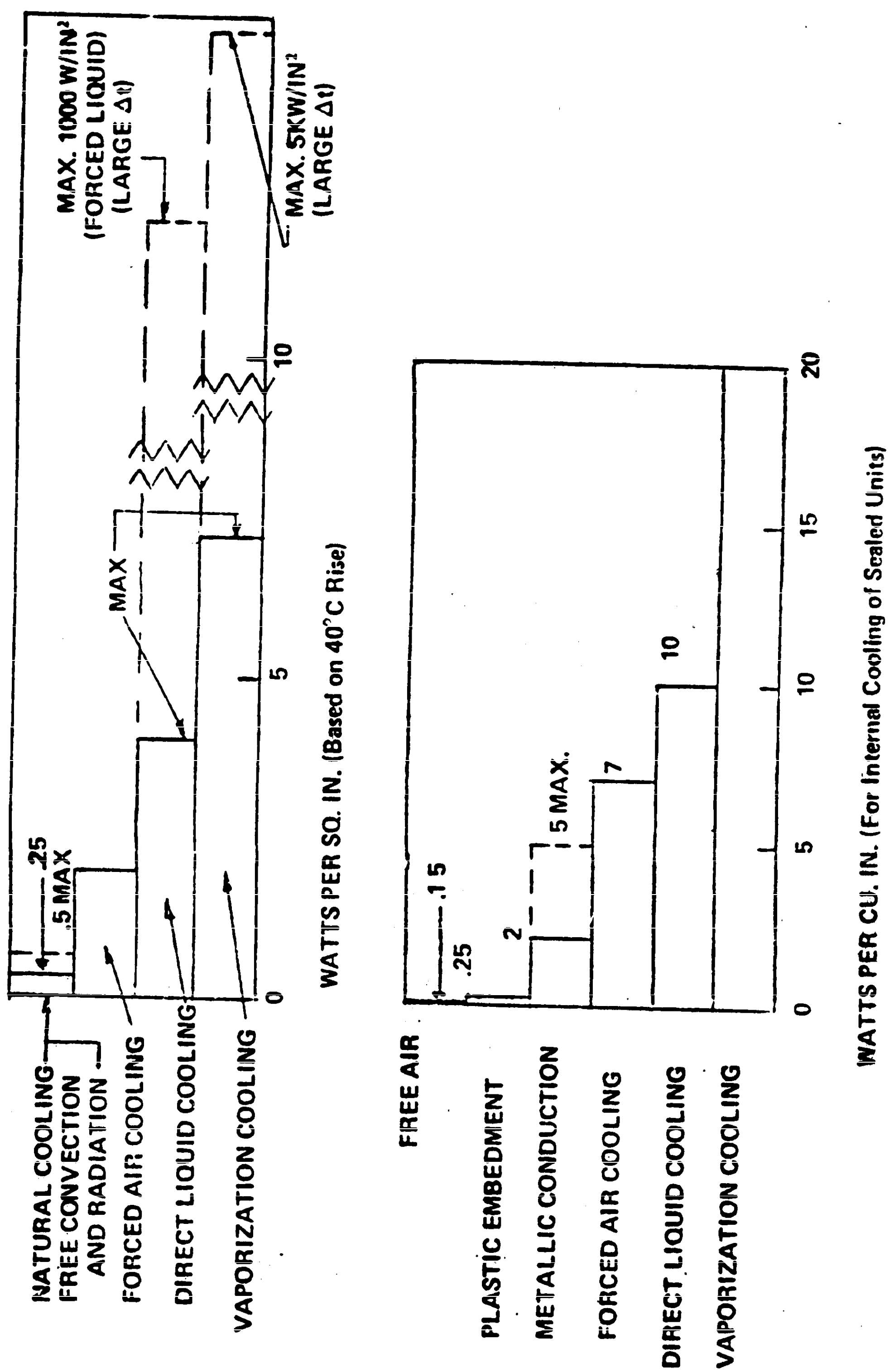


FIGURE 5. Comparison of Methods of Cooling

It should be noted that isolation from environmental effects is essential for many equipments which must operate under rigorous climate and environmental conditions. Overall package sealing simplifies the problem to the extent that only one large seal must be made and all parts are protected by it.

With conduction cooling, metal to metal contact interface resistances of the order of 0.25°C/watt/sq. in. and insulated contact resistances of the order 0.5°C/watt/sq. in. are achievable. Typical metallic heat conductors exhibit thermal conductivities of 6 (watts/sq. in.)/(°C/in.).

7.2.3 Forced air. Forced air cooling is an excellent cooling method, which can be used if the spacing between parts within the assembly is adequate for air flow or local heat sink fins are incorporated. Considerable heat can be removed by this method (See Figure 5.) However, the power required to force air over objects and through ducts and heat exchangers may be considerable.

7.2.4 Direct liquid cooling. This cooling means is particularly applicable to assemblies having high heat concentrations or those which must operate in high temperature environments with small temperature gradients between parts and cooled surfaces. Unfortunately, direct liquid cooling can be used only in circuits which can tolerate the increased stray capacitance and electrical losses due to the high dielectric constant and power factor of liquids. Typical thermal resistances are of the order of 8°C/watt/sq. in.

New equipments can be designed for several types of liquid cooling systems, any one of which may have cooling capacities greater than that of forced air systems (See Figure 5). The cases of sealed assemblies can be designed for direct immersion in the coolant (indirect liquid cooling) or the electronic assembly can be filled with a liquid such as a silicone fluid (direct liquid cooling). Cooling of directly immersed equipment may be significantly increased by forced circulation of the coolant. Thermal resistances as low as 0.2°C/watt/sq. in. are possible. However, this increased cooling is at the expense of power to operate the pump and the additional equipment such as a heat exchanger. The weight of directly immersed equipment may be reduced somewhat by spraying the coolant over the heat producing parts and collecting the heat bearing coolant in the bottom of the container and then pumping it through a heat exchanger and back to the spray nozzles. Such a cooling system represents a saving in the amount of coolant liquid required, but requires a higher pressure pump and consequently more power to run the pump than in the case of the completely immersed equipment.

Liquid cooling is most applicable to power supplies, modulators, servo amplifiers, and wide band low frequency amplifiers. It can also be used with radio frequency circuits, if consideration is given to the dielectric constant and dissipation factor of the fluid. The coolant must be chemically and electrically compatible with the electronic parts and the case. If liquid cooling is applied to equipments which operate over a wide range of environmental temperatures, care must be exercised in making sure that the coolant can not freeze at the lower temperatures.

Liquid cooling frequently permits a greater degree of miniaturization because of the larger permissible heat concentrations. Further, if a coolant with a high dielectric strength is used, voltage ratings can be increased. On the negative side, liquid cooling requires that the containers accommodate expansion at elevated temperatures. Unless the coolant is chemically inert, it may decompose the electronic parts. Also, maintenance difficulties are increased, and a leak may disable the unit. Repairing of direct liquid cooled equipment is complicated by the necessity for draining the fluid from the unit before working on it. Further, the fluid may be contaminated when the unit is unsealed unless extreme care is used.

7.2.5 Direct vaporization cooling. Vaporization cooling is the most effective heat removal method known. It has the advantages and disadvantages of direct liquid cooling together with greatly decreased thermal resistance. Thermal resistances as low as 0.04°C/watt/sq. in. are possible under maximum conditions (See Figure 5). Expendable systems are simple, but involve disposal of the vapor and replacement of the coolant. Non-expendable or continuous systems are complex, expensive, and necessitate the use of a heat exchanger to condense the vapor back into a fluid. Vaporization cooling systems are particularly suited to installations with extremely high heat concentrations and those installations wherein no sink is available or the sink if remotely located.

7.3 Heat transfer to the ultimate sink

7.3.1 General. The method of transfer of heat from the assembly or unit chassis to the sink is dependent upon the method of heat removal from within the assembly due to the common connection between the two phases of heat rejection. Further, the selection of the optimum method of heat transfer for use in this phase is dependent upon the type of sink available, its location, and its temperature. The sink temperatures, both before and after installation of the equipment, must be considered, since the temperature of local or intermediate sinks may increase when the additional heat is added.

7.3.2 Natural methods. Natural heat transfer from miniaturized assemblies to the intermediate sink can often be accomplished by metallic conduction cooling. In general, the reasoning discussed in the preceeding section is also applicable. However, the intermediate sink cannot be located at any significant distance from the subassemblies. Small temperature gradients are only obtained over appreciable distances with metallic conduction cooling when large heat conductors are used. The cost and weight of such conductors will probably be excessive. In certain instances the equipment may be thermally fastened to the adjacent structural parts.

Natural convection and radiation may be used at the sink if the sink is air of a relatively low temperature. The maximum heat dissipated by the surfaces should seldom exceed 0.25 watts per sq. in. and should be limited to approximately 0.50 watts per sq. in. Even so, relatively high temperatures can easily be achieved. It is therefore, recommended that this mode of cooling be used only with equipments of low heat concentration, provided that the rejected heat is not introduced into other nearby equipment. Supplemental data is presented in chapter 8.

7.3.3 Forced air. Forced air is more applicable to this phase of cooling than natural methods, particularly if the sink is nearby air. The air should be properly directed and distributed over the subassemblies or their finned heat sinks (heat exchangers). Unit heat dissipations of the order of 2 watts per sq. in. can be obtained readily. Supplemental data are presented in chapter 9.

7.3.4 Indirect liquid cooling. When electronic equipment is to be operated in high temperature environments at high heat concentrations or when the sink is located at a distance from the equipment, optimum cooling can be achieved by a forced liquid cooling system. Greatly increased cooling over that obtained by forced air is possible. Supplemental data are presented in chapter 10.

7.3.5 Indirect vaporization cooling. This mode of heat transfer will provide the maximum obtainable cooling. It is recommended for use only with devices having extremely high heat concentrations. Supplemental data are presented in chapter 11.

7.3.6 Thermoelectric cooling

7.3.6.1 Refrigeration techniques. Details on this cooling technique are given in chapter 12. In brief, thermoelectric cooling is a refrigeration technique which provides a negative thermal resistance. It offers the advantages of no moving parts and high reliability at the expense of considerable added weight and a low coefficient of performance (COP). Typically the electrical power required by a thermoelectric cooler is about ten (10) times the wattage being cooled; i.e., a COP of 0.1. This compares unfavorably with, for example, freon cycle refrigerators which have a COP in the neighborhood of 3.5; i.e., 35 times better.

7.3.6.2 Thermoelectric devices as heat conductors. If thermoelectric coolers are used as positive thermal resistances (not refrigerators to depress the temperature) the COP will exceed unity with small temperature rises. The possibility of using a thermoelectric cooler as a controllable series thermal resistance of very low value is often overlooked. Not only is a low resistance provided between two locations, but it can offer a controllable variable thermal resistance. Solid state controllable power supplies for thermoelectric coolers are available. (See chapter 12 for additional information on thermoelectric devices).

7.3.7 Heat pipes. Heat pipes are discussed in detail in chapter 12. A heat pipe is basically a heat conductor which is far superior to metallic heat conductors. Thermal resistances of milli-degrees C per watt can be obtained and heat in excess of 50 kilowatts can be transferred. Heat pipes offer important advantages. First, a heat pipe has several thousand times the heat transfer capacity of the best heat conducting materials on a weight and size basis. Second, heat pipes exhibit an essentially uniform temperature at the heat input end. Third, the areas and configurations of contact at each end of a heat pipe are independent and can be designed separately to suit the application. Fourth, a heat pipe can transfer heat up to several feet with a very small temperature drop. Fifth, a heat pipe can be made entirely of insulating material to provide electrical isolation for high voltage equipment. Sixth, heat pipes require no power for their operation, since capillary pumping is provided by the heat being transferred.

Heat pipes can be designed for operation at any temperature pertinent to cooling electronic equipment. In general, they must be designed specially for each application. One of the major difficulties in heat pipe application is in achieving adequately low thermal resistances at the interfaces at the heat pipe ends. Also, the capillary pumping in a heat pipe is influenced by gravity and external forces. Some heat pipes have exhibited as much as an order of magnitude change in internal thermal resistance due to gravity; (i.e., transferring heat downward vs upward). Further, heat pipes exhibit startup difficulties because enough condensate must collect to saturate the wick before pumping can be initiated. If a heat pipe is overloaded it can rupture, permitting high temperature gas to escape with the possibility of resultant damage to the equipment and flesh burns to personnel.

7.4 Tradeoff techniques. Tradeoffs among the various cooling techniques can be accomplished by various procedures. One acceptable method involves the assignment of Figures of Merit to each factor or aspect pertinent to the application. Figure of Merit values are assigned arbitrarily (usually on a 1 to 10 scale) to the advantages and disadvantages and summarized for comparison of candidate cooling techniques. For example, thermal resistance can be assigned a larger Figure of Merit than weight in shipboard application. However, in an airborne application, weight could have a higher Figure of Merit. Typical factors to be assessed in a cooling technique tradeoff are: thermal resistances, weight, maintenance requirements or maintainability, reliability, including the ancillary equipment such as blowers and pumps, cost, manufacturing tolerances, logistics aspects (special parts and coolants), thermal effectiveness, efficiency or coefficient of performance (COP), environmental resistances and ruggedness (shock, vibration, corrosion), personnel hazards (toxicity of coolants or vapors), size, complexity, power consumption, and effects on the electrical performance of the equipment. The later design chapters in this handbook discuss many of these factors in detail.

The Figure of Merit values to be assigned in a tradeoff evaluation can only be established when the specific requirements for the application of the equipment under consideration are known. The designer must have detailed knowledge of the application in order to make a valid assessment. DOD technical personnel can significantly assist in these determinations. Table II presents representative magnitudes of heat transfer processes.

7.5 Selection of optimum cooling method. Example 7-1: Construction of electronic equipment which dissipates 300 watts is contemplated. It is planned to package it in a cabinet 9.75 in. x 15 in. x 17 in., which is to be located in air at normal room temperature.

a. Will any special cooling considerations be required for this package?

TABLE II. Representative Magnitude of Heat Transfer Processes

	Btu / (hr) (sq.ft.) (^{o}F)	Watts / (sq.in) (^{o}C)	Unit Thermal Resistance (^{o}C) (sq.in.) / Watt
Conduction through copper 0.1 in. thick	26160.	95.20	0.0105
Conduction through pyrex glass 0.1 in. thick	87.36	0.322	3.11
Conduction through cork board 0.1 in. thick	3.0	0.011	91.
Free convection from 6 in. high vertical plate at 120^{o}C, air at 80^{o}C.	0.96	0.00348	287.
Forced convection air over 6 in. plate at 8ft/sec., mean temp. air & plate of 100^{o}C	2.84	0.0104	96.
Forced convection 40^{o}C. water flowing at 5 ft/sec. in a 2 in. dia. pipe	1420.	5.19	0.193
Water boiling on a flat plate at atmospheric pressure	2000.	7.30	0.137
Steam condensing on a flat plate at atmospheric pressure	1000.	3.65	0.274
Radiation between two blackbodies at 100^{o}C. and 50^{o}C.	1.72	0.0063	158.
Radiation between two blackbodies at 500^{o}C. and 50^{o}C.	7.81	0.0287	34.8

b. Can this package be made smaller?

$$\text{Heat concentration} = \frac{\text{Dissipated power}}{\text{Volume}}$$

$$= \frac{300}{9.75 \times 15 \times 17} = \frac{300}{2486}$$

$$= .12 \text{ watts/cu. in.}$$

This is a low heat concentration. No particular cooling considerations are required provided that the unit heat dissipation is adequate. (See Figure 5).

$$\text{Unit heat dissipation} = \frac{\text{Dissipated power}}{\text{Area of cooling surface}}$$

$$= \frac{300}{2x9.75x17+2x9.75x15+2x15x7} =$$

$$\frac{300}{1135}$$

$$= 0.26 \text{ watts/sq. in.}$$

Referring to Figure 5, note that the maximum unit heat dissipation for free air cooled surfaces is in the neighborhood of 0.25 watts/sq. in. Thus, the package with 0.26 watts/sq. in. surface area will be satisfactory and no special cooling means will be required.

From Figure 5 note that a unit heat dissipation of one watt/sq. in. of surface area may be feasible if, for example, forced air cooling is used on the external surfaces. Thus, with forced air cooling it may be possible to miniaturize the package from a surface area of 1135 sq. in. to a surface area of 300 sq. in. or less, dependent upon the Reynold's number and provided that the heat concentration is not excessive.

External dimensions of 7 in. x 5 in. x 10 in. appear in order for the miniaturized unit cooled by forced air.

$$\text{Unit heat dissipation} = \frac{300}{7x5x2+5x10x2+7x10x2}$$

$$= \frac{300}{310} = \text{approx. 1 watt/sq. in.}$$

$$\text{Unit heat concentration} = \frac{300}{7x5x10} = \frac{300}{350}$$

$$= 0.85 \text{ watts/cu. in.}$$

This is fairly high heat concentration. Metallic conduction cooling could be used within the externally forced air cooled unit satisfactorily if paths of low thermal resistance to the external surfaces are incorporated.

7.6 Application of selected cooling method to electro-thermal analog. The selected cooling method should be evaluated using the detailed procedures presented in the design chapter(s) pertinent to the technique(s) selected to make sure that the ranges of thermal resistances desired can be readily achieved. This will provide the designer with "insight and a feel" for the values obtainable. Based on this information the final values of thermal resistance can be assigned to the electro-thermal analog of the thermal system. This procedure of converging on the final values may require a little additional time but, it is well worthwhile, since it can lead to a near optimum or optimum thermal design.

8. NATURAL METHODS OF COOLING

8.1 Theory. Natural methods of heat transfer are defined as those which do not require any additional energy from an outside source, such as blowers, compressors, pressurized gas, etc. Natural methods are frequently the only practical means of removing heat from within miniaturized assemblies. Hermetic sealing and dense packaging of parts may prevent the utilization of other cooling techniques.

Electronic parts are generally designed and rated by manufacturers' for natural cooling. There are certain exceptions to this, such as high power vacuum tubes, diodes, and transistors, klystrons, etc., which may be designed for liquid or forced air cooling in accordance with the manufacturers' specifications.

8.1.1 Conduction. Heat conduction is considered to be caused through molecular oscillations in solids and elastic impact in liquids and gases. The basic law of heat conduction in its most general form, known as Fourier's equation, is written in differential form:

$$\frac{\partial q}{\partial A} = (k \cdot \nabla)\, T \qquad (8\text{-}1)$$

The heat flux density $\partial q/\partial A$ is a vector quantity. Temperature is a scalar quantity. Thermal conductivity k is a dyadic, since it is a function of both temperature and position. However, in practical situations, amenable to engineering solutions, it may be considered constant.

Thus,

$$\partial q/\partial A = k \cdot \nabla T \qquad (8\text{-}2)$$

The continuity equation for heat flow states that the net heat flowing into a closed region equals the difference between the time rate of heat produced and absorbed within the region.

$$q = \oint \frac{\partial q}{\partial A} \cdot dA = \frac{\partial}{\partial t} \int - \left(\frac{\partial E}{\partial v} - c_p T\right) dv \qquad (8\text{-}3)$$

where E is the internally generated heat, if any. Applying Gauss's relation between surface and volume integrals:

$$\nabla \cdot \frac{\partial q}{\partial A} = \frac{\partial p}{\partial v} - \rho c_p \frac{\partial T}{\partial t} \qquad (8\text{-}4)$$

where $p = \partial E/\partial t$ is the rate of internal heat production. From Equation 8-2:

$$\nabla \cdot \frac{\partial q}{\partial A} = -k\, \nabla \cdot \nabla T \qquad (8\text{-}5)$$

Equating Equations 8-4 and 8-5, Equation 8-6 for temperature distribution within the region is derived.

$$\frac{k}{\rho c_p} \nabla^2 T = \frac{\partial T}{\partial t} - \frac{1}{\rho c_p} \frac{\partial p}{\partial v} \qquad (8\text{-}6)$$

The parameter $k/\rho c_p = \alpha$ is known as thermal diffusivity.

These vector differential equations apply to very general heat conduction problems. In most practical engineering cases they can be greatly simplified.

When heat flows through a region of constant cross section area A, Equation 8-2 becomes:

$$q = \frac{-kA}{L}(T_1-T_2) = \frac{\Delta T}{L/kA} \qquad (8\text{-}7)(D.E.)$$

Heat flow by conduction is therefore analogous to electric current flow.

$$\frac{L}{ka} = R = \text{Thermal resistance} \qquad (8\text{-}8)(D.E.)$$

While the equation for heat conduction is analogous to that for Ohm's law, the range of thermal conductivity is much narrower than that for electrical conductivity. Electrical insulators have resistivities ten to twelve magnitudes greater than that of good conductors. Thermal insulators have resistivities four or five magnitudes greater than that of good heat conductors. Physically, this is due to the fact that molecules are more tightly bound and have much less mobility than do electrons.

Heat spreads out and is not confined to exact geometrical paths to anything like the extent that applies with electric current. Therefore, a heat conduction calculation must usually be treated as a field problem instead of a circuit problem if precise results are desired.

If q is measured in watts, lengths in inches, and T in °C, k is given in watt-inches/sq. in.-°C, commonly called watts/in.-°C.

Thermal conductivity of most materials does vary with temperature. However, the variation for metals over the range of temperature of interest in electronic cooling design is not great. Consequently, for the purpose of this handbook the variation of thermal conductivity with temperature is disregarded. Tables in the Appendix present the thermal conductivity values for various materials.

8.1.1.1 Electrical analogy. An electrical analogy which may be readily applied by electronic engineers is presented in Figure 6.

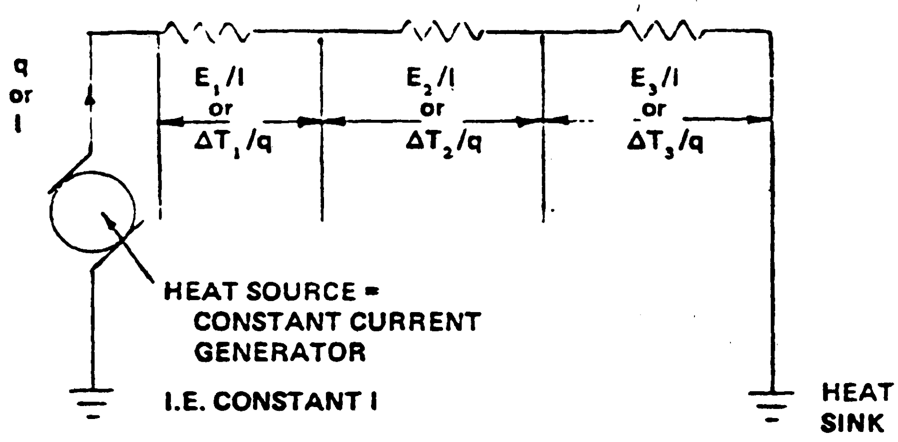


FIGURE 6. Thermal Analog Circuit

where:

Voltage drops (E) are analogous to temperature differences (ΔT) in degrees C. The constant current I is analogous to the constantly dissipated power q in watts. The electrical resistance R (E/I=volts/amps.) is analogous to the thermal resistance R.

Since:

$$R = \frac{\Delta T}{q} \tag{8-9}$$

Then, thermal resistance may be expressed in degrees C per watt

In this handbook the electric circuit analogy is utilized wherever heat conduction is involved. It is also applied in many problems involving convection and radiation heat transfer through the use of equivalent thermal resistances.

8.1.1.2 Calculation of thermal resistance. Example 8-1: Calculation of the thermal resistance of a bar insulated on all sides:

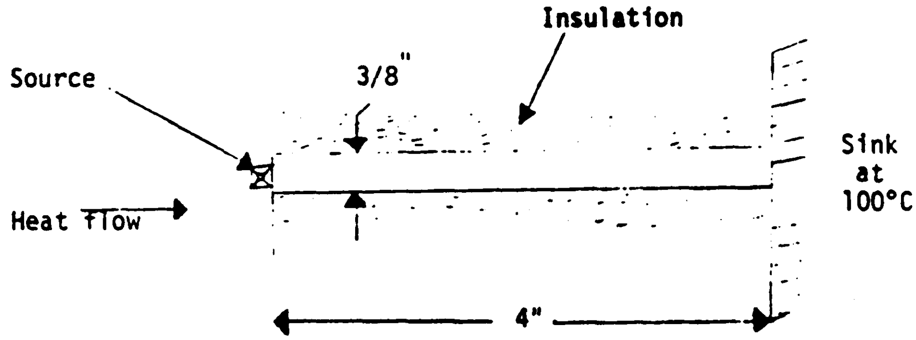


Figure 7. Conduction Through a Bar

Fig. 7 shows a soft steel bar 3/8 in. diameter, 4 in. long, one end of which is maintained at 100°C, and the other is heated by a resistor dissipating 2 watts. The sides of the bar are considered to be perfectly insulated so that heat flows only in the direction parallel to its axis.

Example 8-2:

Problem: Determine the temperature of the resistor, assuming perfect thermal contact with the bar.

Solution: The thermal conductivity (k) of steel is 1.18 watt-in./sq. in.-°C. The area of the bar is:

$$A = \frac{1}{4} \pi (3/8)^2 = 0.110 \text{ sq.in.}$$

Thus, the resistance of the bar is:

$$\frac{4}{1.18 \times 0.110} = 31°\text{C/watt}$$

The temperature rise for 2 watts is 2 x 31 or 62°C.

Consequently, the temperature of the end of the bar and the surface of the resistor is 162°C.

8.1.1.3 Conduction through composite walls:

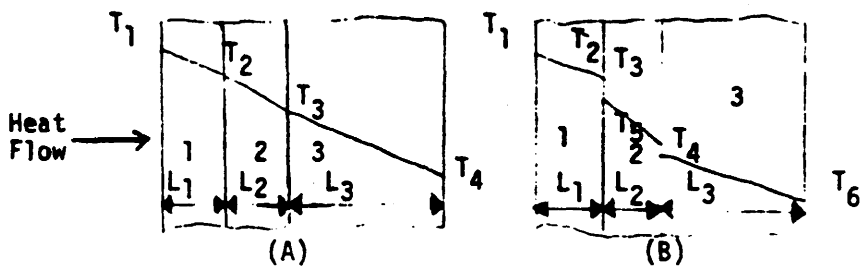


Figure 8. Conduction Through Composite Walls

Figure 8(A) shows a composite wall or bar insulated on the ends and made up of three different materials. It is assumed that the materials make perfect contact at the joints, a condition which is difficult to attain unless the materials are metals and bonded to each other by solder or other such means. Perfect contact at the joints eliminates the high resistance to heat transfer caused by any surface roughness with accompanying air films between the joints. The equation for the heat flow is:

$$q = \frac{(T_1 - T_4)}{\frac{L_1}{Ak_1} + \frac{L_2}{Ak_2} + \frac{L_3}{Ak_3}} \qquad (8\text{-}10)$$

Each term in the denominator is a resistance to heat transfer and since this is a series thermal circuit, the resistances are additive. The temperature drop across each material is proportional to its thermal resistance. For example, the temperature drop for the second material is:

$$T_2 - T_3 = \frac{\frac{L_2}{Ak_2}}{\frac{L_1}{Ak_1} + \frac{L_2}{Ak_2} + \frac{L_3}{Ak_3}} (T_1 - T_4) \qquad (8\text{-}11)$$

In cases where the materials are not actually bonded together, the thermal resistance at each joint should be considered. Figure 8 (B) shows a composite wall or bar of three materials with imperfect thermal joints, having resistance at each joint. Since there is an abrupt temperature drop at each joint, there are five thermal resistances: one for each material and one for each contact or joint. The equation for the heat flow is:

$$q = \frac{(T_1 - T_4)}{\frac{L_1}{Ak_1} + R_{1-2} + \frac{L_2}{Ak_2} + R_{2-3} + \frac{L_3}{Ak_3}} \qquad (8\text{-}12)$$

where: R_{1-2} and R_{2-3} are the thermal contact resistance offered by the first and second joint respectively. This contact resistance is complex because of the nature of the variables affecting it, such as the surface finish or roughness, the flatness of the contacting surfaces, the pressure holding adjacent materials together, and the materials used. It appears reasonable to neglect contact resistance where two surfaces are welded, soldered, or brazed together so that the contact is practically perfect. On the other hand, if two surfaces are not so bonded, the thermal contact resistance should be estimated.

It is of interest to convert the contact resistance into an equivalent length of material whose resistance due to pure conduction would be the same. For example, a contact resistance for a cross-section of unit area, (i.e., actually the contact restivity) of 1.0°C-in.^{2}/watt is equivalent of 1.18 inches of steel of unit cross-sectional area

$$\frac{1.0°\text{C-in.}^2}{\text{watt}} \times \frac{1.18 \text{ watt-in.}}{\text{in.}^2\text{-°C}} \text{ (steel)} = 1.18 \text{ in. steel}$$

in pure conduction. (The thermal conductivity of mild steel is 1.18 watts/(sq.in.) (°C)/in.) With aluminum the thermal resistance of a good joint will be equivalent to that of a length of 1/2 in. to 1 in. of aluminum of the same area.

The thermal resistance decreases if one surface of a steel joint is copper plated.

Example 8-4: Conduction with and without contact resistance.

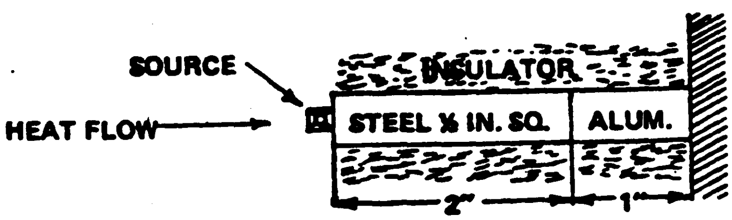


FIGURE 9. Composite Bar With Contact Resistance

Figure 9 presents a composite bar 1/2 in. sq. made of steel and aluminum and insulated on the sides. One end of the bar is attached to a thermal sink at 100°C and the other end is connected to a heat source dissipating 10 watts. Assume a contact resistivity of .34°C/watt/sq. in. corresponding to a contact pressure of 10 psi at the joint.

Problem: Determine the temperature rise along the bar. The thermal resistance of the joint is .34/.25 = 1.36°C/watt. The temperature rise across the joint is 1.35 x 10 = 13.6°C. The thermal resistance of the steel, R_s is:

$$R_s = \frac{L}{K_s A} = \frac{2}{(1.18)(0.25)} = 6.8°C/watt$$

The temperature rise in the steel is 6.8 x 10 = 68°C. The thermal resistance of the aluminum, R_A is:

$$R_A = \frac{1}{(5.1)(0.25)} = 0.78°C/watt$$

The temperature rise in the aluminum is .78 x 10 = 7.8°C. The total rise is 7.8 + 68 + 13.6 = 89.4°C. Thus, the hot end of the bar is at 189.4°C. Note that the thermal resistance of a 1/2 in. sq. aluminum bar 1 in. long is 0.78°C/watt whereas that of a steel bar the same size is 3.4°C/watt. A copper bar of the same size would have a thermal resistance of 0.41°C/watt.

8.1.1.4 Theoretical consideration of contact resistance. The mechanism of heat transfer across surfaces in contact is exceedingly complex. However, from a design standpoint the data available indicate the order of magnitude and range of the conductance values, as well as the trends of their variation. It appears that on a microscopic scale there exists no sharp demarcation between contact and separation: and, even if such demarcation could be conceived, as long as the air between the surfaces is a conductor, the transition between finite resistance and zero resistance at any place on the interface must be continuous and gradual. Instead of islands of contact and seas of separation the interface should be visualized as a region varying in thickness from the order of atomic spacing to that of a few ten-thousandths of an inch. In this region air molecules of finite size move about randomly under thermal agitation: some are reflected by a change of conductance and others are not. It is to be expected that the more intimately the two surfaces are in contact the more a small change in the matching configuration will be reflected by a net change of conductance. This accounts for the fact that pressure has a more pronounced effect, as evidenced by the absolute rise of conductance, on smoother surfaces than on rougher surfaces, and that the amount of scattering increases with increasing pressure and decreasing roughness. (References 4 & 5) Of all the factors which contribute to the change in interface matching configuration, the factor of interface pressure is perhaps the most important. The effect of pressure is more pronounced in harder materials. Section 8.3.3 presents detailed information on interface contact resistance and Appendix B includes tabulated data for a variety of interface configurations.

8.1.1.5 Conduction through thin air gaps. Although gases are poor conductors of heat, some conduction does occur between heated surfaces separated by short gaps filled with air or any other gas. It has been established experimentally that heat transfer occurs chiefly by conduction, and that convection is negligible, for gaps 1/4 inch thick and less; and that convection is predominant for gaps thicker than 1/2 inch. Radiation occurs in all cases. The thermal resistance of the gap is calculated in the same way as for any piece of material:

$$R = L/kA \tag{8-13}$$

Since the heat flow lines fringe out, the same as magnetic flux lines, it is advisable to compute area by adding the gap thickness to each dimension, thus reducing the resistance slightly.

It should be understood that conduction, as well as radiation, always occurs, but that as the gap becomes longer convection currents develop and account for a larger proportion of the transmission.

8.1.1.6 Conduction through complex shapes. It is evident that thermal resistance calculations are relatively simple for regular shapes of constant cross section. However, when the shape is complex, calculation is more difficult. Heat flows in streamlines which terminate on isothermal or equipotential surfaces, the same as fluid flow and magnetic flux. The same methods of calculation thus apply.

8.1.1.7 Estimation of thermal resistance. At least four methods are available for estimating thermal resistance. First: If a simple equation connecting length and area can be written to describe the shape, ordinary integration can be used. For example, to calculate the resistance through a spherical shell of radii r_1 and r_2,

$$R = \int \frac{dr}{kA} = \int_{r_2}^{r_1} \frac{dr}{4\pi kr^2} = \frac{1}{4\pi k}\left(\frac{1}{r_1} - \frac{1}{r_2}\right) \tag{8-14}$$

Second: Flux mapping or relaxation techniques can be used, just as they are for magnetic reluctance and fluid flow resistance. (References 6, 9, 41, and 42) See page 76 for notes on field mapping and relaxation techniques.

Third: For heat flow through fluids between isothermal surfaces, a capacitance measurement between similarly shaped electrodes immersed in a dielectric fluid is proportional to thermal conductance. This is true because the field pattern for electric and heat flux is the same.

$$R_{thermal} = \frac{1}{C} \cdot \frac{E}{k} \tag{8-15}$$

where C = capacitance

E = dielectric permittivity

As an example, it was required to find the thermal resistance through air between two parallel wires which crossed other wires in a computer memory bank. Capacitance between the two wires measured 6.6 picofarads. For air, E = 8.85 pf/meter and k = 0.026 watt/meter-°C. Thus,

$$R_{thermal} = \frac{1}{6.6 \times 10^{-12}} \cdot \frac{8.85 \times 10^{-12}}{0.026} = 51.5°C/watt$$

Fourth: Electrical measurement of thermal conductivity: Over 100 years ago it was found that the electrical and thermal conductivities of pure metals at a given temperature have a nearly constant ratio. Quantum theory has established the quantitative relation:

$$\frac{k}{\sigma} = CT \qquad (8\text{-}16)$$

where k = thermal conductivity
σ = electrical conductivity
T = absolute temperature
C = a universal constant

The constant C is known as the Lorenz Number and is theoretically equal to 2.45×10^{-8} watt - ohms/degree squared. Actual values of the constant differ somewhat from this value, especially at low temperatures. The proportionality of thermal to electrical conductivity holds only above the Debye temperature, the point at which the motion of free electrons ceases to contribute to heat transmission. Since the Debye point is cryogenic for most metals, a useful means of determining thermal conductivity by measuring electrical conductivity is available.

If k is given in watt-in./sq. in.-°C and σ in sq. in./in.-ohm, then:

$$\frac{K}{\sigma}\left(\frac{\text{watt-ohm}}{°C}\right) = C\left[\frac{\text{watt-ohm}}{°C \times °C}\right] T(°K) \qquad (8\text{-}17)$$

Values of the constant C are difficult to obtain except for pure metals. However, the theoretical relation is useful for determining the thermal resistance of contacts and composite joints if the Debye temperature is low and if the contact is between similar metals.

When a thermal resistance between pieces of the same metal in contact is required, the electrical resistance is measured between the points, and is then multiplied by the ratio of thermal to electrical conductivities expressed in a consistent set of units.

If dissimilar metals are involved this method fails. However, it provides a rapid and useful means of inspecting interface thermal contacts in production. Assuming that a prototype joint has been evaluated by thermal tests, the electrical resistance is measured with a milliohmeter. This measurement is then used for production checking.

8.1.2 Convection. The process of heat transfer by means of moving masses of fluid is known as convection. The fluid adjacent to a surface at a higher temperature absorbs heat, which is convected to a cooler surface as the fluid moves. Two types of convective heat transfer are possible. When the fluid is forced to move by external means, as by a fan, blower, or pump the process is called forced convection. Free convection, which is the type discussed in this chapter, occurs when the fluid motion is caused by the decreased density of the heated fluid and the consequent buoyant force. Since this effect depends on gravitation, free convection is inoperative in a zero gravity environment.

It is evident that free convection is influenced by many factors: density, viscosity, specific heat, thermal conductivity, and thermal expansion coefficient of the fluid; shape, size, and orientation of the heated surface, and the temperature gradient through the fluid stream.

The basic equation for convective heat transfer is:

$$q = h_c A_s \Delta T_{s-f} \tag{8-18}$$

where h_c is the convective heat transfer coefficient, A_s is the heated surface area, $\Delta T^s{}_{s-f}$ is the temperature difference between the surface and the fluid. The latter is difficult to define, and h_c is ordinarily calculated for ΔT between surface and "free stream" or entering air temperature. Since h_c is itself a function of ΔT a non-linearity is involved, and convective heat transfer is much more difficult to calculate theoretically than conduction. The thermal resistance concept can be applied to convection, as well as to conduction, but the determination of an equivalent thermal resistance depends on an estimate of some average value of h_c over the entire surface to be cooled.

Free convection is a bilateral phenomenon for a given geometry. In this section the transfer of heat from a hotter surface to a colder fluid is discussed, but a hotter fluid gives up heat to a colder surface in an exact reversal of the process, for the same orientation of the surface.

Free convection heat transfer coefficients are relatively small so that the equivalent resistance is high compared to those representing other modes of heat transfer (except possibly for radiation). Thus, for example, if conduction and convection act virtually in series, convection is usually the dominant limit to heat transfer. If they act in parallel, conduction is dominant.

8.1.2.1 Theoretical considerations. Dimensional analysis applied to the physical mechanism of free convection identifies three so-called "dimensionless groups," which have been named after early researcher, the Nusselt (Nu), Grashoff (Gr), and Prandtl (Pr) Numbers. In terms of physical parameters these dimensionless groups are:

$$(Nu) = h_c L/k \tag{8-19}$$

$$(Gr) = g\, \beta \Delta T L^3\, \rho^2/\mu^2 \tag{8-20}$$

$$(Pr) = c_p \mu/k \tag{8-21}$$

The functional relation Nu = f(Gr, Pr) can be expressed by Equation 9-22:

$$(Nu) = C(Gr)^m(Pr)^n \tag{8-22}$$

$$\text{or } h_c = C\,\frac{k}{L}\,(Gr)^m(Pr)^n \tag{8-23}$$

C is a dimensionless factor related to the size, shape, and position of the heat transfer surfaces. L is a so-called characteristic length dimension which also varies with the size, shape, and position of the convecting surface. Table III lists values of L for common conditions, and may be used for irregularly shaped electronic parts if the most similar condition is selected. Table IV lists corresponding values for the constant C. Appropriate values of C and L for specific electronic parts are given in later paragraphs of this chapter.

A great deal of theoretical and experimental research has been performed on free convection. About 1880, Lorenz published the first good theoretical work on convection from high vertical surfaces (Reference 7) and showed that the exponents m and n are both equal to 1/4. Later work showed that the height of vertical surfaces is an important parameter. The flow pattern evidently becomes fully developed in a vertical distance of about 24 inches, which accounts for the fact that Table III specifies this as the maximum value to be used. This restriction is not important in electronic cooling except for large structures such as cabinets and enclosures.

It has been found by a large number of investigators that free convection data covering a wide range of fluids, configurations, and temperatures, falls close to a smooth curve when log (Nu) is plotted vs log (Gr, Pr). This agrees with the theoretical studies by Lorenz and others. Equation 8-24 may be used:

$$Nu = h_c L/k = C\,(Gr,\ Pr)^m \qquad (8\text{-}24)(D.E.)$$

The constants c and m are determined from the experimental curves. For values of (Gr, Pr) less than 10^{-5}, Nu approaches a constant value of about 0.4, so low as to be practically useless for cooling electronics. For relatively large surfaces, m is 1/4 when (Gr, Pr) is less than 10^8, and 1/3 for larger values (Reference 6). For small parts which occur in electronic equipment, 1/4 is recommended for $10^3 < Gr\,Pr < 10^9$, and 1/3 for greater values. (Reference 7) Equation 8-24 is frequently written:

$$Nu = C\,(a\,L^3\,\Delta T)^m \qquad (8\text{-}25)$$

where "a" is called the convective modulus.

$$a = (Gr,\ Pr)/L^3 \cdot \Delta T = \frac{g\rho^2 \varepsilon c_p}{k\mu} \qquad (8\text{-}26)$$

8.1.2.2 Pressure-altitude effects. It has been noted that free convection is ineffective in a zero gravity environment. Since gravitational acceleration enters into the Grashoff Number, free convection cooling is affected by the local gravity. It may, therefore, act in spin stabilized space vehicles, on the moon, and in other extraterrestrial environments, provided there is a fluid coolant present. Thermal conductivity, specific heat, dynamic viscosity, are all nearly independent of pressure for gases and liquids, over a range from "soft" vacuum to several atmospheres. But the density of gases varies widely with pressure. Air at high altitudes is therefore a poor medium for free convection.

All of the equations given in this chapter may be used for any environment if the correct values of g and of density are known.

8.1.2.3 Methods of calculation. Equations 8-24 and 8-25 apply strictly to each infinitesimal area of the convecting region, and the parameters of the fluid should be evaluated at the prevailing local fluid temperature. This is impracticable, and fortunately unnecessary for most problems of cooling electronic equipment, since the range of temperature is not great and the convective modulus may be evaluated at an average temperature. These equations are therefore, used as integral forms applying to the entire surface.

Values of Gr and Pr and occasionally of a, may be found in tables in various handbooks and manufacturers' publications for commonly used fluids. These are usually given as functions of temperature. If it is necessary to calculate Gr and Pr, the parameters should be evaluated at an average of surface and fluid temperatures.

The unit heat dissipation, q/A_s is a frequently used quantity. From equations 8-18, 19, and 24, equation 8-27 is derived:

$$q/A_s = C\ k/L\ (Gr,\ Pr)^{0.25}\ \Delta T \qquad (8\text{-}27)(D.E.)$$

where the exponent 0.25 is used since electronic cooling situations usually fall within the range where that value is applicable. Alternatively, the equation may be written:

$$q/A_s = Ck(a/L)^{0.25}\ \Delta T^{1.25} \qquad (8\text{-}28)(D.E.)$$

In practical calculations the question of units and conversions always enters. Since Nu, Gr, and Pr are dimensionless groups they have the same values no matter what unit system is used to compute them, provided only that consistent units are used for all of the parameters involved. Therefore, the constant C is independent of units. The convective modulus however, has the dimensions of reciprocal volume x temperature. Therefore, when Design Equation 8-28 is used, "a" must be computed for the desired units of length, power, and temperature. If Design Equation 8-27 is used, k/L must be so treated. For example, if the recommended unit of watts/sq. in. is desired, it is necessary only to express k in watts/in.°C, L in inches, and ΔT in °C. Reference should be made to paragraph 4.10, Units and conversion factors.

The majority of tables of thermal properties of materials use the units of BTU, pounds of mass, hours, °F, and feet, as is still customary with American mechanical engineers. Some tables prepared for chemists and physicists use calories, grams, seconds, and °C. Various other units are found in use, and constant care must be exerted to guard against errors in unit conversions. It is recommended that all of the raw data used in every calculation be converted initially into terms of watts (or joules), pounds of mass, inches, and °C, and that dimensional checks of the calculations be made in these units.

It is particularly important to express the gravitational acceleration in the correct units.

Air is the most common coolant for free convection cooling of electronic equipment. Except for cabinets and large enclosures, most of the parts to be cooled are much less than 24 inches in any dimension. Also, as seen from Table V, the value of (k al/4) varies ± 6% from 70 to 400°F, (approximately 20 to 200°C), the range for most electronic equipment. Equation 8-28 can therefore be simplified for sea level air by using C = 1.45 and (kal/4) = 0.00394 watts/in. $^{7/4}$ °C $^{5/4}$

$$q/A_s = 0.00570\ \Delta T^{1.25}/L \qquad (8\text{-}29)(D.E.)$$

where q/A_s = watts/in.
ΔT = °C
L = in.

Equation 8-29 will yield good approximate values of q/A_s for most small electronic parts cooled by sea level air.

TABLE III. Characteristic Length

Surface	Position	Characteristic Length, L
Plane	horizontal	$\frac{(\text{Length}) \times (\text{Width})}{\text{Length} + \text{Width}}$
Plane (rectangular)	vertical	vertical height but limited to 24 in.
Plane (non-rectangular	vertical	$\frac{\text{area}}{\text{horizontal width}}$
Plane (circular)	vertical	0.785 x diameter
Cylinder	horizontal	diameter
Cylinder	vertical	height of cylinder but limited to 24 in.
Sphere	any	radius (diameter/2)

TABLE IV. Values Of The Constant C

Shape and Position	C
Vertical plates	0.55
Horizontal cylinders (pipes and wires)	0.45
Long vertical cylinders	0.45 - 0.55
Horizontal plates facing upward	0.71
Horizontal plates facing downward	0.35
Spheres (L = radius)	0.63
Small parts (see text)	1.45

For more general use it is desirable to include the proper value of "C" to allow for different shapes and orientations. Equation 8-28 is rewritten:

$$q/A_s = 0.00394\ C\ (\Delta T)^{1.25}/(L)^{0.25} \qquad (8\text{-}30)(\text{D.E.})$$

where q = watts
ΔT = °C
A_s = sq. in.
L = in.

Figure 10 is a nomographic chart for solving Equation 8-30. The range of characteristic length L, the configuration factor C, and the temperature differences are sufficiently broad to cover all free convection problems in air which will be met in electronic equipment cooling problems.

In the usual problem, C and L are found from the known shape, size, and orientation. A straight line through C and L locates a point on the uncalibrated X scale, which represents ($C/L^{0.25}$).

If ΔT is known, a straight line through X and ΔT gives unit heat dissipation q/A. Conversely if q and A are known, ΔT is found.

Example 8-4: The average surface temperature of a metal enclosure located in free air at 35°C is 85°C. The enclosure is 24 in. long x 12 in. wide x 12 in. high. How many watts are dissipated from the top surface only by free convection? (Neglect conduction and radiation, and assume that the outer air temperature remains at 35°C because the surrounding space is well ventilated).

Solution by Chart

Step (1): Determine significant dimension L from Table III, L = (length x width)/(length + width), for a horizontal plane, = (24 x 12)/(24 + 12) or 8 in.

Step (2): Determine constant C from Table IV (C is 0.71 for a horizontal plate facing upward).

Step (3): Determine ΔT by subtraction of air temperature from surface temperature. ΔT = (85-35) or 50°C.

Step (4): Determine area by multiplying length by width. A = (24 x 12) or 288 in.^{2}.

Step (5): Apply straight edge to L and C scales, at 8 in. and 0.71 respectively, to determine a crossing point on the uncalibrated X-scale.

Step (6): Connect this X-point with a 50°C-point on the ΔT-scale to determine a crossing point on the righthand q/A-scale. Read 0.23 w./in.^{2}.

Step (7): Multiply 0.23w./in.^{2} by 288 in.^{2} to obtain 66 watts.

Solution by Calculation

$$q = 0.00394C\ (\Delta T)^{1.25}\ A/L^{0.25}$$
$$= 0.00394 \times 0.71 \times 50^{1.25} \times 288/8^{0.25}$$
$$= 64.\ \text{watts}$$

TABLE V. Convective Properties Of Sea Level Air

T °F	°C	Pr	Gr/L^3-T	k watts/in-°C	a 1/cu.in-°C	$ka^{1/4}$ *
0	-18	0.73	3.8×10^{6}	5.7×10^{-4}	2950	0.00420
50	+10	.72	2.7	6.30	2020	.00422
100	38	.72	1.78	6.85	1330	.00414
150	65	.70	1.22	7.44	890	.00406
200	93	.69	0.86	7.96	618	.00397
250	121	.69	.61	8.45	439	.00387
300	149	.69	.45	8.93	322	.00383
350	177	.68	.34	9.50	239	.00375
400	204	.68	.26	10.00	187	.00367
450	232	.68	.20	10.48	141	.00359
500	260	.67	.15	11.00	104	.00352

* Note: The units of $ka^{1/4}$ are $\dfrac{\text{watts}}{\text{in}^{7/4} - {}^{\circ}\text{C}^{5/4}}$

8.1.3 Radiation

8.1.3.1 Theory. Hot bodies emit thermal radiation in the form of electromagnetic waves ranging in wave length from the long infrared to the short ultraviolet. Radiation emitted from a body can travel undiminished through a vacuum and through gases with relatively little absorption. When radiation is intercepted by a second body, part may be absorbed as thermal energy, part may be reflected from the surface, and part may be transmitted still in electromagnetic wave form through the body as in the case of glass. In terms of the radiation coefficients:

$$\tau + \alpha + \rho = 1 \qquad (8\text{-}31)$$

where
τ = Transmissivity
α = Absorptivity
ρ = Reflectivity

If all the incident radiation to a body in space receiving radiant energy is absorbed with zero energy being reflected or transmitted, it is a perfect absorber, a "black body." There are no perfect absorbers in nature although some surfaces approach black body characteristics. A cavity with walls of which are all at the same temperature is frequently used as a laboratory "black body." A furnace with a peephole through the wall is a good example. The ratio of the energy absorbed by a body to that of a perfect "black body" is the "absorptivity." In the absence of conduction and convection, a body at thermal equilibrium which receives radiation must necessarily emit radiant energy equal to that absorbed. Hence, a body which is a good receiver or absorber is a good radiator or emitter. The ratio of the amount of radiant energy emitted by an actual body to that emitted by the ideal black body is the "emissivity" and is numerically equal to the absorptivity (at thermal equilibrim). Its value is always less than unity. The emissivity of polished copper, for example, is 0.023, whereas that of oxidized cast iron may be as high as 0.95.

Tables in the Appendix 2 list emissivity values of various surfaces. In general, dull, dark surfaces are good absorbers (or emitters) and have high emissivity values. Polished surfaces have low values and can be used as radiation shields to protect parts from radiant heat sources.

At the temperature normally encountered in electronic equipment (100-200°C) the radiation wave length is large, peaking in the neighborhood of 7 microns. In this portion of the spectrum (the long infrared region) the emissivity is not necessarily related to the color in the visible region. However, most paints (of any color) are "dark brown" in the infrared spectrum due to the vehicle and binder. Black paints are just slightly blacker than light colored paints in the medium and long infrared region. The emissivity of paints can vary, dependent upon the constituents of the paint. Further, the emissivity varies as a function of temperature (wave length) both at the emitting and receiving surfaces.

The emissivity of a metal surface is also related to the roughness in that a brightly polished surface has a higher reflectivity than a rough surface. At higher temperatures these effects are less pronounced. Also, the surface material is important. For example, a polished metal surface with a low emissivity will have a high emissivity if painted black. It is possible to have two surfaces of the same color and smoothness, but of different materials, which will have different emissivities at infrared. At high temperatures (source or receiver) the emissivities may approach equality.

The basic equation for the radiation from a black body is:

$$q_b = \sigma_s A T^4 \tag{8-32}$$

where:

q_b is the rate of energy emitted by a black body.
σ_s is the Stefan-Boltzmann constant ($=36.8 \times 10^{-12}$ watt/in.^{2}-°K^4).
A is the surface area, in.^{2}.
T is the absolute Temperature, °K.

For actual bodies, Equation 8-32 must be modified for departure from ideal blackness and, since the net exchange of radiant energy between two bodies is usually required, it must be modified depending on the geometry of the system. The general equation for the net rate of exchange of radiant heat between two non-black bodies is:

$$q_r = F_e F_a A \sigma_s (T_1^4 - T_2^4) \tag{8-33}$$

where:

F_e is an emissivity factor to allow for departure from black body conditions.
F_a is a configuration factor based on the geometry of the system (not all of the radiation emitted by a body may be intercepted by the second body).
T_1 and T_2 are absolute temperatures of the hot and cold bodies respectively, °K.

The net radiation between two bodies is thus proportional to the difference in the fourth powers of the absolute temperatures, whereas conduction and convection in general are proportional to the difference in the first powers of the temperatures.

The emissivity factor F_e allows for the departure of two radiating surfaces from ideal blackness or unity emissivity. In general, F_e is a function not only of the emissivities of the radiating surfaces but of their geometric arrangement as well. For parallel planes which are large compared to their distance apart and also for a completely enclosed body which is large compared to the enclosing body, F_e is given by:

$$F_e = \frac{1}{\frac{1}{E_1} + \frac{1}{E_2} - 1} \qquad \text{(8-34)(D.E.)}$$

where:

E_1 and E_2 are the emissivities of the two surfaces.

For a completely enclosed body which is small compared to the enclosing body, such as an electronic box in a large compartment or room, the emissivity of the enclosing surface has little effect on F_e and $F_e = E_1$.

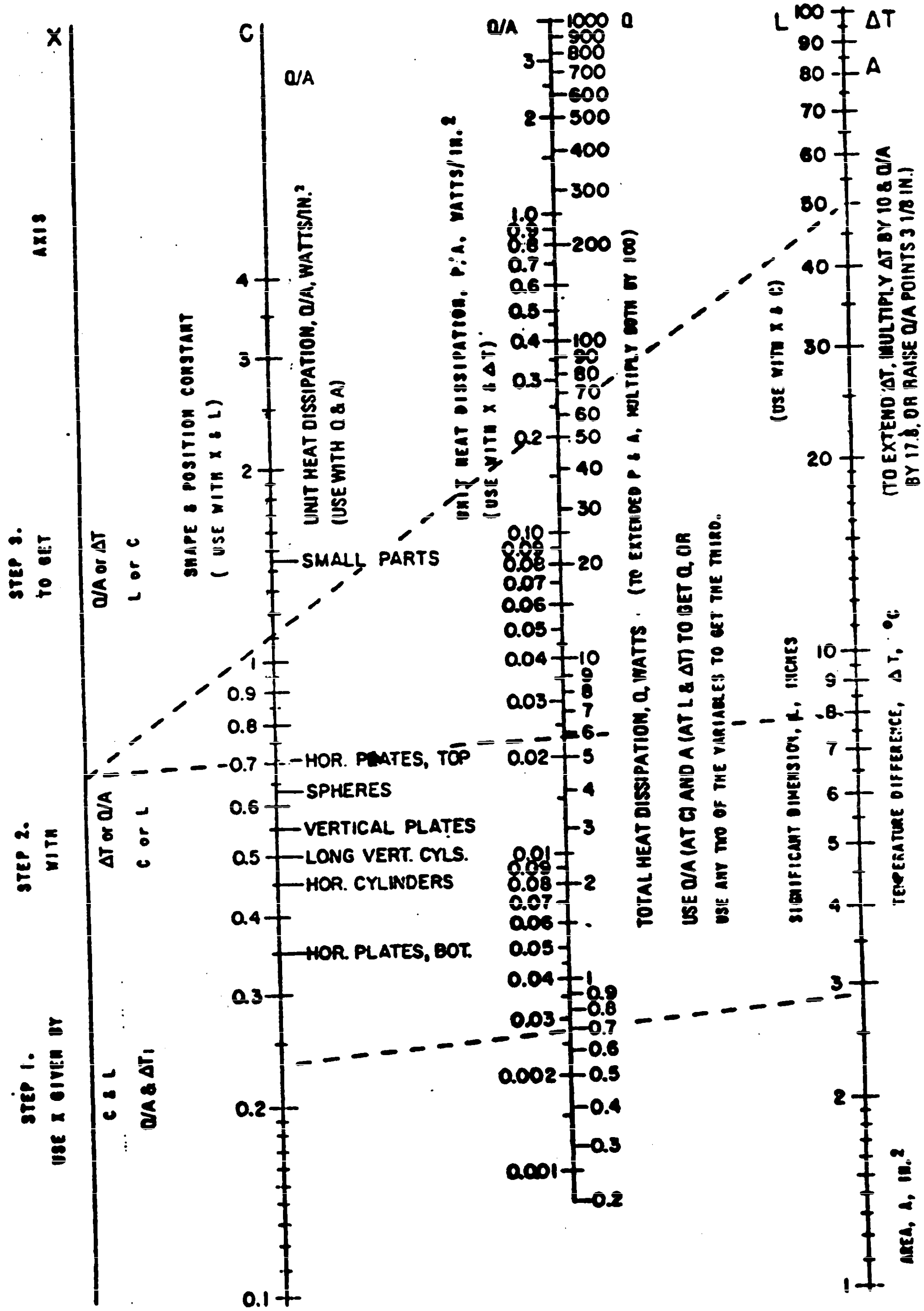


FIGURE 10. Free Convection Nomograph

where:

E_1 is the emissivity of the enclosed body. These are the more general configurations, but for others, see "Introduction to Heat Transfer" by Brown and Marco (Reference 8), "Principles of Engineering Heat Transfer" by Giedt (Reference 9), Jakob (Reference 6).

In the case of enclosed bodies, the area A in the radiation equation is that of the enclosed body. The configuration factor F_a takes into account the geometry of the radiating surfaces and the fact that not all of the radiation from one surface may reach the receiving surface. In many cases, such as the large parallel planes and enclosed bodies, F_a is unity. For certain other configurations F_a may vary widely. In general, the exact calculation of F_a involves integration over both surfaces. The cited references give calculating methods, and values for many specific cases.

Inserting the value of σ, Equation 8-33 may be written in the watt-sec.-sq. in.-°K system of units:

$$q_r = 0.00368A\ F_e\ F_a\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \text{ watts} \qquad (8\text{-}35)(D.E.)$$

The General Electric Company, Aerospace Electronics Department, Utica, New York sells a convenient and versatile slide rule "Radiation Calculator" for solving radiation heat transfer problems. Use of this calculator is advantagous. The price is only two or three dollars.

A less versatile but more simple device in the form of a nomographic chart is given in Figure 11. The chart solves the equation:

$$q/A = 0.00368E(T/100)^4 \text{ w./in.}^2 \qquad (8\text{-}36)(D.E.)$$

where:

E = emissivity of the radiating surface, and
T = absolute temperature, °K.

The chart contains three scales and relates the three variables, q/A, E, and T. Any variable can be found when the other two variables are known by simply connecting two known points with a straight edge and noting the crossing point on the third scale.

Equation 8-36 gives the rate of heat transfer or heat power radiated from one side of a unit area into hemispherical space at zero absolute temperature. No body can radiate as much heat power per unit area as the equation indicates, regardless of E and T values, because no environment has zero absolute temperature.

The chart is used twice in solving any practical radiation problem where there is a net heat exchange from one surface to another at a different temperature: first, with the emissivity and temperature of the enclosed surface to obtain the total radiation from the source; second, with the emissivity and temperature of the enclosing surface to obtain the total radiation from the receiver. The net difference is the heat exchanged by radiation. If the temperature of the enclosed surface is unknown, it can be found by using the chart backwards the second time.

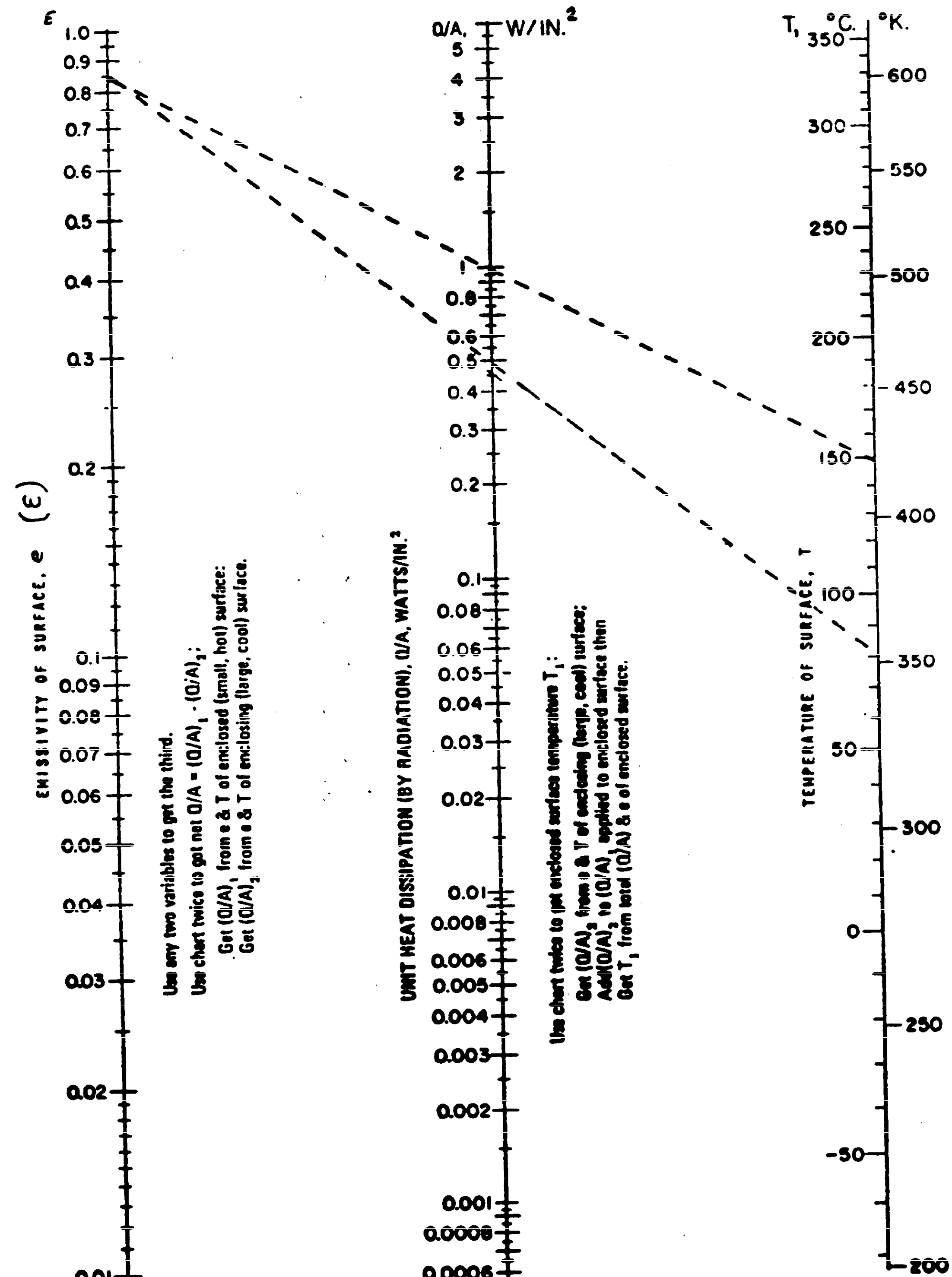


FIGURE 11. Radiation Nomograph

8.1.3.2 Radiation calculations. Example 8-5: Calculation of radiant heat exchange. Problem: An electronic assembly is housed in a steel box, 6 in. x 12 in. x 6 in. high, painted with a dull paint. It is mounted in a rack such that the total surface area is exposed. The average surface temperature is 90°C and the objects surrounding the box are at 60°C. Also, the surfaces surrounding the box are in relatively close proximity. Estimate the radiant energy dissipated from the box. Solution: A_1 = 360 sq. in. Assume that the emissivity of the surrounding objects is 0.90. From Appendix the emissivity of the surface of the box is 0.94. Since the surrounding objects are in close proximity to the box, F_e is calculated by

$$F_e = \frac{1}{\frac{1}{0.94} + \frac{1}{0.90} - 1} = 0.852$$

F_a = 1.0
T_1 = 90 + 273 = 363°K
T_2 = 60 + 273 = 333°K
$q_r = 0.0037\,(360)\,(0.852)\,(1.0)\left[\left(\frac{363}{100}\right)^4 - \left(\frac{333}{100}\right)^4\right]$
q_r = 58 watts total radiant heat dissipation

An alternative solution is by use of the radiation chart, Figure 11. Compute F_e as above

F_e = .852
Draw a line from .85 to 90°C = 0.54 watts/sq. in.
Draw a line from .85 to 60°C = 0.38 watts/sq. in.
The difference is .54 - .38 = .16 watts/sq. in.
0.16 x 360 = 58 watts

Example 8-6: Comparison of radiation with free convection. To show that radiation is an important mode of heat transfer, it is of interest to compare it with free convection. In the foregoing problem, free convection acts simultaneously with radiation. Assume that the surrounding air is at 60°C. The following tabulated calculations for the free convective heat transfer are from the free convection chart, Figure 10.

Free Convection From Box of Example 8-5

Surface	Significant Dimensions, in.	Area Sq. in.		Watts/ Sq. in.		Total Watts
Top	$\frac{12 \times 6}{12 + 6} = 4$	72	x	0.14	=	10.1
Bottom	$\frac{12 \times 6}{12 + 6} = 4$	72	x	0.07	=	5.0
Sides	Height = 6	216	x	0.10	=	21.6
				Total	=	36.7

Hence, in this case, the total dissipation from the box is 58 (radiation) plus 37 (convection), or 95 watts, of which 61 percent is by radiation.

Radiation and convection usually occur simultaneously, except in a space or vacuum environment. Since convective heat transfer is usually described by a convective coefficient of heat transfer, h_c, it is convenient to use an equivalent coefficient of radiation, h_r, so that the two coefficients are additive. Thus, for a surface temperature T_s and of area A transferring heat by convection to the surrounding air at temperature T_a and simultaneously transferring heat by radiation to a radiant receiver surface at temperature T_r, the total heat transfer rate from the surface by these two modes is:

$$q_t = q_c + q_r = (h_c + h_r)\,A\,(T_s - T_a) \qquad (8\text{-}37)$$

The radiation coefficient is:

$$h_r = \frac{\sigma_s F_e F_a \,(T_s{}^4 - T_r{}^4)}{T_s - T_a} \qquad (8\text{-}38)$$

If the surrounding receiver surface temperature is equal to the air temperature, T_a is substituted for T_r.

8.1.3.3 Comparison of conduction with radiation. Heat transfer by conduction can frequently exceed that by radiation. The predominant influence of conduction under favorable conditions can be illustrated by quantitative examples. First, it will be noted that soft copper with a uniform cross section of 1 sq. in. has a thermal resistance of 0.1°C/watt per in. of length.

Therefore, a unit heat dissipation of 1 watt/in.^{2} at one face of a one inch cube causes a temperature rise of only 0.1°C, above the temperature of an opposite face. However, if the cube is insulated from other metal to avoid conduction and is placed in a vacuum or dead-air space to avoid convection, it can transfer heat only by radiation. Then, even though the surface is blackened to provide an emissivity approaching unity, its temperature will rise at least 130°C above a black body environment at 30°C. This particular temperature-dependent radiation resistance is 1300 times the thermal resistance to conduction of the 1-inch cube; hence, it appears that radiation is negligible when conduction is effective. A similar conclusion regarding convection can be reached, in view of the fact that convection is about as effective as radiation at moderate temperatures. Finally, it can be concluded that the effective thermal resistance between a metal-connected heat source and sink is practically equal to the conduction resistance because the convection and radiation resistances, which are virtually in parallel, are both very much higher.

8.1.3.4 Design notes on radiation. Even though the thermal circuits in densely packaged electronic equipment are complex, approximate radiation calculations may be made. There are several design principles which may be used to advantage and which should be kept in mind:

a. At high temperatures the heat transferred by radiation may exceed that transferred by convection. Thus, radiation can be an important mode of heat transfer.
b. For maximum heat transfer by radiation, "black" surfaces must be used. This should not be interpreted to mean that all surfaces

should be painted black because the emissivity of glass at temperatures as low as 100°C is equal to that of the best black paint and with increasing temperatures becomes slightly greater than that of the paint.

c. For a given difference between radiating and receiving surface temperatures, the higher the level of temperature, the greater will be the radiant heat dissipation. This situation is, of course, expressed by the radiation law that the heat transferred is proportional to the difference in the fourth powers of the absolute temperatures.

d. Uncontrolled radiation can cause impaired reliability. It is desirable to protect temperature-sensitive, or low-rated-temperature parts, from overheating, due to their proximity to higher temperature heat sources. Hence, low temperature parts must be located so that they do not "see" these sources or radiation shields must be used. Highly polished sheet metal shields placed between such parts can be very effective as radiant heat barriers. The shield should be polished on both sides. Because perfect reflectors do not exist, it is desirable to thermally bond the shield to the case or chassis to provide a good conductive heat path. One of the best reflectors of long infrared radiation is polished gold, with a reflectivity of the order of 98% at 5 microns. Polished silver and aluminum are good reflectors. Chromium is less efficient because of certain absorption bands in the infrared spectrum. If a chromium surface must be used, it is recommended that chrome aluminum be utilized rather than pure chromium. All reflectors must have polished and smooth outer surfaces. The reflectivity of such coatings is very low (high emissivity) and the efficiency of the reflecting surface will be impaired.

e. Placement of parts in an electronic assembly to provide maximum radiant heat dissipation requires careful consideration. For example, a tube surrounded by other tubes could dissipate little radiant heat with a consequent higher temperature rise than its neighbor. There may be occasions due to assembly limitations wherein a certain part must be given special treatment to provide adequate cooling.

8.2 Natural methods of cooling electronic parts. Note: The thermal limitations of electronic parts are given in chapter 14.)

8.2.1 Semiconductor devices. Individual solid state devices have such small areas that natural convection and radiation from them are relatively ineffective. For a dissipation of a few watts, the surface may be increased by applying a so-called "heat sink" or extended surface. Several manufacturers produce standard forms of heat sinks and supply design information on their installation and capabilities.

Since solid state circuitry is closely packed, heat interchange among the components is very important. Cooling calculations applied to individual parts are not strictly applicable to the assembly. This applies particularly to convection and radiation.

Conduction is the most useful means of heat transfer for solid state equipment. The electrical leads conduct heat directly into a P.C.B. or the equivalent, which serves as a spreading resistance to convey the heat to a large dissipating surface. Also, heat conducting studs and cases which are thermally connected to the internal junctions may be heat sinked to the chassis or to other heat dissipating surfaces. In general, the thermal design should make maximum use of conduction. Figure 12 shows a thermal circuit for a low power transistor.

C = thermal capacitance
T_j = junction temperature
T_E = environmental temperature (usually air)
T_C = case temperature
T_S = sink temperature
R_{jc} = resistance, junction to case
R_L = resistance of leads
R_1 = resistance due to free convection and radiation from case
R_2 = contact resistance case to sink
R_3 = resistance due to convection from sink

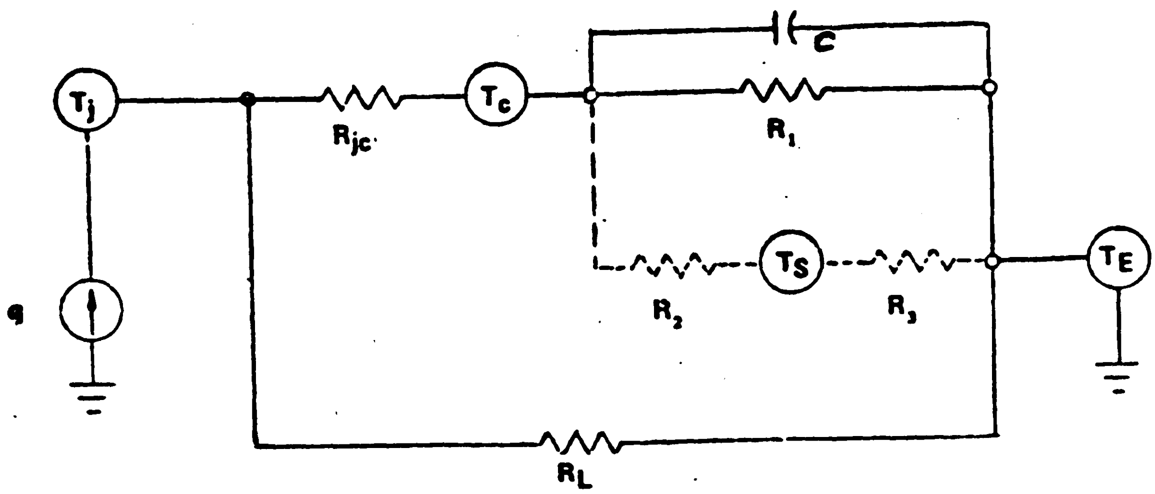


FIGURE 12. Transistor Thermal Circuit

In order to maintain T_j below the required maximum, with average dissipation of q watts and environmental temperature T_E given, it is required to satisfy the condition that $(T_j - T_E)/q$ is equal to or less than the resultant thermal resistance. The internal resistance R_{jc} is fixed for a particular transistor.

Manufacturers customarily give a power derating factor for small signal transistors, and a value of junction to case thermal resistance for ratings above 0.5 watt. The reciprocal of the derating factor is the thermal resistance from junction to a free air environment at sea level pressure. The value ranges from 175 to 1000°C/watt. The junction to case resistance, when given, ranges from 50 to 250°C/watt. The resistance due to the leads is difficult to estimate, and varies widely depending on the thermal resistance of the wiring complex to which the transistor is connected and the length of the transistor leads. In general R_L will be larger than R_{jc}, perhaps 2 or 3 times as much. R_l is due to radiation and free convection in parallel. In close packed circuitry radiation is a minor factor because the transistor "sees" warm surfaces. The derating factor does not include radiation because of the test method by which it is determined. It is advisable to neglect radiation, and to make an estimate of conduction through the wiring, i.e., R_L, in detailed thermal design.

Free convection cooling can be computed from equations or the nomograph given in paragraph 8.1.2.

The only satisfactory means available for the enhancement of natural cooling of low power transistors is the provision of some parallel heat flow path from case to environment. The most commonly used devices are small sheet metal parts having one or more fins and spring fingers which tightly grip the case. Several manufacturers supply these devices and publish heat transfer data for them. Use of such sinks can increase the allowable power dissipation by a factor of 2 or 3, if the internal resistance is sufficiently low.

Various forms of thermal retainers are also available which are secured to a sink by threaded studs, solder, or cement. Case to sink resistance of 4 to 25°C/w are achieved, depending on the design (Reference 10). Several styles are available from various manufacturers.

When transistors are mounted on a metal structure capable of conducting and dissipating heat, conducting caps can be fitted to the case as shown in Figure 13. It has been shown that such a cap can effectively reduce the external thermal resistance to a negligible value (Reference 11).

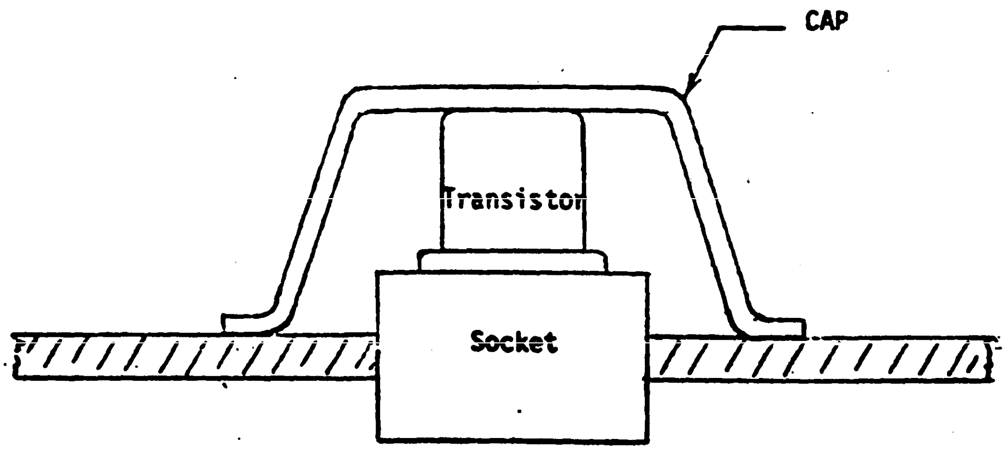


FIGURE 13. Heat Sinks For Low Power Transistors

Obviously this device cannot be used with any transistor in which one of the junctions is electrically connected to the case, unless suitable electrical isolation is provided.

The internal junction to case thermal resistance ultimately limits the allowable dissipation. The resistance representing free convection and radiation from a TO - 5 case varies from 100 to 200°C/watt depending on the temperature difference between the case and the environment. Since this is the same order of magnitude as the internal resistance of small signal type transistors, dissipation can be increased (or temperature reduced) by a factor of two or three at most, no matter what devices or cooling methods are applied.

8.2.1.1 Power transistors. Transistors are categorized with respect to power rating as either "small signal" or "power." They are also categorized as to use; i.e., "switching," "RF power," "general purposes," "entertainment," "economy," etc.

With regard to thermal characteristics there are no sharp distinctions, but only a trend toward lower junction to case resistance with increased power rating. Equipment thermal design is expedited by using the thermal data now furnished by manufacturers.

Power transistors in general have large, flat mounting surfaces, with provision for attachment to a heat sink by screws or by a threaded heat conducting stud. Internal thermal resistance is low, one or two °C/w. The case is connected electrically to the collector, and the mounting design must provide electrical isolation. Figure 14 shows the essential parameters involved in natural cooling of power transistors under steady state conditions.

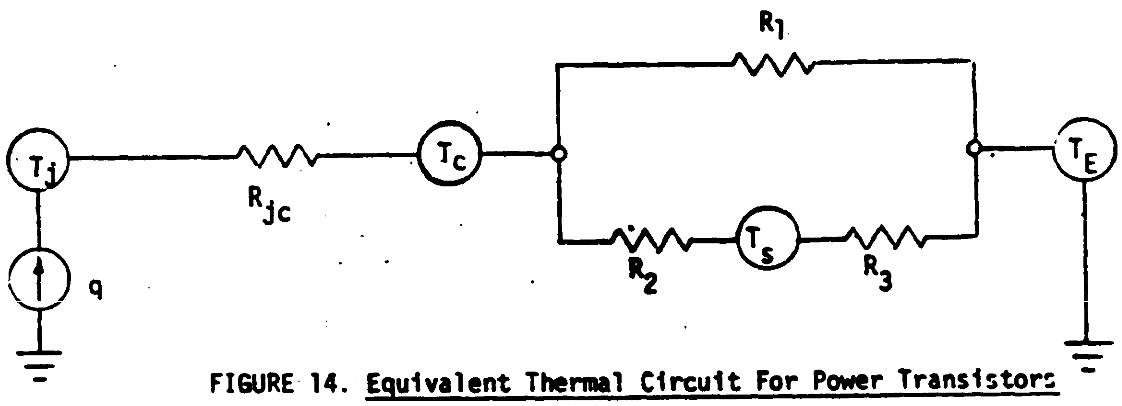


FIGURE 14. Equivalent Thermal Circuit For Power Transistors

R_1 = convection and radiation from case
R_2 = case - sink interface contact resistance
R_3 = convection and radiation from sink

Heat conduction through the leads and wiring is omitted since it has little effect. Manufacturers' thermal data include the internal resistance to environment resistance R_1. Allowable dissipation is also given for stated case temperatures. The reciprocal of the derating factor is the thermal resistance from junction to environment.

Manufacturers furnish mounting hardware for the several types of cases, and publish the effective thermal interface resistance between case and sink, usually for stated torgue or contact pressure. The contact resistance varies with the surface flatness and the torque applied to tighten the screws. These variations are greatly reduced by applying silicone compounds to the mating surfaces.

Special silicone greases are now available for use in minimizing the contact resistance. (References 10 and 38) Dow Corning 340 heat sink compound is a grease-like silicone material, heavily filled with heat conductive metal oxides. It is claimed that this combination provides high thermal conductivity, low bleed, and high temperature stability. The thermal conductivity is three times greater than that of ordinary silicone grease. The resulting compound will not dry out, harden, or melt, even after long term exposure to temperatures up to 200°C. This compound does not require replacement during the normal life of military electronic equipment. Should a solid state device later need replacement, examination will reveal the quantity of compound remaining on the surfaces. Additional compound may or may not be added dependent upon the quantity remaining.

It is important to note that DC340 will slightly outgas under high vacuum conditions. Thus, for space applications, controlled volatility compounds are required. Such compounds have been developed for NASA and are available.

After selections of transistor to be used, the only part of the heat flow path over which the designer has control is that through the mounting interface to some form of intermediate heat sink. This path includes a series contact resistance.

Currently, insulating washers made of Teflon, anodized aluminum, mica, and beryllium oxide are commercially available. The choice of washer material depends on fragility, ease of assembly, cost, and manufacturing methods. Flexible materials such as Teflon and mica partly compensate for surface imperfections. However, the heat sink surface should always be reasonably flat and smooth. All burrs from drilling and tapping must be removed. Beryllium oxide is brittle and anodized aluminum is hard so that very flat mounting surfaces are required. Table VI gives typical values of thermal resistance from case to sink for washers applied to TO - 3 and TO - 66.

8.2.1.2 Integrated circuits. With regard to natural cooling methods, IC's and multiple solid state devices are similar to small signal and low power transistors. The significant differences are the larger number of leads and the larger surface available for free convection, particularly for "flatpacks." Details on the configurations and thermal characteristics of the many various types of integrated circuits are given in chapter 9.

Manufacturers publish maximum junction temperatures, allowable power dissipations, and derating factors. Heat sinks are commercially available in designs to fit standard IC packages. As an example of achievable cooling, (Reference 13) a typical flat pack dissipated 1.5 watts at 155°C junction and 125°C case temperature with natural convection in free air at 25°C, when mounted on an epoxy board with a suitable heat dissipating clip.

TABLE VI. Typical Case-Sink Thermal Interface Resistance

Material & Source	Thermal Resistance		°C/W
Motorola (Ref. 12)	dry	w/Dow Corning No. 4	W/Wakefield 120 Compound
No washer	0.20		
3 mil Teflon	1.45	0.80	-
2 mil Mica	0.80	0.40	-
20 mil anodized Al	0.40	0.35	-
Wakefield (Ref. 10)			
45 mil anodized Al	1 to 1.4	-	0.3
62 mil Be O	0.4 to 1.0	-	0.14

Note: DC-340 data not listed above

As with individual transistors, the allowable dissipation is limited by the internal thermal resistance and the available environmental temperature. The use of intermediate sinks of any type can increase the allowable dissipation by a factor of two or three at most, when natural cooling methods are used. Any further increase requires a lowering of environmental temperature by some external means.

8.2.1.3 Large scale integrated circuits. LSI devices under development range in size and power dissipation from somewhat enhanced integrated circuits to single packages three (3) inches in diameter dissipating 20 to 25 watts. The thermal design for natural cooling of the lower dissipation units can be the same as for conventional integrated circuits. Because of packaging and space limitations, the largest devices are usually cooled by forced air or liquid cooling. However, if desired they can be cooled by natual methods using large finned heat sinks similar to those used to cool power transistors and discussed later in this chapter.

8.2.1.4 Rectifiers and diodes. In general thermal designs for rectifiers and diodes are similar to those for transistors. The case sizes and mounting configurations are similar dissipation wise. The principle difference is that significant voltage may be applied to rectifier diodes and the usual mica or BeO washer used with transistors and heat sinks may not provide adequate electrical isulation. Unfortunately, the required increased electrical insulation results in increased thermal resistance, which must be considered in the thermal design. Alternatively, a diode can be directly bolted to an electrically isolated heat sink resulting in a reduced interface thermal resistance. The heat sink will be at the same electrical potential as the diode, however, and the possibility of electrical shock hazard to maintenance personnel must be taken into account.

Large high power dissipation SCR's usually are cooled by means other than natural cooling.

8.2.1.5 Microwave semiconductor devices. The manufacturers of Gunn Effect devices, LSA-Gallium arsenide, and varactor diode devices usually package the device in a special cavity or other housing which provides a low thermal resistance to the surface of the housing case. The manufacturers expend a major effort in reducing the thermal resistance local to the active device elements to a minimum. In fact, the successful operation of these devices often depends on control of the local thermal resistance. Several manufacturers plan to use diamonds to mount the semiconductor elements, since diamond offers the lowest obtainable thermal resistance. (Approximately 15 percent greater thermal conductivity than silver.)

The total dissipations (except for varactors) seldom exceed 10 watts. However, extreme care must be used to reduce the temperature of the outer surface of the housing. These devices are extremely sensitive to temperature and only a few degrees rise in case temperature above "ambient" should be permitted.

8.2.2 Electron Tubes

8.2.2.1 General. Because electron tubes operate at much higher temperatures than most other electronic parts, the heat from tubes can damage adjacent parts such as semiconductor devices. Care must be exercised in the thermal design of hybrid equipment using both tubes and transistors. Thermal isolation methods are discussed later in this chapter.

8.2.2.2 Unshielded receiving tubes. The major mode of heat transfer from a bare electron tube in free air is radiation. A significant portion of the heat is removed by convection and less than five percent is removed by conduction. If a tube is surrounded by lower temperature surfaces which are at distances greater than one inch from the tube, increased natural convection will occur. Even so, the heat transferred to these surfaces by convection will be less than that transferred by radiation. If the surrounding surfaces are very close, say, less than one-half inch away, and if the tube is enclosed in a air-tight container, then free convection becomes ineffective and heat will be tranferred by gaseous conduction, but still to a smaller degree than that by radiation.

Receiving type electron tubes larger than the subminiature tubes can be cooled satisfactorily without shields if the environmental temperature is relatively low, if adequate convection is provided, and if the tubes can "see" cooler surfaces (radiation). Convection cooling can be enhanced by incorporating "chimney" effect devices to guide the air flow. Unshielded tubes should not be mounted in closely spaced groups. At best, unshielded tubes should only be mounted in a single line spaced at least 1 1/2 bulb diameters on centers. If tubes must be mounted in groups, then each should be mounted within a blackened "chimney" which is thermally bonded to the chassis. Actually, of course, this constitutes a crude form of tube shield.

8.2.2.3 Shields for cooling receiving tubes. An acceptable tube shield should provide adequate cooling and electrical shielding. It should support the tube securely in its socket against vibration and impact in any plane, and it should protect the tube and its leads from mechanical injury.

A thermally satisfactory shield must absorb the radiation from the tube envelope and plate, and in addition, remove heat from the envelope by conduction. The shield should fit the tube envelope as tightly as possible to reduce the air gap to a minimum. Perfect contact with the bulb glass is difficult to achieve. In general, the most practical method is to provide some flexibility in the shielf to accommodate expansion and variation in bulb dimensions. For example, this can be accomplished by slotting or splitting the shield or incorporating corrugated inserts.

Further, it is necessary to increase the absorptivity of the inner surface of the shield to increase the heat transfer by radiation from the envelope. A brightly polished surface is a poor radiation absorber and should not be used. A dull, oxidized and blackened surface is preferred.

The best tube shield will be useless unless its absorbed heat is removed, preferably by conduction. In removing heat from the shield, the first and most important consideration is to provide a minimum of thermal contact resistance between the tube shield, its base and the chassis or mounting surface.

The mounting surface or chassis should be of metal with a heat flow path of low resistance to a local sink. Nothing is gained by mounting tube shields on, for example, a plastic chassis, or on materials of low thermal conductivity.

8.2.2.4 Miniature tube shields. The old standard "JAN miniature tube shield" is not satisfactory from the heat transfer standpoint, especially at high heat concentrations. Such a shield is usually a heat barrier. The blanket of air enclosed between the shield and the tube envelope is too thin for free convection currents to form and heat transfer from the tube to the shield is possible only by gaseous conduction and radiation. Due to the low emissivity (high reflectivity) of the brightly finished internal surface, a large portion of the radiation is reflected back to the tube rather than being absorbed. Further, the contact resistance between the shield and its base is high and heat transfer to the chassis is impeded.

MIL approved conduction tube shields, similar to those made by IERC should be used. These shields have flexible inner contact fingers or clips in good thermal contact with the envelope and have a special low thermal resistance joint between the shield and base.

8.2.2.5 Glass transmitting tubes. Low and medium power transmitting tubes with glass envelopes are usually operated without shields. There are two basic thermal classes of these tubes i.e.: those with relatively low temperature anodes, and those with high temperature anodes. The low temperature anodes operate at temperatures of the order of 400°C and are made from blackened metal or carbon. The high temperature anodes operate at a dull or cherry red temperature and radiate the heat from the anode directly through the glass envelope which is transparent to medium wavelength infrared radiation. With the lower temperature anodes, the radiation from the anode is at wavelengths where the glass is not transparent. Consequently, most of the heat from the anode is absorbed by the glass which reradiates and convects the heat to the surrounding air at a much lower temperature.

Thermal designs for equipments using the higher temperature direct radiation cooled tubes should include radiation shields or other features to protect from the intense radiation, all electronic parts which a tube can "see." The shields or surfaces intercepting the radiation must have a low thermal resistance to the local heat sink to maintain their surfaces at a low temperature. This will prevent significant reradiation and uncontrolled scattering of the heat into other portions of the equipment. Also, the tube temperatures will be reduced. It is important to note that over 75 percent of the dissipated heat will be radiated and the thermal design of the surfaces surrounding the tube must accommodate this added heat. It can be proportioned based on the radiation theory of this chapter.

The thermal design of equipments using the lower temperature anodes must be treated similar to the high temperature anode tubes, except that the radiation is reduced (as determined by the envelope temperature) and the cooling by convection will be much increased. Because the dissipation of the tubes is relatively large, care must be used to make sure that the hot convected air does not heat parts mounted above the tubes.

8.2.2.6 Conduction cooled transmitting tubes. Conduction cooled external anode transmitting tubes dissipating up to 300 watts are now available. These tubes incorporate a heat conducting electrical insulator (usually Beryllium Oxide) which is fused to the outer surface of the anode and to the inner surface of a heat conducting pad. The pad has a relatively large flat surface for attachment to a heat sink or to a conductive heat flow path to a local heat sink. The tube manufacturers provide thermal ratings for these tubes and their heat conducting pads. It is important to note that these tubes dissipate significant quantities of heat directly to the point of attachment of the pad. Thus, the heat flow path to the sink should have a very low thermal resistance or else an independent heat flow path should be provided so that other connected parts thermally are not overheated. Interface contact resistance is discussed later in this chapter.

8.2.2.7 Microwave tubes. Care must be used in the thermal design for cooling special microwave tubes such as Klystrons, Traveling Wave Tubes (TWT's) and Magnetrons. Low power Klystrons obtain appreciable heat transfer from their outer surfaces into the associated wave guides when directly connected to plumbing. Usually, small receiving type Klystrons are conduction cooled by their waveguides and other heat sinking is not required. TWT's and magnetrons typically require conduction cooling either through their mounting pads or by means of metal grounding straps into a heat flow path. When the total dissipation for a microwave tube approaches 100 watts, natural cooling is usually inadequate and other cooling techniques are required. Tube manufacturers ratings differ and the thermal requirements provided by the manufacturer for each specific tube should be met as a minimum.

Note on field mapping and relaxation techniques. In a laminar flow field the flow rate density and the pressure drop are orthogonal functions. In a two dimensional flow pattern the flux lines and the traces of the equal pressure surfaces intersect at right angles. That is, the tangents to these two sets of lines are always perpendicular to each other. Since the pressure lines are perpendicular, and the flow lines are parallel to the solid boundaries of the field, it is relatively easy to sketch a field map for any flow channel of almost any shape. The total flow for a given total pressure drop is the ratio of the number of flow tubes to the number of pressure zones.

This technique is strictly applicable to two dimensional flow. Three dimensional configurations can be treated by making two flow maps, one for each of two sections, the x-y and x-z planes for example.

A more exact but more complicated technique is the numerical one of relaxation. This method is extremely tedious for anyone, but the most simple configurations can be handled easily with a computer to any desired degree of precision.

8.2.3 Resistors. Resistors are represented thermally by the same circuit as transistors, with internal, "hot spot" temperature in place of junction temperature. There is generally no way in which hot spot temperature can be measured, except in the case of open wire wound types. Therefore, the surface temperature is the only index of thermal condition.

The only limiting factors on power dissipation are voltage, and internal temperature high enough to cause resistance change beyond tolerance or catastrophic failure. A resistor will continue to act as such until it burns or melts. Almost any power can be dissipated in a resistor, provided only that the internal element temperature is sufficiently low and voltage is below the breakdown value.

As an example of natural cooling of resistors, laboratory measurements have shown that the average 1/2 watt composition carbon resistor in conventional equipment will reject approximately 40 percent of its heat by free convection, 10 percent by radiation, and 50 percent by conduction through the leads. It will exhibit a difference in surface temperature between center and the ends because of heat transfer through the leads. The center point may be as much as 40°C hotter than the ends.

Calculated values of free convection cooling at 40°C ambient from the resistor body, for MIL-R-11 fixed composition resistors, indicate the following values of thermal resistance: 1/2 watt, 200°C/watt; 1 watt, 100°C/watt; 2 watt, 58°C/watt. The values are based on average surface temperature. Thermal conductance of a lead 1/2 inch long is 0.03 to 0.02 watt/°C, depending on the rating. Since heat flows through the leads into the spreading resistance of the wiring, the cooling effect of the leads is difficult to estimate. Several installations which have been studied by calculation indicate an overall thermal resistance through the leads to the environment of approximately 200°C/watt, so that this path accounts for a substantial fraction of the heat dissipation.

The industry has established the power dissipation ratings of resistors on the basis of ambient temperature, and this method is used in nearly all of the current applicable Military Specifications. Resistors are categorized by material and construction, and for each type curves of dissipation vs. ambient temperature have been established by testing in free air. These curves show a constant dissipation up to a certain temperature and a linear derating thence, down to zero at some higher temperature. Failure rates are also given vs. ambient temperature, for different values of actual/rated wattage ratio, and factors are given for different applications such as ground, airborne, missile, etc.

The derating curve may be used to determine a value of thermal resistance from the surface to environment. For example, fixed composition resistors, per MIL-R-11, dissipate their full rating up to 70°C ambient, and are derated to zero at 130°C. For a 1/2 watt resistor, $R = (130-70)/0.5 = 120°C/watt$. For a 2 watt resistor, $R = 30°C/watt$.

The ambient temperature for 100% derating may be taken as the maximum allowable surface temperature. The temperature difference for a given power dissipation is computed by the methods given in paragraph 5.1, considering the leads as well as the body. For example, a MIL-R-11 resistor of 1/2 watt rating, with leads 1/2 inch long, placed horizontally in free air, dissipates 1/4 watt at a temperature rise of 55°C. Assuming the same amount of dissipation through the leads and wiring, the surface temperature in a 50°C environment is 105°C at a total dissipation of 1/2 watt, well below the maximum of 130°C.

There is not much that an equipment designer can do to increase the cooling of resistors by radiation and natural convection other than to reduce the environmental temperature. In close packed equipment radiation is best neglected. The conduction cooling of resistors can, however, be greatly increased. With small resistors, as noted above, considerable conduction cooling can take place through the leads. Further, with a 1/2 watt resistor, it has been found that a 36° rise above ambient temperature was obtained with zero length leads connected to a sink (Reference 14). With leads one inch long, the rise was 51°C. Correspondingly, the lead length had a greater effect when larger resistors were tested. Therefore, it is suggested that larger diameter leads be used with resistors and that the lead length be minimized. If possible, the leads should be conduction cooled.

Currently available resistors can best be cooled by metallic conduction from the resistor body to a metallic chassis or sink. Fuse-clip type mounts are excellent. Also, clamping to a metal chassis has been found to be very effective. The width of the clamp is not as important as the fact that, by clamping, the resistor body is in intimate contact with the chassis. Resistors are available with wraparound metal strips designed to be clamped to metal mounting surfaces by screws. Typically, a resistor which is rated at 3 watts dissipation with conduction cooling, is a conventional 1 watt resistor. A conduction cooled resistor rated at 4 watts dissipation, is normally rated at 2 watts in free air.

Variable resistors, rheostats, and potentiometers present special design problems because the effective surface for convection and radiation varies with the values of resistance. Careful attention must be paid to the rating, and to the current value expected over the range of resistance settings. Failure of a unit which is apparently amply rated is quite possible at low resistance setting.

8.2.4 Transformers and reactors. Iron core transformers can contribute significant heat into electronic equipment dependent upon their efficiency. Unfortunately, small transformers usually operate at lower efficiencies than larger transformers and, consequently, transformers in miniaturized equipment are frequently important heat sources. The heating of transformers is due to hysteresis and eddy current losses in the core and copper loss in the windings. Chokes and inductors usually have low core losses since they operate at relatively low flux density.

The thermal limitations of inductors are fixed by the temperature tolerance of the insulation and the Curie temperature of the core material. Insulation gradually deteriorates at high temperature and most transformer failures are short circuits through weakened and embrittled insulation after long service and the unavoidable temperature cycling.

The Curie points of magnetic steels run from about 700 to 900°C, far higher than tolerable for insulation. Ceramic ferrite cores have Curie points as low as 300°C. Insulation is classed by allowable temperatures up to 500°C.

A transformer rating normally gives a "maximum operating temperature" and a "temperature rise above ambient," at rated operating conditions. Failure rates are given for maximum operating temperature. A simple thermal circuit representation is a single resistance between heat source and environment, as shown in Figure 15.

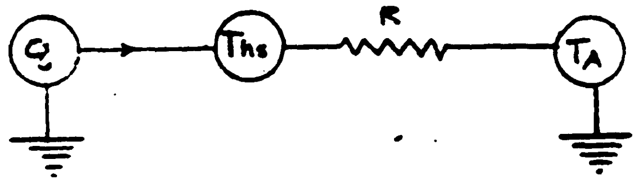


FIGURE 15. Elementary Thermal Circuit of an Inductor

T_{hs} = hot spot temperature
T_A = ambient or free air temperature
q = power loss = actual volt amperes x efficiency

T_{hs} is found by adding the specified temperature rise to the known or assumed value of T_A. R is found by dividing the specified temperature rise by the value of q for rated full load. This simple representation is useful when studying a large assembly of parts, when for example, the ambient temperature at the transformer depends on the entire thermal complex. However, it does not present a picture of the actual heat transfer in the part. Further, ambient temperature is a poorly defined quantity.

The most important temperature in a transformer is the coil hot-spot temperature. The largest temperature rise is usually between the coil hot-spot and the core. If the maximum allowable coil hot-spot temperature is exceeded, the coil insulation is in danger of breakdown with subsequent failure of the transformer and possible damage to the connected equipment. Therefore, the actual hot-spot temperature must be predicted with the best possible accuracy.

Figure 16 presents a more complete and accurate view of the thermal situation and is useful in assessing the effect of steps which may be taken to enhance cooling and lower internal temperature, (or increase power capacity). Convection and radiation from the case can be estimated by the methods given in paragraph 8.1.

The characteristic dimension L used in both the Grashoff and Nusselt numbers must be considered. A transformer for example, may not have a simple shape surface unless it is encased in a cylindrical or rectangular container. Fortunately, most transformers are not long in any one dimension and as an approximation may be considered to be a sphere of equivalent surface area. The conduction path between case and environment is chiefly a contact, and can be estimated as such. The parallel value of these three resistances; subtracted from the total resistance in the elementary diagram, yield a value of internal resistance.

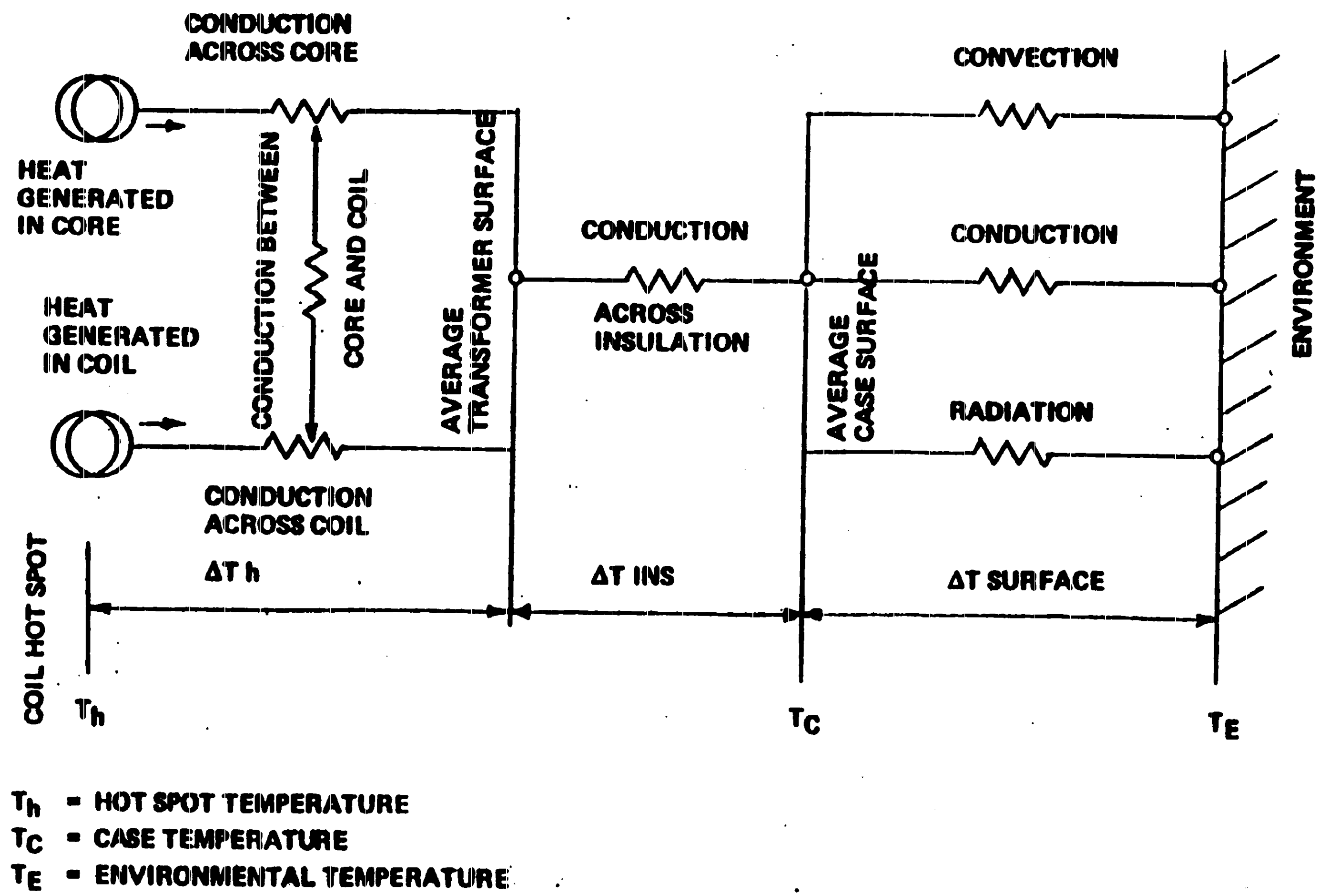


FIGURE 16. Heat Flow Analogue of a Transformer

Once a transformer has been selected, all that the equipment designer can do to enhance natural cooling is to improve the conductive heat flow paths. The leads should be large and should be as well bonded thermally to the mounting structure as possible. The mounting surfaces should be smooth and flat, and the hold down screws properly tightened. Aluminum foil in the mounting contact will improve heat conduction. If the transformer contains electrical shields they should be thermally bonded to the chassis if possible. Copper straps secured to the case or core and the chassis will help. Minor improvement in cooling can be made by attaching convective fins to the unit, but conductive heat transfer is more effective than free convection.

Since there is a large uncertainty in estimating the various thermal resistances, resort should be made to testing when the situation appears to be marginal. Measurements of case temperature will indicate the effectiveness of improvements. Case temperature plus rated temperature rise gives a practical, probably conservative value for actual hot spot temperature.

Air core reactors are not standardized to the same extent as solid core types. Most air core coils do not present a severe thermal problem since the currents and dissipations are small. Tuning and loading coils for high power VLF transmitters are one notable exception, since they may dissipate a kilowatt or more. Natural cooling of air core coils is chiefly by convection. Close wound coils should be treated as smooth cylinders, with free convection cooling on both outer and inner surfaces when placed vertically and on the outer surface only when horizontal. Open wound coils present nearly all of the conductor surface to free convection, but the local heat transfer coefficient varies widely. In the rare event that free convection on an air coil inductor need be estimated, it is reasonable to use the constants specified for cylinders and use one half of the total conductor surface.

8.2.5 Passive parts. Passive parts are those which do not contain heat sources. This includes capacitors, fuses, circuit breakers, switches, connectors, and structural elements. Relays are also passive unless they carry current for hold on.

Although not in themselves heat sources, passive parts are subject to degradation and failure due to excessive temperature. They can receive heat from neighboring active parts by all three natural modes, and it is quite possible for their temperature to rise above the sink temperature. In any thermal design it is necessary to examine the heat transfer to and the resulting temperature of these parts in order to estimate their failure rates. The design must be such as to assure a failure rate no higher than that required by the reliability design.

Capacitors are particularly temperature sensitive, the Q value decreasing as temperature rises. Fuses decrease in interrupting current value with increasing temperature. Temperature sensors must obviously be installed so that they detect the temperature desired and do not give unwanted or false readings. All passive circuit elements should be thermally isolated from heat sources, to protect them against excessive temperature and to prevent them from acting as unwanted heat flow paths.

8.2.6 Thermal interaction. Since heat always travels "downhill" there is a continuous temperature gradient throughout any operating electronic equipment. There is a hottest location and a coolest location. One object of thermal design is to control the temperature gradient and the position of each part on it.

It is obvious that heat sources are the hotter points, and that air or any other coolant in free convection increases in temperature as it passes through the package. The air temperature rise depends on the rate at which cooling is effected by free convection. If free convection is the major mode of heat removal, the air temperature will have a relatively steep gradient in the vertical direction.

The pertinent temperatures and heat flows associated with an electronic part are indicated in Figure 17.

The total heat flow into the part is zero:

$$q_p + \sum_{i=i}^{m} \frac{T_{ci} - T_p}{R_{condi}} + \sum_{j=1}^{n} \frac{T_{rj} - T_p}{R_{radj}} + \frac{T_{air} - T_p}{R_{conv}} = 0 \qquad (8\text{-}39)$$

Equation 8-39 gives a crudely quantitative picture of thermal interaction, which is a very complicated and involved matter.

Of the three types of thermal resistances, only R_{cond} is relatively constant and readily calculable. R_{conv} varies inversely as the 1/4 power of the temperature difference and therefore, may be considered constant as a first approximation. R_{rad} can be seen to vary inversely with a function of the two temperatures which may be written as $(T_{rad} + T_p)\,(T^2_{rad} + T_p^2)$. Thus, R_{rad} decreases rapidly as all temperatures increase. However, as a practical matter in electronic equipment, radiation and free convection heat transfer are usually the same order of magnitude. Exceptions to this rule occur only in the line of sight of very hot surfaces of high emmisivity.

Substituting thermal conductances (G) for thermal resistances (R), and solving Equation 8-39 for T_p, Equation 8-40 is derived.

$$T_p = \frac{q_p + T_{air}\,G_{conv} + \sum T_{ci} G_{condi} + \sum T_{rj} G_{radj}}{G_{conv} + \sum G_{condi} + \sum G_{radj}} \qquad (8\text{-}40)$$

This equation shows that the temperature of a part is a function of the temperatures of all the other parts which are thermally connected to it.

Several approximations can be made to simplify the equation and clarify the problem of interaction: (1) The denominator is the sum of all thermal conductances connected to the part. Radiation conductances are of the same order of magnitude as convection conductance. Further, in close packed equipment there are only a few high temperature bodies in line of sight of the part. Therefore, it is reasonable to neglect the term G_{radj} in the denominator. (2) If conductive paths to the sink are of low resistance according to recommended design practice, convection becomes relatively unimportant. Then:

$$T_p = \frac{q_p + T_{air}G_{conv} + T_oG_{cond}}{G_{cond}} \qquad (8\text{-}41)$$

$$= \frac{q_p}{G_{cond}} + T_{air}\frac{G_{conv}}{G_{cond}} + T_o$$

where T_o = sink temperature
G_{cond} = total thermal conductance to the sink.

Thus, air temperature has a minor effect and part temperature is minimized. (3) For a passive part $q_p = 0$, the part temperature approaches the sink temperature as conduction paths to the sink are improved.

$$T_p = T_o + T_{air}\frac{G_{conv}}{G_{cond}} \qquad (8\text{-}42)$$

(4) A passive part which is thermally isolated by shielding it from radiation and minimizing conductive connections is approximately at the temperature of the surrounding air.

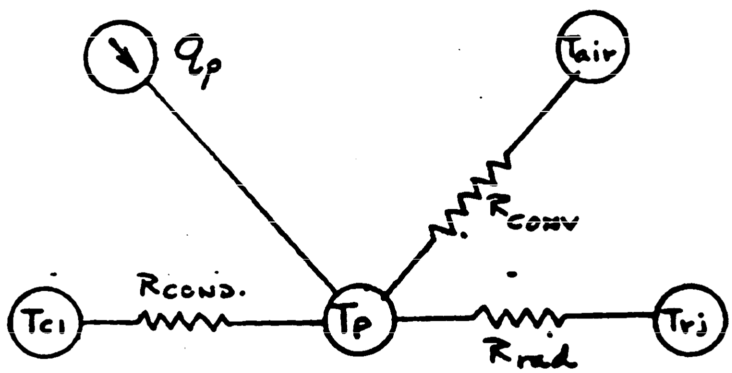


FIGURE 17. Heat Flow Diagram for an Electronic Part

T_p = part surface temperature
T_{air} = local air temperature
T_{ci} = temperature of the i th conductively coupled heat source (or sink)
q_p = internally generated heat
R's = equivalent thermal coupling resistances

8.2.7 Thermal isolation and protection. Thermal interaction, if not controlled, can result in excessive parts temperatures. Passive, temperature sensitive parts in particular require thermal protection. Active parts may require derating. See for example, the discussion of resistors mounted in groups, paragraph 8.3.1.1.

Thermal isolation is achieved by connecting conductive heat flow paths as directly as possible to the ultimate sink, by reducing radiative coupling between high and low temperature parts, and by designing for a low temperature gradient in the air or other coolant. High temperature parts such as electron tubes and resistors may be enclosed in surfaces with high absorbtivity on the inside and low emissivity on the outside. These envelopes should be conductively connected to the sink. Electrical leads to parts are significant conducting paths, and the wiring complex is a useful intermediate sink. Leads should be as large and heavy as feasible.

In close packed equipment, radiation shields are often difficult to fit due to space limitations. Parts placement should be planned to avoid direct radiation paths from high temperature sources, as discussed in paragraph 8.3.

8.3 Thermal design of equipment for natural cooling. The method of heat removal from within electronic equipment must be such as to provide a low temperature gradient between the heat producing parts and the cooled surface or the local connection to the ultimate sink. The cooling method must be simple, light weight, reliable, easily maintained, and economical. Further, it should occupy a minimum of volume, preferably utilizing the voids between densely packaged parts.

Natural cooling is recommended for use inside small, close packed electronic equipment packages. When properly designed, with maximum use of thermal conduction, it is usually adequate, even when the heat concentration is so high as to require artificial cooling methods external to the package. Among the advantages of natural cooling are simplicity, compactness, and freedom from a requirement of power for operation. Artificial cooling requires power to drive it, and this power adds to the heat load. The efficiency of a heat pumping system generally increases with capacity. Thus, the additional heat load is reduced if natural cooling is used extensively in small components of an electronic system.

Metallic conduction should be generally used as the primary mode of natural heat transfer within equipment. Cooling by radiation is not recommended as a primary means because large temperature differences are required for appreciable heat transfer. Also, control of the heat flow paths is difficult due to scattering of the radiant energy. Convection cooling is not recommended as a primary mode of heat transfer, since large areas, which are seldom available in military equipment, are necessary to maintain a reasonable temperature rise. Convection currents frequently transfer the heat into other areas which in turn will require increased cooling. Modern equipment is usually so densely packaged that heat is also transferred between parts by gaseous conduction. Control of the heat interchange through these diffuse heat flow paths may be difficult, and extensive use of metallic conduction can minimize their effects.

The cooling problem is less severe as the electrical efficiency of the circuitry is increased. Further, the size, weight, and cost of the equipment will be decreased, and a probable increase in reliability achieved. It is therefore, very important that the equipment be designed for minimum heat dissipation.

8.3.1 Packaging considerations - Parts placement and mounting. When natural cooling means are used, the placement of electronic parts within assemblies is particularly important. Since there is a vertical temperature gradient in the package, a few general rules should be observed.

Temperature sensitive parts should be located at the bottom if possible, and never directly above heat producing parts.

Placement should be graded by allowable parts temperatures, those which can run hotter above those which must be cooler.

Parts should be so oriented as to have the best free convection, as determined by calculation of the heat transfer coefficient. In general it is found best to orient parts with the longest dimension vertical.

Parts mounted in a vertical array should be staggered horizontally.

Mounting boards and subchassis should generally be placed vertical.

Large heat sources should not be in vertical line with other parts, and should be as remote as possible horizontally.

Large heat sources should in general be placed as near as possible to and in conductive contact with the coolest surfaces, which are usually the inside of a metal enclosure, metal chassis, and metal supporting structure.

Electrical leads and wiring should be as large in crossection and short as feasible.

The mounting interfaces of heat producing parts should be made with as low a thermal resistance as possible. Careful workmanship and inspection during assembly here is even more important than design.

The thermal design must be such that conductors having pre-determined crossectional areas and short heat paths are provided at specified locations in order to prevent the flow of heat into unwanted paths.

8.3.1.1 Resistors. Resistors must be derated when mounted in groups. This is especially applicable to resistors dissipating appreciable power. Figure 18 illustrates the percent of single unit rating vs. the number of resistors in the group. Separate curves are given for five different spacings and, in any group of three or more, the spacing between resistors is identical. Two percentage scales are shown, one for free air and the other for a mesh enclosure. Note that these data are for vertically mounted convection and radiation cooled power resistors only. Power resistors with suitable clamps and mounting lugs for cooling by conduction into their supporting structure are generally recommended.

Large wire-wound resistors dissipate considerable heat and must be mounted not only for adequate cooling, but also as to minimize heating of adjacent parts by radiation. Power resistors usually operate at high temperatures and much of their heat will be rejected by radiation if conductive paths are not provided. Vertical mounting is recommended when more than a single resistor is involved. Individual power resistors over 4 inches long should be mounted horizontally, if possible. The average temperature of a long, horizontal resistor is only slightly higher than that

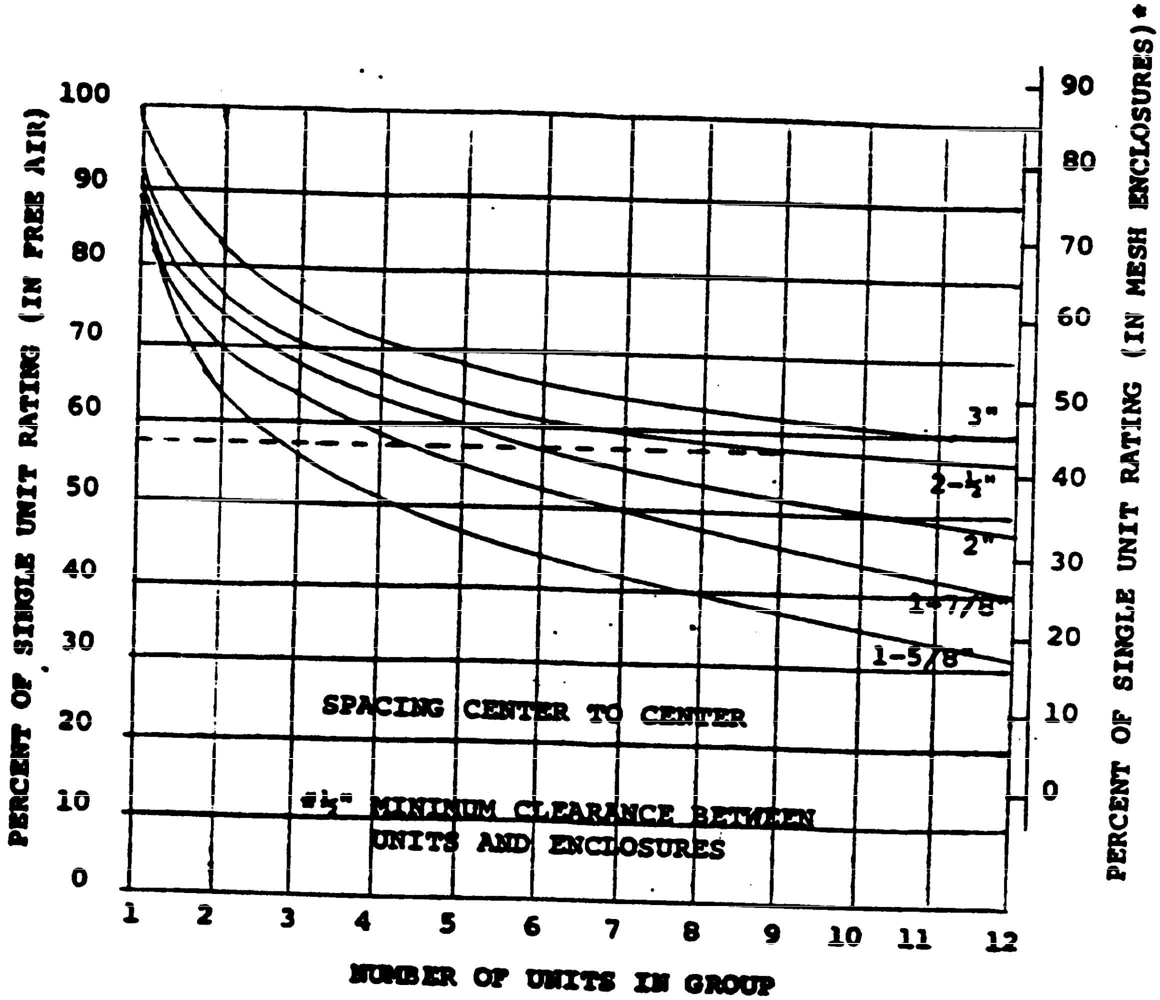


FIGURE 18. Group Mounting of Power Resistors

obtained when vertically mounted. However, the hot-spot temperature will be much lower and the temperature distribution will be more uniform with horizontal mounting. Resistors of significant length will have a temperature "build up" due to convection, if vertically mounted. It is necessary that radiation shields, preferably of polished metal, be placed between power resistors and temperature-sensitive parts, if the parts are less than two inches away from the resistors.

When composition carbon and other similarly shaped resistors are mounted within 1/8 inch of a cool metal surface, gaseous conduction occurs and their surface temperature rise will be less than in free air under equivalent conditions. Conversely, when such resistors are mounted from 1/8 to 1/4 inch of a cool metal surface, their temperatures will be increased over the equivalent free air values because of the impairment of convective air flow. If such resistors are mounted near each other, mutual heating will be initiated when the spacing is 1/4 inch or less. The mounting position, either horizontal or vertical, has little effect. Should the separation between resistors dissipating their rated power be reduced to 1/8 inch, each resistor will increase from 10 to 15° in surface temperature. Fuse clip-type recommended for high-temperature operation or when the power dissipation is high. This conduction-cooling technique will produce the lowest thermal resistance obtainable by natural means.

8.3.1.2 Electron tubes. Mutual heat transfer between electron tubes can easily lead to excessive temperatures. Consequently, it is recommended that closely grouped tubes be provided with shields capable of conducting the heat into the mounting structures. If shields cannot be utilized, and if tubes must be mounted in groups cooled only by natural methods, then the bulb-to-bulb separation must be at least 5/8 inch and preferably 1 1/2 bulb diameters, when the bulb diameter exceeds 1/2 inch. The greater the separation the better, especially if the tubes are dissipating appreciable power.

8.3.1.3 Solid state devices. Low power (small signal) transistors, Zener diodes, IC's, etc., should be so located as to minimize the absorption of heat from large heat sources and hot parts of metallic conduction paths. Radiation and heat insulation shields may be required. Solid state devices dissipating one watt or more, having an attached heat sink as an extended convective surface, should be placed and oriented for best free convection cooling. For dissipation above 5 watts, a conductive path to a cool metal surface should be provided. Circuitry is generally designed for thermal stabilization of transistors. Zener diodes are quite temperature-sensitive and should be located with special care to avoid thermal interaction.

Composite card circuits are constructed by mounting discrete parts such as transistors, diodes, resistors, capacitors, etc., on boards which support or contain the interconnecting wiring. Mounting boards generally have on one edge a terminal strip which plugs into a multiterminal receptacle.

The equivalent thermal circuit of a composite card or a hybrid consists of a more or less complicated series-parallel thermal resistor network connected to several power sources. For non-steady state operation several thermal capacitors must also be included.

Some, indeed most, of the thermal resistances are actually distributed more or less continuously over a region. However, it is customary to treat them as discrete resistors connected between temperature modes. One example is a transistor heat sink cooled by convection and radiation. Since the sink material does have thermal resistivity and the heat dissipation is distributed over the surface, integration is required to accurately determine the lumped heat transfer effect. Heat sink manufacturers publish data on the thermal body and interface resistances and information on methods of installation. The equipment designer need only select units which will achieve the resistance values required to obtain the desired temperatures.

Heat sink ratings by manufacturers should be in accordance with the EIA test procedures outlined in EIA bulletins 5 and 5A. Some heat sinks are rated in terms of the thermal resistance from the transistor mounting surface of the sink to the air. Other ratings are from the transistor case to the air and include the interface mounting thermal resistance as recommended by EIA.

Printed circuit boards, mulitlayer boards, and interconnecting wiring generally, vary so much in design that standard values of thermal resistance are not available. Since they furnish a very important spreading thermal resistance between heat sources and sink it is usually necessary to estimate their values in the thermal circuit, either by measurement or by approximate calculation.

As explained in preceding paragraphs, manufacturers of power transistors, diodes, etc., publish values of thermal resistances from junction to case, to mounting stud, and to heat sink. Characteristic values of junction to case resistance for low power solid state devices are not as readily available. Manufacturers do generally publish factors for derating the power dissipation in terms of environmental or ultimate sink temperature, and these factors may be used to estimate the thermal resistances from the junctions (the heat sources) to the sink interfaces.

In typical designs, composite card circuits are built on "cards" or "pages" of the order of 4 x 6 inches, which are plugged into terminal strips. Thay are held in parallel position by guide strips with air spaces between them. Thus, both sides of the card are cooled by free convection and the electrical connections furnish parallel heat conducting paths to the ultimate sink. PCB's have a substantial fraction, 60 or 70%, of the back area covered with copper, which affords a reasonably good surface for free convection cooling. The discrete parts on the other side of the card each present their own surfaces to convection. The thermal circuit is typically as shown in Figure 19. As noted previously, the thermal resistance R_b is the most difficult to evaluate. For boards made of insulating materials, R_b depends chiefly on the volume of metal present in leads, plated through holes, and the plated conducting surface. For circuits mounted on substrates of aluminum or other thermally conducting material, the electrical insulation is obtained by very thin layers of oxide. Heat flows uniformly through these substrates, and R_b is generally lower than in non-metallic PCB's. Multilayer boards contain relatively more copper than single layer boards, which somewhat reduces the thermal resistance. Examples of estimating representative thermal resistances for composite circuits cards are given in paragraph 8.3. The SHP chapter of this handbook describes the thermal details of SHP Modules.

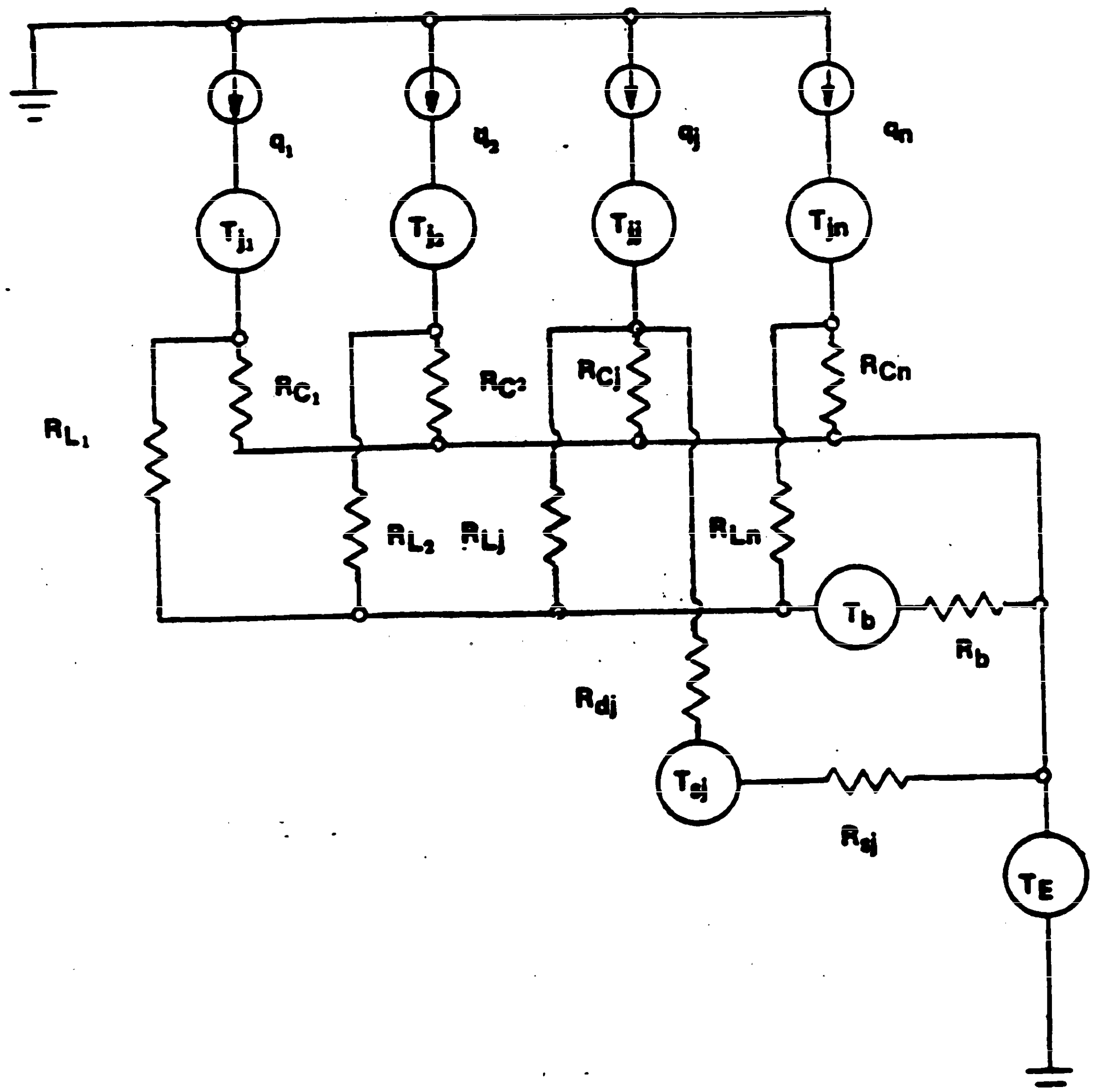


FIGURE 19. Typical Thermal Circuit of a Hybrid

Metal core circuit boards (MCCB's) are available, from at least one manufacturer. Aluminum boards are coated with a 0.004 inch thick dielectric coating which is claimed to be excellent mechanically and electrically. Typical conductor capacitance to the grounded aluminum card is 250 ppfd/in^2. Satisfactory operation of solid state circuits to 150 MHZ has been obtained. Thermal resistances from the metal card to the attached parts are very low. This concept offers an excellent approach to significant reductions in the thermal resistances of card mounted type hardware. (Reference 13)

8.3.1.4 Transformers and reactors (inductors). Iron core reactors dissipate heat roughly in proportion to the square of the current and frequently (see chapter 17). The heat rate is generally low, but may be significant for filter chokes in power supplies, for example. Power transformers are important heat sources. When the temperatures of iron core devices are high, they should be carefully heat sinked and located so as to minimize thermal interaction with other parts, preferably in a separate section of the package, or in a separate package.

8.3.1.5 Heat conducting parts. It should be recognized that, when radiant and convective heat transfer between heat producing parts is minimized by conduction through metals, considerable thermal interaction can occur if the parts are transferring their heat to a common metallic conductor. However, if the common mounting structure or conductor has a low thermal resistance to a sink, then it is relatively cool and little mutual heating can occur. Alternatively, the parts can be mounted on separate heat conductors.

The main point is that is a common metallic conductor is used the thermal resistance between parts will be lowered and the mutual heating will be increased, unless a good heat path to a sink is included.

8.3.1.6 Passive parts. Non-heat producing parts may be temperature sensitive and must be located so that the transfer of heat from heat sources is minimized. If heat is transferred to non-heat-producing parts, it may be desirable to provide these parts with a separate low-resistance thermal path to a sink. It is futile to insulate parts thermally when they are located in or near a high-temperature region. Insulation will only lengthen the time to reach thermal equilibrium, but the parts still will be heated. It is advisable to place the non-heat-producing parts in the region of lowest temperature which will usually be near that location having the lowest thermal resistance to the sink (see Figure 20). This arrangement is recommended. Figure 21 presents an alternate configuration which, under equivalent conditions, will result in somewhat higher temperatures than obtained with the arrangement of Figure 22.

8.3.2 Environmental considerations. Heat transfer in electronic equipment is significantly affected by environmental conditions. The ultimate sink is an environmental factor. The thermal properties of materials depend on environmental temperature. With equipment exposed to the weather, radiation, chiefly from the sun, is a heat input which increases the cooling system load. Pressure and centripetal forces affect free convection. The structural requirements for withstanding stresses of vibration, acceleration, pressure, and potentailly hazardous environmental factors such as corrosion, particle impingement, and weather, all have an influence on the cooling system.

The thermal conductivity of all materials changes with temperature. For metals it decreases moderately with temperature and may be considered constant over the range of electronic equipment temperatures. For liquids and gases the effect is larger. For example, between 0 and 200°C, k increases about 80% for air, tenfold for hydrogen, 20% for typical liquid coolants used in electronics. The local temperature of the coolant should be estimated when computing heat transfer from an individual part. It is evident that conduction cooling is the most desirable mode of heat transfer when natural means only are used, since it tends to maintain lower temperature gradients in the equipment.

The requirements of vibration and shock resistance generally require secure mounting and substantial support structure. The supports and structure offer convenient paths for metallic heat conduction, and should be so employed as far as possible.

Free convection heat transfer decreases with decreasing density and increases with increasing temperature difference, as discussed in paragraph 8.1.2. It also decreases with decreasing gravitational acceleration and ceases to act in a zero gravity environment.

Moisture and humidity have minor effects on natural cooling designs. Free convection is slightly enhanced by high humidity due to the high specific heat of the water vapor. Moisture proofing treatment of equipment may somewhat impair conduction cooling if the varnish seeps into contact joints. All thermal contact bonds should be secured before the application of moisture and fungus treatment.

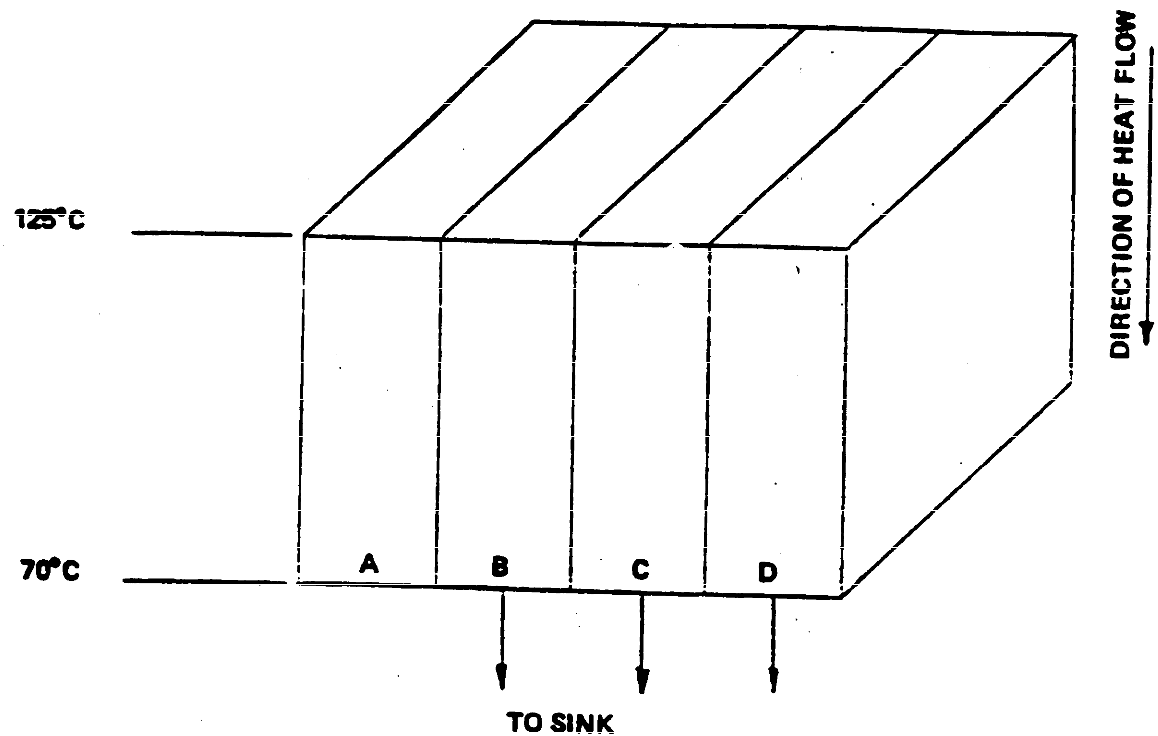
High temperature bodies adjacent to electronic equipment make it necessary to incorporate thermal isolation in the design. In case of direct contact, thermal insulating material is required. The more usual situation is radiant heat coupling.

Thermal radiation from the sun or from adjacent high temperature equipment such as nuclear reactors, steam boilers, furnaces, etc., is an environmental factor of great importance. Electronic equipment operating in such an environment must incorporate radiation shielding, which is more effective than thermal insulation. The shield should be thermally connected to ultimate sink so that the radiant heat bypasses the equipment. This technique becomes difficult in space applications where the only sink available is the "darkside." Heat conducting path design then becomes of first importance.

8.3.2.1 Ground based equipment. Ground based electronic equipments operate in a wide variety of environments. They may be exposed to extremes of weather conditions, arctic, arid, or tropical, or may be housed in a wide variety of structures.

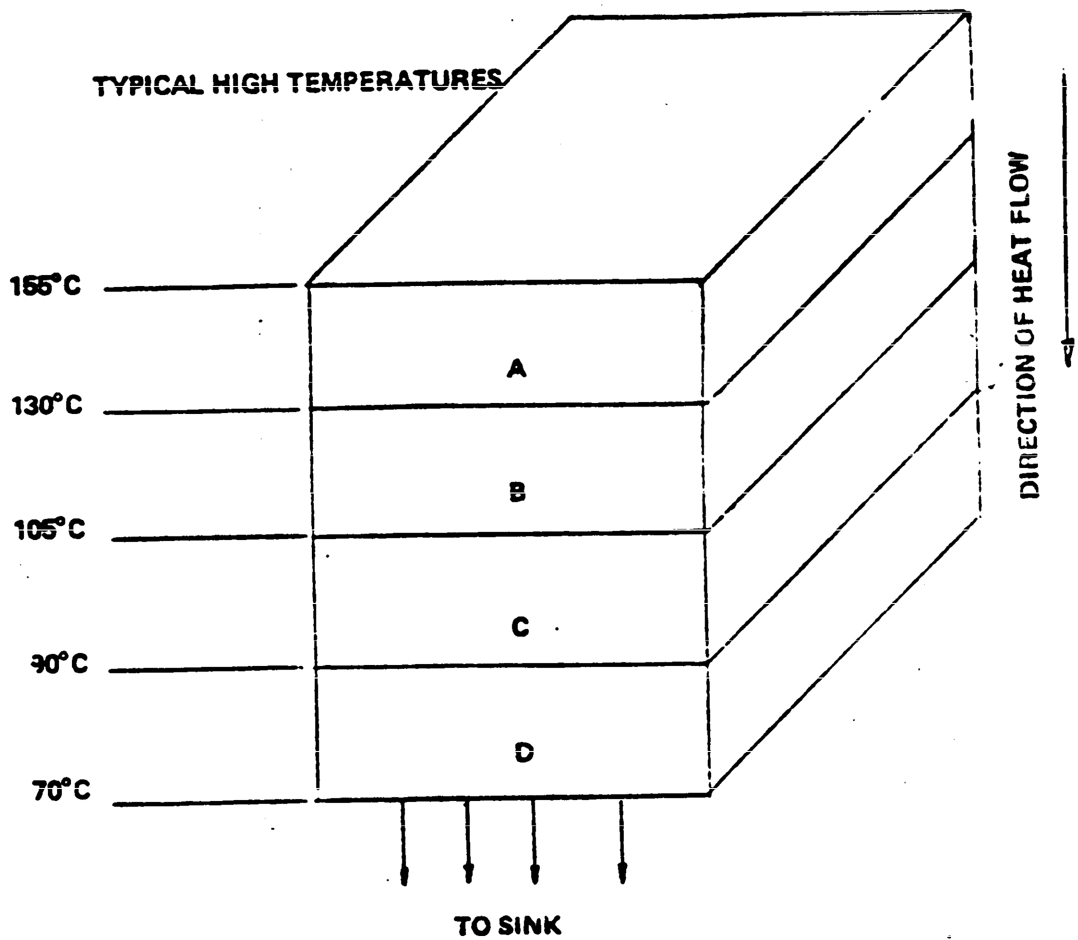
When natural cooling is the only means used, the following conditions apply:

Equipments for exposed outdoor installation must be in weatherproof cabinets. Internal conduction cooling to a convective surface, louvers to confine and direct the upward flow of air, and to exclude rain, snow, sand, etc., extended surfaces in the form of vertical fins, external heat reflecting surface finish, are useful design features. Portable equipments are frequently covered with fabric cases, which have high absorptivity and impede air circulation. The cases should be designed so that they can be opened or removed while the equipment is operating.



Region A - High temperature region with parts having large heat concentration
Region B - Medium temperature region with heat producing parts
Region C - Cool region with non heat producing parts
Region D - Coolest region with temperature sensitive parts

FIGURE 20. Recommended Method of Locating Parts



Region A - High temperature region with parts having large heat concentration
Region B - Medium temperature region with heat producing parts
Region C - Cool region with non heat producing parts
Region D - Coolest region with temperature sensitive parts

FIGURE 21. Alternate Method of Locating Parts

Many ground based equipments are installed in buildings, portable shelters or mobile vans, sometimes with air conditioning facilities. If the equipment heat load is dumped into the compartment air, it is added to any other heat sources present and may require additional compartment cooling. One person dissipates a minimum of 100 watts even when inactive. Solar radiation may be as high as one watt/sq. in. Electronic equipment heat dissipation may increase by a factor of 2 or more the heat produced by personnel and absorbed through the roof and sides of a shelter.

Free convection is always operative on the outside surfaces of equipment, and whatever cooling system is used, the internal heat flow must be so designed that the temperatures of these surfaces do not become intolerably high for personnel.

Electronic equipment installed in drawers which must be pulled out for adjustment and service frequently present a serious cooling problem. An open drawer generally relies on natural cooling even if forced air cooling is used inside the cabinet. The thermal design must be such that electronic parts do not become overheated when a drawer is opened during operation or maintenance.

8.3.2.2 Shipboard Equipment. Two general methods of heat sinking on shipboard are: compartment air ventilation or conditioning units, and cooling water supplied from a coolant to sea water heat exchanger. Similar conditions apply to shipboard and ground based equipment, but the contaminating and corrosive effects of sea environment must also be considered. Natural cooling methods may be used partially or entirely inside the enclosures. Except for the customary weight and ruggedness of shipboard equipment, the cooling problems are much as for ground based equipment. The comments regarding enclosure surface temperatures and equipment installed in drawers applies.

Equipment inside waterproof enclosures in exposed locations on deck, mast head, etc., will in general be cooled by free convection from the outer surface. In this case, heat conducting paths to the case are required. The possible extremes of external air temperature and the heat input due to solar radiation are important considerations. Waterproof enclosures do not permit direct contact of external air with electronic parts.

Equipment such as sonar gear, installed outside the ship's hull below the waterline may be cooled by free convection to the water. Sea water temperature varies from -2 to 35°C at usual depths of the installation.

8.3.2.3 Airborne Equipment. Electronic equipment in aircraft is exposed to widely varying environmental conditions, ranging from high temperature tropical or desert air on ground or deck, to the cold, low pressure air at 40,000 feet altitude or higher. Also, the skin of high performance aircraft is exposed to ram air temperatures of several hundred °C.

Various forms of ultimate sink are provided. Compressor bleed air, ram air, fuel, expendable evaporative coolants, and refrigerating units are all in use. Natural cooling means are generally applicable only inside of equipment enclosures, and their effectiveness is limited to situations of relatively low heat concentration.

Ground cooling of airborne electronic equipment should receive careful attention to assure that under worst case conditions, it is at least equivalent to the airborne environment. Further, the ground cooling system should be designed so that the avionics can not be operated without ground cooling present.

At altitude, free convection is only slightly effective in non-pressurized compartments, as explained in paragraph 8.1.2.

Aircraft electronic equipment is almost always densely packed, with a minimum of structural support, and thin sections of metal. Such light weight features restrict the usefullness of structural parts as thermal conductors. Also, due to the weight penalty, it is not usually permissible to add metallic parts merely to provide conductive paths.

Heat pipes are light weight thermal conductors of very high conductivity. They are discussed in detail in paragraph 12.2. Application of heat pipes in aircraft is advantageous. However, their peculiar performance during changes in inclination and gravitation must be carefully considered.

8.3.2.4 Space and Missile Applications. The thermal environment of space vehicles and long range missiles include intense solar radiation, high acceleration, zero gravity, zero air pressure, intense cold on the "dark side," short period high vehicle skin temperature (400°C for two minutes, Reference 15).

The equipment is built with as low weight as possible, but structural elements must be heavy enough to withstand the mechanical stresses and are therefore useful as thermal conductors.

Free convection is generally inoperative except during pre-launch checkout. Since this is a lengthy process, during which the vehicle heat control system may be quite different than in normal operation (Radiation cooling to the "dark side" not functional), the thermal design of electronic equipment must accommodate two quite different environments. Free convection may be advantageously used during pre-launch.

Conduction is the most useful natural heat transfer method. Heat pipes are attractive possibilities as passive, evaporative thermal conductors. However, as in aircraft, their performance characteristics under changing centripetal force must be considered.

8.3.3 Conduction cooling. The thermal resistances between the heat sources and the sink connectors or heat transfer surfaces of a typical conduction-cooled assembly are outlined in Figure 22. If each major heat source is considered and the minor heat sources are lumped, a schematic of the thermal circuit can be drawn for analysis. The power dissipated by each heat source and the desired operating temperature should be known, together with the temperature of the sink connector. The thermal resistances representing electron tubes, solid state devices, tube shields, mounting hardware, resistors, and transformers, are determined as described in paragraph 8.1.1. Thermal resistances through metal parts such as chassis can be determined from electrical resistance measurements if necessary, or by calculation as outlined in paragraph 8.1.1.

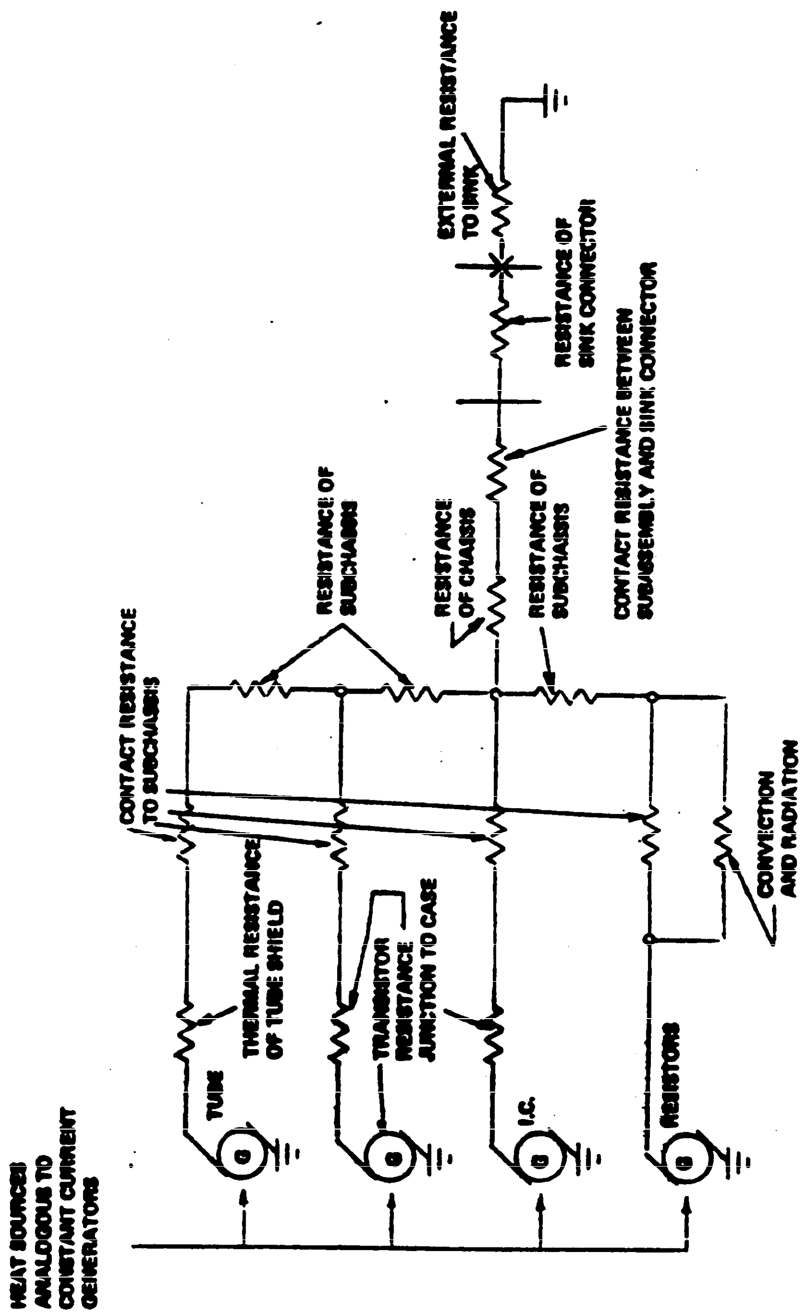


FIGURE 22. Major Thermal Resistances in a Conduction Assembly

Mounting devices made of insulating material such as printed circuit boards and multilayer boards are important heat conductors due to their copper content. Such boards are complicated composite conductors consisting of the embedding material in parallel with the copper. An estimate of thermal resistance can be made by measuring electrical resistance as described in paragraph 8.1.1. However, since insulating material does not obey the Wiedemann-Franz Law, this method may be erroneous and should be checked by studying the structure of the board. The method of estimating is illustrated by two examples.

Example 8-7: A PCB of fiberglass 0.05 in thick carries copper conductors 0.002 inches thick, covering an estimated 40% of the surface. The board is 4 x 6 inches with connector pins on one of the short edges. Estimate the thermal resistance from the pins to a lead of 0.02 inches diameter at point P. (See Figure 23)

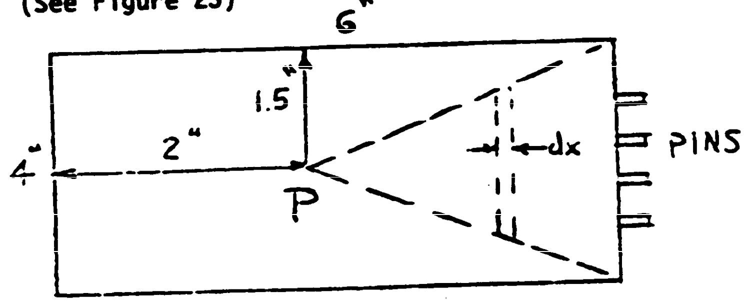


FIGURE 23. Estimate of PCB Thermal Resistance

It is assumed that the heat flow pattern comprises straight radial lines from P to the edge.

For the fiberglass,

$$R = \int_{0.01}^{4} \frac{dx}{k\,ty}$$

Since $y = 0$ when $x = 0$

and $y = 4$ when $x = 4$

$y = x$

$k = 0.004$ w/in-°C

$t = 0.05$ in.

$$R = \int_{0.01}^{4} 5000 \frac{dx}{x} = 5000 \ln 400$$

$$= (5000)(5.99) = 3x\,10^{4}\ ^{\circ}C/w$$

For the copper $k = 9.7$ w/in,-°C

$$R = \int_{0.01}^{4} \frac{dx}{(9.7)(.002)\frac{(40\%)}{(100)}\,x} = (129)(5.99)$$

$$= 772^{o}C/w$$

The parallel combination is 750°C/w.

Example 8-8: Multilayer Board (MLB). An MLB has three layers of circuitry separated by 0.015 inches of fiberglass. Dimensions are the same as in the previous example.

One layer of plating developes the same resistance as previously calculated, 772°C/w. The other two layers form parallel heat flow paths. The resistance diagram can be assumed as in Figure 24.

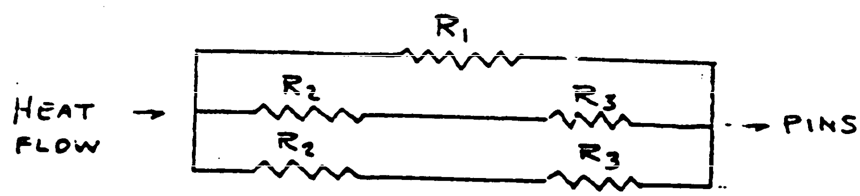


FIGURE 24. Thermal Resistance Diagram of an MLB

R_1 = resistance of connected layer
R_2 = resistance of fiberglass layer
R_3 = resistance of copper layer
R_2 is the resistance of a triangular sheet of fiberglass.

$$R_2 = \frac{t}{KA} = \frac{0.015}{(0.004)(4)(4)(0.5)}$$

$$= 0.47°C/w$$

R_2 is thus, negligible.

R_3 is estimated as one half of the value for the electrically connected layer, due to the spreading effect.

$$R_2 + R_3 = \frac{772}{2} = 386°C/w$$

The total resistance is:

$$R = \frac{1}{\frac{1}{772} + \frac{1}{386}\quad\frac{1}{386}}$$

$$= 155°C/w$$

Another method of estimating is to consider the MLB as a homogeneous conducting sheet. On this basis R is composed of six paths in parallel, three of copper and three of fiberglass. Then R = 750/3=250°C/w.

The internal and/or surface temperatures of the parts are stipulated by the reliability design. Thus, when the dissipated power of each heat source, the required operating temperatures of the parts, and the sink temperature are known, the various thermal resistances in the network can be calculated from the circuit diagram.

Some of these resistances are fixed by parts selection and any changes might be expensive. Other resistances, such as intermediate heat sinks, mounting board structure, wiring, and structural elements can be changed at the discretion of the designer to meet the requirements of the thermal design.

When heat flows through interfaces between parts the contact resistance must be calculated, since it is often significantly large. Thermal contact resistance is discussed in paragraph 8.3.3.1. Contact resistance occurs as a series element in most thermal equivalent circuits, and it should be minimized by using conductive sandwich material, by maintaining flat surfaces, and by high contact pressure.

If the surfaces of two dry metal blocks are placed in contact there remains a considerable resistance to heat flow from one block to the other, unless the surfaces are bonded together by a solid metal bond as in welding, brazing, or soldering. This thermal resistance, known as "contact resistance," is a function of the actual contact area, the presence of a solid, fluid, or vacuum in the gap between the surfaces, and the presence of oxide layers on the contact surfaces. Significant changes in the mean interface temperature may also produce changes in any or all of the above conditions. The actual contact area is a function of the physical properties of the contact material, and the contact pressure.

Contact resistance is present in addition to the usual thermal resistances of the materials themselves. The resistance of the contact may be large compared with the other resistances and should rarely be neglected for metals. However, the contact resistance may be neglected for thick blocks of nonmetals with low conductivity.

The contact resistance in a vacuum or at very low interstitial gas pressure is considerably greater than in the presence of air or other fluids.

If the surfaces considered are warped, then this non-flatness may have more effect on contact resistance than the roughness of the surface. The degree of surface roughness considered herein is for flat samples.

With lessening of contact pressure, at the temperatures and pressures considered herein, the results are repeatable provided the elastic limit has not been exceeded on either of the mating surfaces. When plastic deformation occurs the contact resistance will decrease due to the increase in actual contact area caused by this deformation. If the contact pressure is increased enough to produce plastic deformation, then a subsequent reduction of pressure will increase the contact resistance, but the resistance will not become as high as it formerly was at the same reduced pressure.

The effect of addition of thin metal shims on the overall contact resistance of a joint depends on the relative hardness of the blocks and the shim material. For example, blocks of material which are hard relative to the shim material (as for example, steel with aluminum or brass shims) will show a decrease in the overall contact resistance as compared to the same joints without shims, especially at high pressures. Conversely, blocks which are soft relative to the shim material exhibit contact resistances which are higher than for the same surfaces without shims (e.g., blocks of aluminum with and without brass shims). Aluminum blocks with aluminum shims seem to show no difference from the same aluminum joints without shims.

Non-metallic sandwich material such as asbestos or a conducting cement, may increase the contact resistance by as much as 10 to 1, if applied so thick (as was the case with the data listed herein) that it actually moves the blocks apart rather than merely replacing the air in the interstices.

The statements made above pertain only to truly flat or planar surfaces. The contact resistance of warped or wavy surfaces may be improved (i.e., decreased) by the use of sandwich materials. The reason for this is that with warped surfaces most of the thermal resistance is introduced by a relatively thick layer of gas (or liquid) between the blocks and replacement of a poor conducting medium such as air by a better conductor will effect a decrease in contact resistance. Such a joint, however, will give no better contact resistance than that listed in this section. It merely provides better contact where good mating of the block surfaces is not possible.

The contact resistance investigation results of Reference 21 are reproduced in Figure 25 in which contact resistance is correlated as a function of contact pressure using various steel and aluminum surfaces. The roughness is indicated by the RMS index value which is the root-mean-square value of the heights and depths of the minute hills and valleys which form a machined surface. Thus, a lapped surface with an RMS of 4 μ inches (4 millionths of an inch) would constitute an extremely smooth surface.

Appendix D lists contact thermal resistances of a wide variety of dry and sandwich type joints, the effects of air pressure and of liquid filler materials. These data are reproduced here by courtesy of the General Electric Company.

8.3.3.1 Plastic embedment. Embedment of electronic parts in plastic compounds is sometimes used for mechanical strength and electrical insulation. The thermal conductivity of embedding and encapulating compounds is low, ranging from 0.0035 to 0.01 watt/in.-°C. This compares unfavorably with ceramics such as aluminum oxide (0.4 watt/in.-°C) and beryllium oxide (2.8 watt/in.-°C), and with all metallic conductors.

Heat transfer in embedded assemblies is primarily by conduction through the plastic in conjunction with some metallic conduction in the wiring. Sometimes such assemblies incorporate built-in metal heat conductors and are actually cooled by metallic conduction cooling. The thermal design of this type of assembly should be based on metallic conduction alone, with the plastic serving only as a structural medium. This construction method should be used above 0.25 watts/cu. in. The following discussion is limited to those types which rely on heat conduction through the plastic as the primary cooling mode.

In general, due to the relatively high thermal resistance of plastics, parts dissipating appreciable power should not be potted. Non-heat-producing

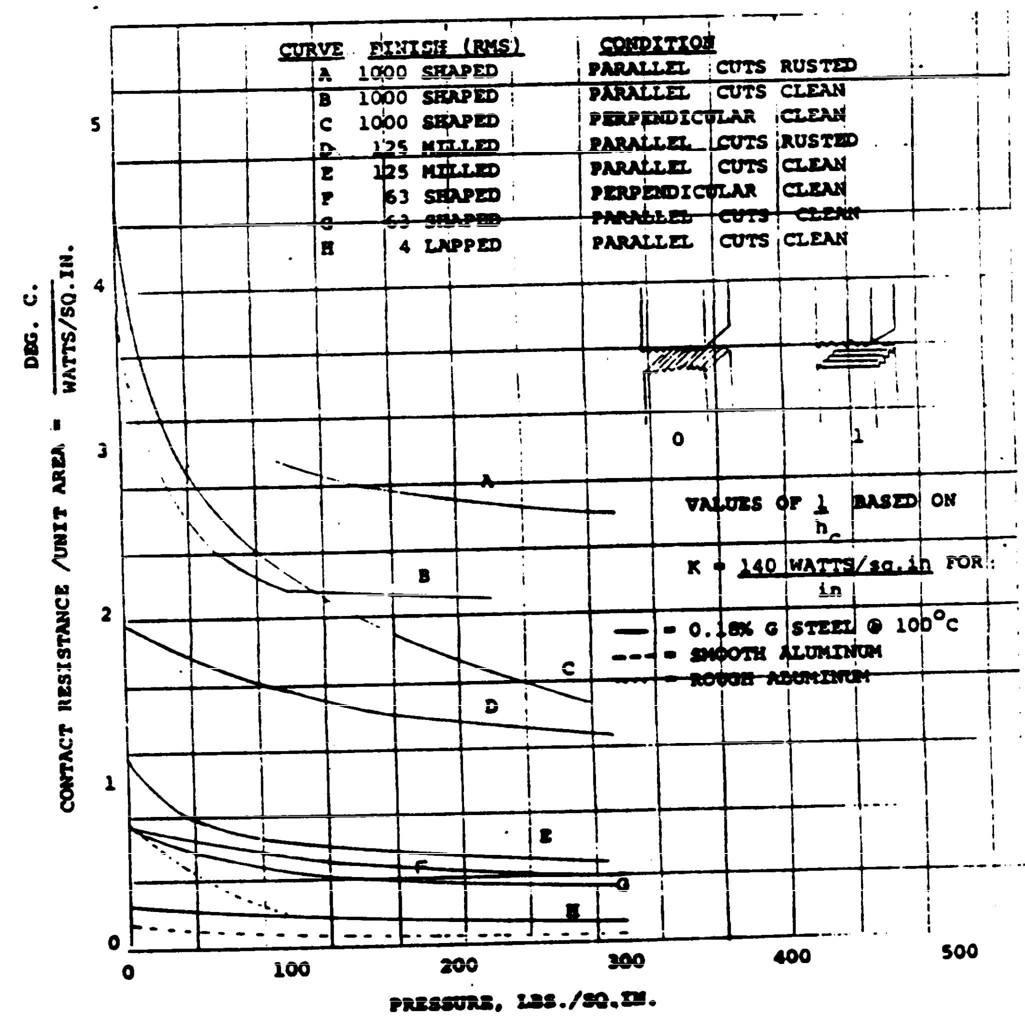


FIGURE 25. Contact Resistance As A Function of Contact Pressure

parts and certain heat-producing parts having small heat concentration can be successfully encapsulated. This is exemplified by Figures 26 and 27. Reference 37 contains similar information. Note that the thermal gain, even at the lower power levels, is relatively small. Although the unit heat dissipation and heat concentration have been significantly reduced by providing the resistors with much larger cooling surfaces, the additional resistance to heat transfer within the plastic minimizes these gains. Further, note on Figure 27 that at the higher powers, the free air surface temperature is lower than that obtained in the plastic. This reversal is believed to be due to the absence of heat transfer from the encapsulated resistor by radiation. Not only is plastic generally a good absorber of long wavelength radiation, but the highly reflective interface between the plastic and the resistor also tends to minimize heat transfer by radiation.

Extreme care must be utilized in designing a potted assembly. Embedment materials which can withstand peak temperatures in excess of 185°C are not currently available. With heat concentration of the order of 0.5 watts/cu. in., excessive temperature gradients can easily occur, leading to mechanical fractures in the plastic and failure of electronic parts. For assemblies having heat concentrations less than 0.25 watts/cu. in., plastic embedment will provide adequate heat removal, if the environmental temperature is reasonably low.

Certain electronic devices which must withstand very severe shock and vibration stresses are potted in synthetic resins containing particles of aluminum in order to achieve satisfactory thermal performance. The addition of fine metal particles will increase the thermal conductivity of embedment materials, depending upon the metal, the quantity involved, and the particle size. Such metal filled compounds are not standardized commercial items and must be made to order when their use becomes necessary.

Embedment of electron tubes is not recommended because of the danger of cracking the envelopes.

8.3.4 Extended surfaces for convection and radiation cooling. When free convection is used as a major means of heat removal it is frequently necessary to apply extended surfaces to the heat sources. These are commonly called fins and will be so called in this discussion.

Standard designs produced by several suppliers are frequently used particularly for transistors as described in paragraph 8.2.1. However, it is often desirable, or necessary, to develop special designs, due to cost or unavailability of standard parts.

Figure 28 shows the pertinent thermal considerations involved in the design. The first consideration is the area of the convecting surface required. This is fixed principally by the environmental air temperature and the heat to be dissipated.

The heat transfer coefficient is a function of the geometry and orientation of the surface and the temperature difference. Since R3 is the largest of the three thermal resistances, it is estimated initially by choosing a tentative value of surface temperature.

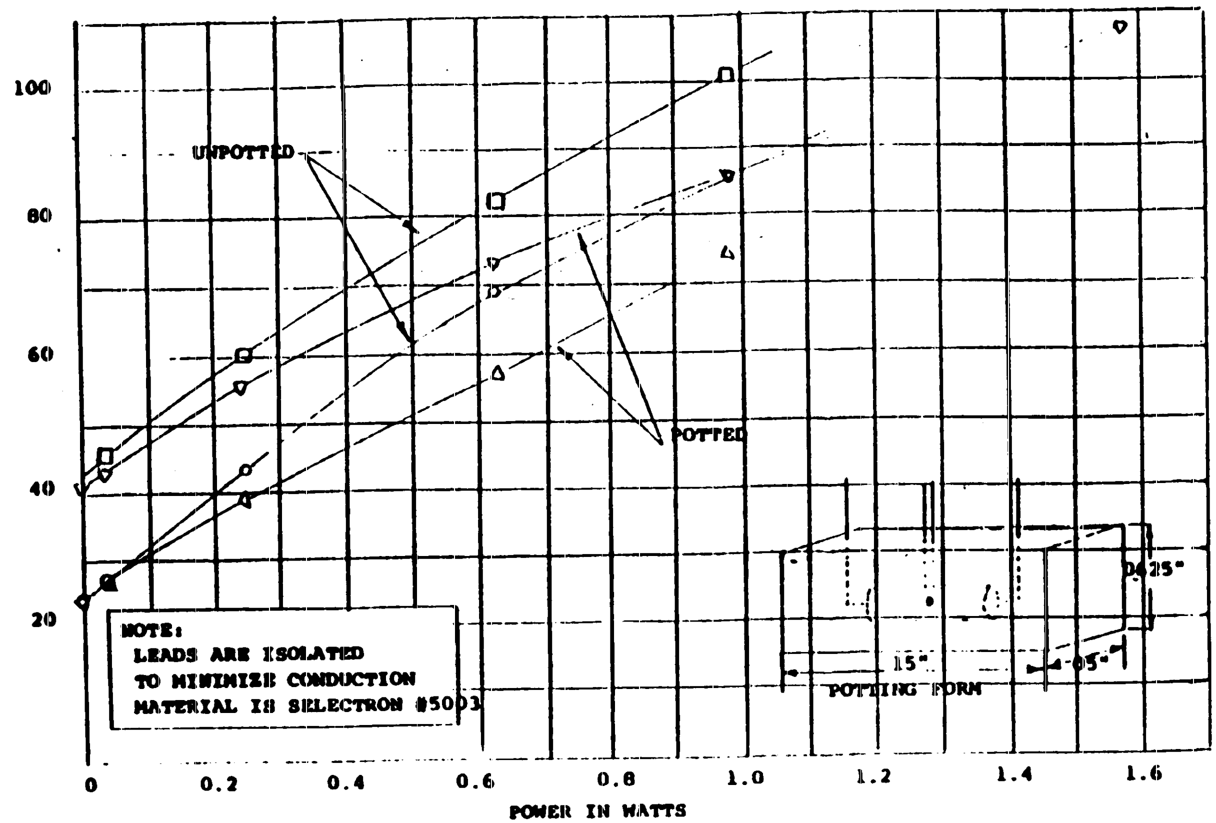


FIGURE 26. Body Surface Temperature of 5600-Ohm, 1 Watt Composition Resistor As A Function of The Power Dissipated

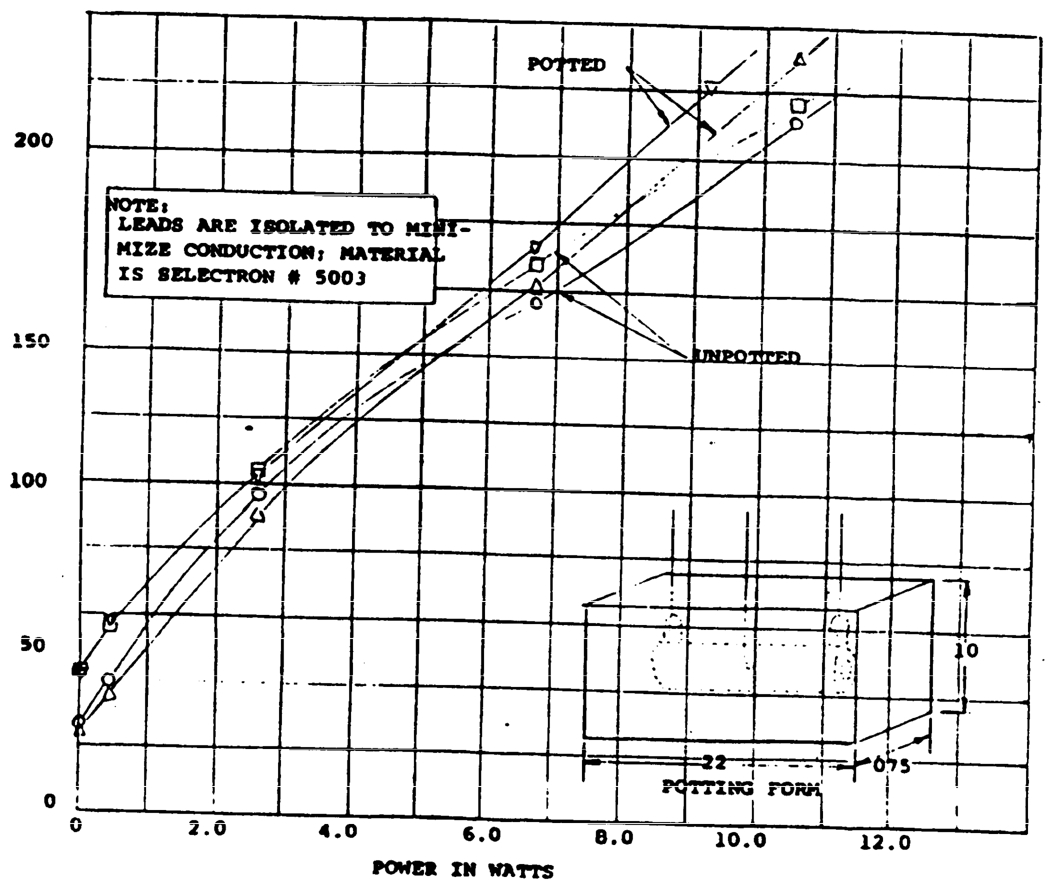


FIGURE 27. Body Surface Temperature of 5000-Ohm, 10 Watt, Wirewound Resistor As A Function of The Power Dissipated

The temperature of the source T_s is fixed by the reliability design. The sum of the three resistances is:

$$R1 + R2 + R3 = \frac{T_s \; T_{-a}}{q} \qquad (8\text{-}43)$$

The contact resistance R1 is fixed by the method of attaching the fin. Intimate contact, with close fit and as high a clamping pressure as the part will sustain are necessary. R1 is estimated from the data in paragraph 8.3.3.1.

A tentative value of R2 can then be selected. This is a spreading resistance through the metal of the fin. It depends on the shape, thickness, and kind of metal used. Fins are generally made of sheet metal, cut or stamped to shape and frequently bent for compactness.

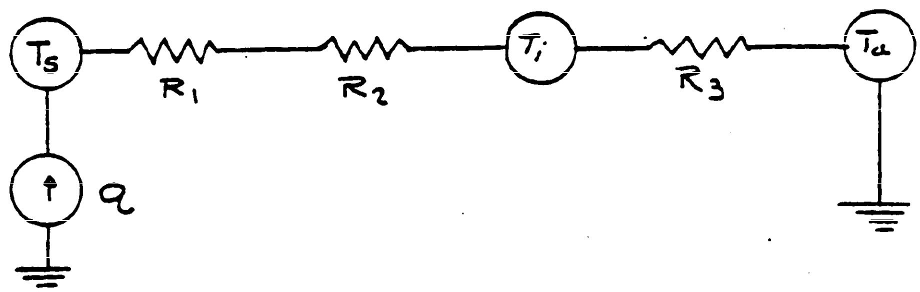


FIGURE 28. Thermal Analog Circuit

T_s = source surface temperature
T_i = convecting surface temperature
T_a = coolant temperature
R1 = thermal resistance source to sink
R2 = spreading metallic resistance of sink
R3 = resistance due to convection

Extruded bars of aluminum are available in many shapes, and can be machined easily for fins. Some suppliers furnish data on convective heat dissipation of these bars.

To estimate a value of R2 it is necessary to assume a field pattern of the heat flow, which is often quite complicated. Precise calculation is difficult, and unnecessary since all metals used for the purpose have high thermal conductivity. Steel is nearly as good as aluminum and is cheaper and sometimes easier to work, particularly in cutting and forming dies.

After R1 and R2 are determined, T_i is calculated and the computation of R3 corrected by recalculation of the heat transfer coefficient.

The design procedure is illustrated by the following example:

Example 8-9: Example of fin design. A power transistor has a mounting flange 1.0 inch in diameter. Applicable thermal data gives R1 as 0.4°C/watt with a mica insulating washer and silicone grease on the mounting surface, at a specified torque on the nut. Air temperature is 50°C. Case temperature must not exceed 75°C. The dissipation is 15 watts.

Design a suitable fin.

$$R1 + R2 + R3 = \frac{75 - 50}{15} = \frac{5}{3}\,^{\circ}\mathrm{C/watt}$$

$$R2 + R3 = \frac{5}{3} - 0.4 = 1.27^{\circ}\mathrm{C/watt}$$

Consider the fin as a circular plate with the transistor in the center. From the free convection nomograph Figure 10, with C for vertical plates, assuming ΔT = 15°C, L = 4 inches, h = 0.045 watt/sq. in. -°C. Compute required area, assuming convection from both sides of the plate.

$$h A \Delta T = q$$

$$(0.045)(2\pi D^2/4)(15) = 15$$

$$0.0706D^2 = 1$$

$$D = \sqrt{\frac{1}{.0706}} = \sqrt{14.2} = 3.76 \text{ in.}$$

The tentative value of R3 is 15°C watts = 1°C/watt. The fin spreading resistance must therefore be less than 1.27 - 1 = 0.27°C/watt.

The heat flow pattern will be chiefly radial in the fin, but the heat flow decreases progressively due to convection. The problem can be formulated with considerable accuracy and can be solved by a computer program, but such refinement is not justified. See Reference 6, page 232, where a solution is given in terms of Bessel and Hankel functions.

It is assumed that the heat flow pattern is radial from the mounting flange radius to the outside radius. R_2 is computed for this pattern:

$$0.27 \doteq \int_{0.5}^{1.88} \frac{dr}{2\pi Bkr} = \frac{0.16}{Bk} \ln (3.76)$$

$$Bk = \frac{(0.16)(1.33)}{0.27} = 0.788$$

For pure aluminum K = 5.8 watt/in.-°C and the thickness B = .136 inches. For steel K = 1.1, and B = .715. For copper k = 8.8, and B = 0.09 inches. American or Brown & Sharpe No. 7 or 8 Gauge aluminum sheet or No. 11 Gauge Copper would be suitable for the fin.

The resistance due to the complicated heat flow directly under the transistor is neglected because of the short flow path.

For convenience the fin may be made square or rectangular with an area of $\pi d^2/4$ = 11 or 12 sq. in. and the edges may be bent to 90°. The fin and the bent edges should be placed vertically. The transistor should be locked at the plate center. The fin should be oxidized or painted flat black, and roughened by wire brushing or rough grinding. A bright shiny surface should be avoided.

If the convective surface area required is so large that a single sheet is objectionable, the effective area may be increased by soldering or brazing parallel fins at 90° angles on a base sheet. Heat transfer textbooks (References 6 & 8) devote considerable space to fin theory. If parallel fins are closely spaced the air velocity in free convection is reduced in the trough so formed. In general, if the trough is 3/16 inch or wider, the effectiveness is not reduced and the entire developed area is effective.

Sometimes it is desirable to machine finned heat sinks from solid blocks by milling parallel grooves. The mass of material has a large heat capacity which filters thermal oscillations such as those which may be present in low frequency pulsing circuits.

The fin surface also removes heat by radiation. A black surface and fine grain roughness is preferable to a bright, cold rolled finish. If paint is used it should be flat black. Copper is readily oxidized by a sodium sulfide solution. Aluminum is usually anodized electrolytically.

8.3.5 Thermal design and packaging of assemblies and subassemblies. Each assembly should be designed to reject its heat through specific paths to the ultimate sink. Further, heat transfer into adjacent assemblies should be minimized. In general, densely packaged assemblies have small surface areas which result in high unit heat dissipations. Assemblies are severely penalized if they are required to act as their own intermediate thermal sinks to a specified environmental temperature and, connections to external sinks are recommended. For example, at a unit heat dissipation of one watt per square inch, the temperature rise of a small box in free air will be of the order of 70 to 90°C. If each assembly were required to have sufficient surface area to limit the temperature rise to a reasonable value, the size would have to be increased and miniaturization would not be practical. (See earlier section on circuit card assemblies).

It is recommended that each assembly be provided with a thermal connection (sink connector) which matches the thermal resistance of the cooling system. Each connection should have a thermal resistance proportional to its share of the total heat load. Each assembly could then be rated for thermal performance in terms of "sink connector" temperature. This would provide an accurate method for reliable prediction of thermal performance of standard assemblies and permit the interchangeability of such units.

The primary mode of cooling within assemblies must be selected so as to provide a path of low thermal resistance from the heat-producing parts to the sink. For the reasons previously mentioned, metallic conduction is recommended.

When designing the conductive heat flow paths, care must be taken to avoid the transfer of heat from hot parts to temperature sensitive parts. Capacitors and fuses in particular, must be protected from this type of thermal interaction. Temperature sensitive parts should have high thermal resistance paths to heat sources.

Radiant heat transfer also causes thermal interaction. Radiant heat flows in straight lines, and each surface transmits some heat to each other surface at a lower temperature which it "sees." In close packed equipment it is very difficult to calculate for each heat source the configuration factor F_a in the radiation equation 8-32, and detailed analysis of radiant heat exchange is practically impossible. Probably the best practice is to select parts surfaces so as to control it. Surfaces of temperature sensitive parts should be highly reflective, or should have reflective, heat shields interposed between them and radiating hot parts. Surfaces intended to absorb heat for transfer to the sink, and heat sinks in general, usually should be of highly absorptive type. Radiant heat transfer is potentially objectionable. It is not very effective unless large temperature differences exist. However, in a good thermal design, temperature differences are minimized. The effect of radiation can usually be neglected, except for possible heating of temperature sensitive parts.

8.3.5.1 Printed circuit cards. P.C. Cards built on substrates of electrical insulation material have chiefly the wiring and plug in fittings available for conductive cooling. Since the substrate is a poor heat conductor, very little is gained by conduction to supporting structure through clamped surfaces. Conduction through the wiring is discussed in paragraph 8.3.1.

Free convection must be well developed when insulating substrates are used. At least one-quarter inch spacing between cards should be used. Cards should be placed vertically. Wide spacing of cards tends somewhat to reduce heat transfer by allowing a thick laminar flow layer to develop. Spacing narrower than one-quarter inch encourages heat interchange among the cards by conduction through the air.

Cards having smaller dissipations should be located below those with higher dissipations. The general rules given in paragraph 8.3.1, substituting the word cards for parts, apply to assemblies.

Cards built on conductive substrates such as beryllium oxide and anodized aluminum can and should be conductively coupled to the supporting structure by clamping the edges. In this way each card has a direct conduction path to the sink, free convection and heat interchange are minimized, and placement is not critical. The entire supporting structure must be made as a thermally bonded entity leading to the ultimate sink. When natural cooling means are used for P.C. cards, conductive substrates are a practical necessity for unit heat dissipation of more than one-half watt per sq. in. of card surface.

The equipment enclosure should be designed to permit free access of air at the bottom and egress at the top. All air passages should be from one-quarter to one-half inch in width. Larger air passages between enclosure walls and equipment assembly should be avoided.

8.3.5.2 Composite cards. When C. Card structures are used, the spacing between cards is governed by the height of the discrete parts mounted thereon. Free convection heat transfer from these parts is somewhat reduced since they are not in a uniformly flat surface.

Power transistors and resistors in C. Card assemblies may require conductive paths in the form of jackets or caps thermally bonded to the supporting structure. It should be remembered that heat can be conducted in the same way as electric current, and that thin insulating washers of beryllium oxide or mica introduce a relatively low thermal resistance.

The electrical leads in C. Card assemblies should be as heavy as feasible to improve conduction cooling.

8.3.5.3 Conventional chassis. Conventional chassis are almost always constructed of metal, and therefore can serve as heat conductors as well as extended surfaces for convection cooling. Heat sources should be thermally connected to such chassis. When chassis are used in rack type cabinets, they can transmit some heat to the supporting structure, which further extends the convective surface. In a thermal survey of a large U.S. Navy radar it was found that the amount of heat dissipation to compartment air, per cabinet, varied from 6% to 68%. The number of cabinets was ten. Since this equipment was designed for closed forced air heat transfer to a water cooling system, the figures indicate poor cooling design. The significance of these percentages is that much of the heat was spread through the cabinet structure and convected from the enclosure to the compartment air, or by conduction through mountings to the steel deck. The fact that 50% or more of the heat can be removed in this way, without special attention to such design, shows the possibilities of this means of heat transfer. Unfortunately, the thermal resistances involved are large and overheating of parts can result. Further, shipboard disposal of heat by this method may overload the air conditioning system. This approach is recommended only when unit heat concentrations are low.

When metal chassis are used, all parts with heat dissipations greater than 0.5 watts/sq. in. should be conductively coupled to the chassis. Resistors and solid state devices can be clamped to chassis members by close fitting metal jackets or caps with silicone or other heat conducting compound, or aluminum foil sandwiched in the contact. Electron tube shields should make good thermal contact with chassis members. Transformers should be securely bolted down with thermal compound in the joint. Power transistors secured to chassis members must usually be electrically insulated therefrom, but care should be taken to minimize the contact thermal resistance. Flexible copper or aluminum straps are sometimes used to conduct heat from an external anode magnetron to the chassis. It is feasible to obtain thermal contacts with resistances of 1°C/watt and less, while still preserving electrical isolation for several hundred volts, by using washers of beryllium oxide or mica.

Some rotating structures such as gyroscopes can be thermally coupled to a stationary heat sink by gaseous conduction through concentric gaps surrounding the bearings. A thermal resistance of 0.5°C/watt was obtained with a gap area of 20 sq. in. in air. The use of heat conducting silicone grease as a lubricant reduces the thermal resistance through ball bearing by a factor of 2 to 4. (Reference 16)

8.3.5.4 Transmitters. Radio, sonar, and radar transmitters, unless they are low power equipments, usually have output devices dissipating significant powers which result in high unit heat concentrations in relatively small sections of their cabinets. Local hot spots can result! It is usually inadvisable to attempt to conduct the heat into other sections of the cabinets because of the possibility of overheating other parts. Thus, only relatively low powered equipments should use natural cooling. In certain instances, microwave transmitters can dissipate significant quantities of heat into the waveguides.

8.3.6 Design of cabinets and enclosures. Enclosures for electronic equipment cooled by natural methods serve to channel the air flow, as well as for obvious purpose of protection and isolation. Reference 40 describes certain standard Navy cabinets including the CY-4516. A cabinet offers attractive possibilities as an extended convective surface. However, it is possible to develop uncomfortably high surface temperatures if heat sources are conductively coupled to the cabinet. For example, a panel 60 inches high by 20 inches wide will dissipate about 40 watts and 60 watts with temperature rises of 20 and 30°C respectively. With compartment air at 40 to 50°C (104 to 122°F) as it may be in some operating situations, cabinet surface temperatures of 60 to 80°C (140 to 176°F) are very objectionable to personnel.

When an enclosure is thermally connected to the heat sources and is used as an extended convective surface, the heat so dissipated is transferred to the air of the compartment. Such a design is feasible if the resulting compartment air temperature rise is not objectionable, if for example, the compartment is fitted with air conditioning equipment of sufficient capacity or well ventilated with cool outside air.

If the heat convected from the cabinet causes undesirably high temperatures, the cabinet should be decoupled from heat sources so far as thermal conduction is concerned. The cabinet should then be designed to take in air at the bottom and discharge the heated air at the top.

The maximum heat dissipation from enclosures should not exceed 0.25 watt/sq. in. Cabinet surface temperature rise above compartment air should not exceed 10°C.

When the enclosure is used for convective heat transfer, it must be conductively connected to chassis and supporting structure. Most of the thermal resistance is in joints. All metal to metal contacts must be clean, smooth, of as large area as possible, and with high contact pressure. Rivetted joints with aluminum give thermal conductances of from one to four watts/sq. in. -°C depending on metal thickness and rivet spacing. The material of the rivet has little effect. Increasing metal thickness increases the conductance. Thickness of 0.1 to 0.2 inch are desirable. Bolted joints can be designed from the contact resistance data in paragraph 8.3.3.1.

Enclosures should have openings proportioned to the entrance and exit volume air flow rates and pressure drops smaller than the bouyant pressure of the heated air. It is possible to carry out detailed calculations to determine suitable entrance and exit areas, but these are time consuming and generally not necessary. As a rule of thumb, the holes or louvres should be at least 3/8 inches diameter, or 0.1 sq. in. area. The total area of entrance hole can be determined from Equation 8-44:

$$A = 0.58 \frac{q}{\rho(T2-T1)} 2/3 \qquad (8\text{-}44)(D.E.)$$

where A = area, sq. in.
q = watts dissipated into air
T2 = average internal air temperature °C
T1 = entering air temperature °C
ρ = density of air at T2 lb./cu. ft.

The temperature of the boundary-layer as it leaves the hot surface is, of course, very non-uniform. But, if this air becomes mixed to a uniform temperature, it will have a "mixed mean" temperature rise, which can be evaluated, for commonly-encountered conditions as follows:

$$T2 - T1 = 0.4\ q/Ah$$

where A = total area of convectin surface inside of the enclosure sq. in.
h = average free convection heat transfer coefficient, watts/sq. in.-°C.

The temperature difference (T2-T1) should be less than 35°C.

These design principles are illustrated by the following examples:

Example 8-10: A cabinet 6 ft. high x 2 x 1.5 ft. containing electronic equipment which dissipates 1000 watts is in a compartment with air at average temperature of 37°C. Maximum allowable equipment temperature is 75°C. Taking the thermal conductance of joints as 4 watts/sq. in. -°C and considering the possibilities of conductive coupling between electronic chassis and cabinet, a total contact area of 10 sq. in. results in 40 watts/°C, a resistance of 0.025°C/watt. This could be reduced further by the use of copper braid or other heat conductors brazed or soldered at appropriate locations. In this example it is assumed that an effective thermal resistance of 0.02°C/watt can be obtained by bonding the cabinet to chassis.

From the convection nomograph Figure 10 a dissipation of 0.015 watts/sq. in. can be reached with a surface to air temperature difference of 10°C. The cabinet surface is (6) (2+1.5+2+1.5)(144)=6000 sq. in. giving a dissipation of 90 watts. The cabinet surface temperature would be (0.02)(90) +10+37=49°C (120°F). This is not excessive, but the thermal bond between cabinet and heat sources should not be much better than the 0.02°C/watt stipulated. With 1000 - 90 = 910 watts absorbed by air passing through the cabinet the hole area must be,

$$A = \frac{(0.58)(910)}{\rho\,(T_2-37)} 3/2$$

Since the air density is 0.071 at 40°C and 0.065 at 65°C, A must be of the order of $7600/(T_2-37)^{3/2}$ sq. in. For any reasonable air temperature difference, T2 at say 50°C, an area of nearly 160 sq. in. is required. With a cabinet base of 2 x 1.5 ft. the opening should be 3 inches high.

It appears quite unreasonable to cool this cabinet entirely by free convection.

Example 8-11: A new cooling design provides for 700 watts to be transferred by liquid coolant, requiring only 300 watts to be removed by convection. Again assuming 90 watts convected from the cabinet, the ventilating openings are,

$$\frac{(0.568)(210)}{(0.065)(T_2-37)}^{3/2} = \frac{1830}{T_2-37}^{3/2}$$

For an air temperature rise of 13°C the required area is 140 sq. in. which is reasonable for such a cabinet.

Assuming T2 = 50°C and 300 watts removed from the electronics by convection, Ah = (0.4)(300)/13 = 9.23 watts/°C. With parts temperature at 75°C, it is feasible to remove 0.2 watt/sq. in. of equipment surface. The electronic equipment must thus, have an effective area of 9.23/0.2 or about 50 sq. in.

The following example illustrates the effect of an air conditioning unit on thermal design of equipment.

Example 8-12: A van 6 x 8 ft. with a 7 ft. ceiling contains 6 equipment cabinets like the one in the previous examples. Solar heat absorption is estimated at 5000 watts and personnel body heat at 600 watts. The total heat load is thus, about 12,000 watts. One ton of refrigeration will handle 3520 watts, so a four ton unit is specified to accept air at 50°C and return it at 0°C.

Since chilled air is available, the thermal resistance from electronics to cabinet walls should be made as low as possible. Assume the same as before 0.02°C/watt.

With a surface to air temperature difference of 20°C, a heat transfer rate of 0.03 watt/sq. in. can be obtained. The cabinet could thus, dissipate (6000)(0.03) = 180 watts. Cabinet temperature is thus, (0.02)(180) +20 + average air temperature. The average compartment air temperature is assumed at 20°C, so the cabinet surface is at about 44°C.

With the electronics at 75°C and air at 20°C, a dissipation of 0.5 watt/sq. in. for small parts is given by the free convection nomograph. The electronics will therefore, operate cooler than 75°C for the same area as previously determined.

Air entrance area required is:

$$A = 0.568 \frac{1000 - 180}{\rho(T_2 - T_1)}^{3/2}$$

With exit air at 50°C and entering air at 20°C, A = 40 sq. in.

8.4 Electronic equipment in spacecraft. Since spacecraft operate in an environment which, from a thermal standpoint, may be considered to be an obsolute vacuum, there is no heat transfer from the outside surface of the spacecraft by conduction or convection. The only remaining method for dissipating heat is by radiation to the heat sink of space, which has an effective temperature near absolute zero, about 4°K. The outside surface also receives heat input by radiation from the sun and from the earth if it is nearby. However, the surface of the sun has an absolute temperature of approximately 6000°K, so that the greater part of its radiant energy is of relatively short wavelength. On the other hand, the temperature of the surface of the spacecraft can and must be relatively low, so that it radiates energy at much longer wavelengths in the infrared range. Now, although absorptivity and emissivity are always equal at the same wavelength, they can differ greatly when dealing with radiation of vastly different wavelengths. Materials exist, notably special white paints, for which the absorptivity for solar radiation is much lower than the emissivity for infrared radiation. By proper selection of the finish, or by applying stripes of different finishes, a considerable degree of control can be exercised over the external temperature, even when heat is generated inside the spacecraft.

It is of interest to note that unmanned spacecraft are often given a fairly high rate of rotation (such as 180 rpm) for stabilizing purposes. To the degree that the spin axis is normal to the sun-spacecraft line, the rotation will even out the temperature differences which would otherwise exist between sunlit and shaded areas.

Heat dissipated by parts within spacecraft must be transferred to the outer surface. This can not be accomplished by natural convection, even if the spacecraft is pressurized, due to the absence of gravity. Forced convection by a fluid, preferably a liquid, is possible. Effectiveness can be increased if the fluid is circulated to an external radiator which is at least partially shielded from the sun. However, such a system requires a circulating pump and imposes penalties due to weight, power drain, additional heat input to the pump motor, and severe reliability problems, particularly in the case of unmanned spacecraft which must operate unattended for long periods.

The most desirable methods for transferring heat from internal electronic parts to the spacecraft surface are by conduction through solid material or by the use of heat pipes. Where the inevitable joint interfaces occur, their thermal resistances will be considerably higher than at normal atmospheric pressure, due to the absence of conduction through air in the narrow interstitial spaces between the contacting bodies (unless the spacecraft is pressurized). The table below from Reference 33, illustrates this point. (See also References 31 and 32).

TABLE VII. Thermal-Contact-Resistance Preliminary-Design Guidelines

Description	Environmental pressure	Approximate interface pressure, psi	R $(^{\circ}C)(in^2)$/watt
Small stud-mounted components (such as stud-mounted transistors)	Sea level	5,000	0.05
		500	0.50
	High vacuum	5,000	0.08
		500	0.80
Mounting feet of equipment with contact areas of about 1 in^2	Sea level	1,000	0.5
		100	1.0
	High vacuum	1,000	2.0
		100	5.0
Large surface contact areas	Sea level	100	1.0
		10	3.0
	High vacuum	100	7.0
		10	20.0

9. THERMAL DESIGN OF FORCED AIR COOLED ELECTRONIC EQUIPMENT

9.1 Theory. The practical limits for natural cooling methods applied to electronic equipment are 0.5 watts/sq. in. unit heat dissipation and 3.0 watts/cu. in. heat concentration. These limits apply generally to a package, an assembly, and to individual components.

When these limits are exceeded it becomes necessary to incorporate into the thermal design some artificial means of transferring heat from the sources to the ultimate sink. This requires the expenditure of energy, usually in a non-thermal form to drive the heat transfer system. Unfortunately, this driving energy usually itself becomes a part of the heat load, which must be transferred in addition to that produced by the electronic equipment.

Heat can be transferred by forcing a cool fluid from the heat sources to the sink. Since air is a generally available fluid in many environments, is cost free, non-toxic, and has reasonably high heat absorbing ability, it is a useful heat transfer fluid. Forced air cooling is accomplished by using fans or blowers to drive air through the electronic equipment. Obviously, the air at every place in the equipment must be at a lower temperature than the hot surface over which it flows, and it must be at a higher temperature than that of the sink surface.

Heat transfer by forced air depends primarily on the phenomenon of forced convection. However, the natural modes of heat transfer by conduction and radiation are always operating and often must be considered in the thermal design. In general, conduction is very helpful and maximum use of it should be made, within the constraints of weight and cost, whereas radiation is not very effective and sometimes causes local overheating. The recommendations given in Chapter 8 on Natural Cooling Methods regarding conduction and radiation are equally applicable to Forced Air Cooling.

Forced convection is a complex phenomenon involving fluid mechanics as well as heat conduction. When a fluid moves with respect to a solid surface, viscous drag requires that the component of velocity parallel to and at the surface be zero (Boundary layer). The velocity component normal to and at the surface is also zero. In forced convection the coolant is directed as much as possible parallel to the surface, but normal or oblique impingement cannot be avoided completely, particularly when small objects such as electronic parts are involved.

The parallel component of coolant velocity increases continuously with distance from the surface from zero at the solid surface to a value called the free stream velocity. For low values of free stream velocity the fluid moves smoothly as if in layers slipping on each other. As free stream velocity increases this smooth flow becomes disturbed, energy is interchanged between layers of the fluid, and the local velocity develops components which are not parallel to the main flow. These turbulent phenomena were first scientifically described by Osborn Reynolds in 1883. He demonstrated the effect visually by injecting fine streams of colored fluid into water flowing through a glass pipe. The familiar appearance of smoke rising from a cigarette or chimney shows the transition from laminar to turbulent flow.

Dimensional analysis of the physics of fluid flow discloses a dimensionless product of parameters known as Reynolds Number, (Re):

$$Re = \rho \frac{DV}{\mu} \tag{9-1}$$

Where ρ = fluid mass density
V = fluid velocity
μ = fluid dynamic viscosity
D = a so called characteristic dimension

These parameters require qualification. The simplest problem to analyze is fluid flow through a long, circular tube. In this case, V is the bulk or average velocity, volume flow rate divided by tube area. D is the tube diameter. For very high values of V, a gas becomes compressed and its density varies with position in the tube, but in most applications of forced air cooling the air can be considered as incompressible.

The characteristic length D is a troublesome parameter, difficult to determine by physical reasoning in any but the simplest configurations. It is generally evaluated by correlating experimental data with theoretical equations and will be discussed in detail in each appropriate subsection for specific configurations.

For values of Re less than approximately 2000, fluid flow of air in tubes is entirely of the smooth, laminar type. Above Re about 2000, the vortices of turbulent air flow begin to develop, and there is a range from Re = 2000 to 10,000 where the type of flow becomes increasingly turbulent. For values of Re above 10,000 it can be safely assumed that turbulent flow is completely developed. The transition range presents serious problems to designers because the degree of turbulence depends to a large extent on the type of fluid-solid interface. Actually it is the local value of Re at each point that determines the type of flow, but it is impossible to make such a fine scale analysis of the problem. The designer must of necessity use bulk or gross values of the parameters. It frequently happens that a heat transfer device designed for Re in the transition range either does not initially work as planned, or ceases to work as planned after long service has resulted in changes in the nature of the surface.

When a cool fluid passes over a hot surface it becomes heated. At the surface the fluid and the surface are at the same temperature because the fluid is stationary there. At a distance from the surface the fluid temperature decreases because heat flows by conduction through a layer of fluid. This is true even when turbulent flow is fully developed because there is always in effect a thin layer of stagnant air near the surface. Temperature and velocity are really continuous functions of position and must be described by differential equations. It is convenient, however, to use the concept of a laminar layer or sub-layer of a definite thickness.

Since the fluid is moving, except at zero distance, the fluid temperature gradient with distance from the surface is steep near the surface and levels off to zero where the temperature becomes the main stream vlaue. Analysis shows that the gradients of temperature and velocity are similar. The temperature gradient is illustrated in Figure 29.

$T = f(x)$ is a continuous function.

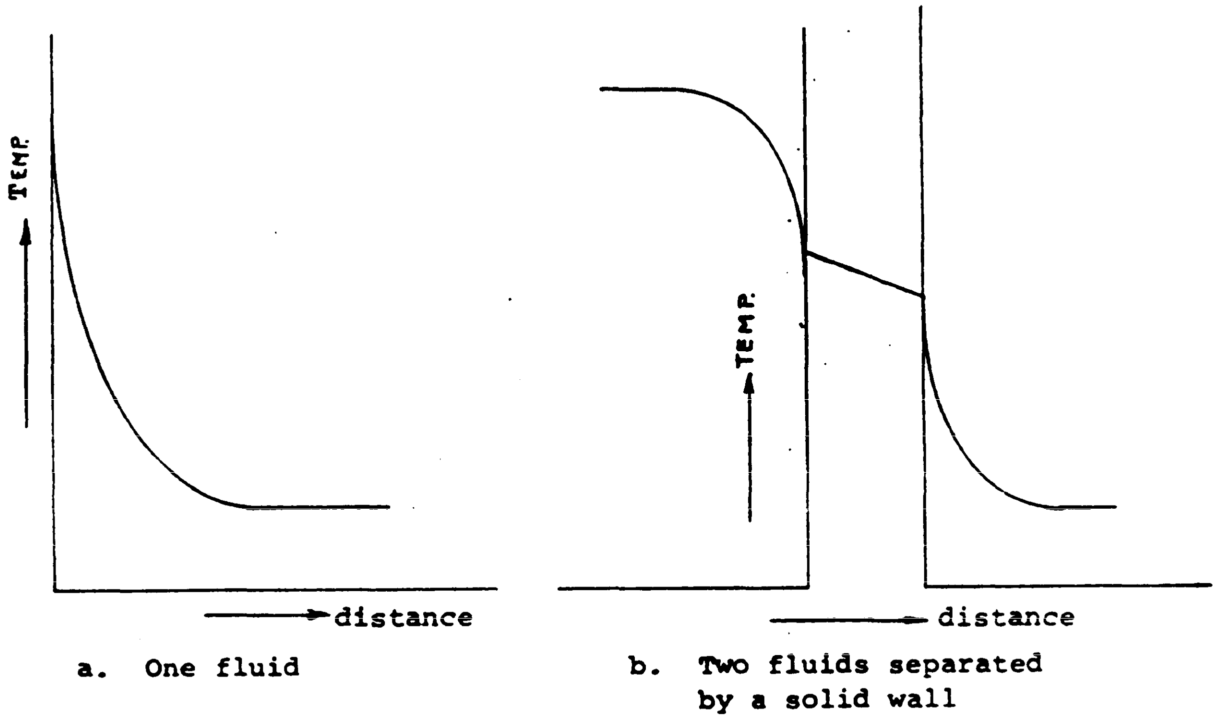


FIGURE 29. Temperature of Fluid Flowing Parallel to a Solid Heated Surface

The basic equation for convective heat transfer is:

$$q = h_c A \Delta T \qquad (9\text{-}2)$$

Where q = rate of heat removal, watts
h_c = convective heat transfer coefficient watts/sq. in. - °C (thermal conductance)
A = convective surface area, sq. in.
ΔT = effective temperature difference between surface and fluid.

Equation 9-2 is deceptively simple. The variables q and A are clearly defined. However h and the pertinent temperatures require careful analysis. The equation should be written for a differential area:

$$\frac{dq}{dA} = h_c (T_s - T_f) \qquad (9\text{-}3)$$

Where h has a specific value at each point, and T_s and T_f are the surface and fluid temperatures at that point. This differential equation must be integrated over the entire surface which requires a knowledge of how h, T_s and T_f vary with position on that surface. Only a few configurations have been found amenable to exact solution.

The engineering procedure has been to formulate functional relations among the large number of variables, and to identify the meaning of the questionable variables by correlating experimental data with the dimensional equation.

The significant variables, using the symbols of this handbook, are: h_c, D, k, μ, c_p, ρ, and V. The dimensional equation connecting these variables is:

$$f\left[\left(\frac{h_c D}{k}\right)\left(\frac{\mu c_p}{k}\right)\left(\frac{\rho V D}{\mu}\right)\right] = \text{CONSTANT} \qquad (9\text{-}4)$$

Where f ( ) means "some function of".

The value of h_c thus determined can be used in Equation 9-2.

Each expression in brackets is dimensionless. These dimensionless groups have been named for early researchers.

$$\frac{h_c D}{k} = \text{Nusselt No. (Nu)}$$

$$\frac{\mu c_p}{k} = \text{Prandtl No. (Pr)}$$

$$\frac{\rho V D}{\mu} = \text{Reynolds No. (Re)}$$

In order to correlate experimental data Equation 9-4 is written in the following form:

$$Nu = C Pr^m Re^n \qquad (9\text{-}5)$$

Where C is a dimensionless constant,
m and n are exponents to be evaluated.

Values of C, m, and n depend on the configuration and on the degree of turbulence developed.

Another effect which is particularly difficult to analyze for electronic cooling problems is the end effect. When a free air stream approaches a small object such as an electron tube, the flow is disturbed. If a surface long in the main flow direction is involved, the flow pattern settles down after a relatively short distance has been traversed. Over this distance the local value of h_c varies from zero to a constant value. The flow and heat transfer pattern in a simple configuration such as a long straight tube can be determined by analysis. However, electronic parts are so small that a steady flow pattern does not generally develop. This is easily and vividly shown by blowing cigarette smoke into the air stream entering a small operating assembly and comparing it with the smoke pattern of a long, smooth walled channel. The exact air flow pattern of electronic equipment defies theoretical analysis and the determination of the heat transfer coefficient must depend on empirical treatment.

As will be shown in the discussion of specific problem areas, h_c increases with air velocity, and is higher when the flow is turbulent than when it is laminar. This is an advantage, since the disturbed flow in electronic equipment results in a great deal of fine grained turbulence. In situations where the main flow is steady, as in a long duct, heat transfer can be enhanced by installing at the entrance turbulators such as screens or rods. While difficult to calculate quantitatively the effect of turbulators can be measured in test models.

Equation 9-5 is still in general use for correlating experimental data. Better correlations are obtained, however, by relating heat transfer and fluid friction. By introducing a new dimensionless product called the Stanton Number (St) an equation similar to 9-5 is derived.

$$St = C\ Pr^{-m}\ Re^{-n} \qquad (9\text{-}6)$$

Where $St = h/\rho V c_p$

Equations 9-5 and 9-6 result in different values of C, m, and n. Specific values of these constants applicable to different configurations and degrees of turbulence will be given in the appropriate subsections of this chapter as part of the design.

9.1.1 Environmental effects. Heat transfer by forced convection with air is affected by environmental influences on the physical properties of air. The significant environmental factors are the pressure, temperature, and moisture content of the air. Thermal conductivity, viscosity, specific heat, and density vary with these environmental conditions.

The thermal conductivity of air k, as of all gasses, is nearly independent of pressure, until the pressure becomes so low that the probability of collisions between molecules is greatly reduced. At pressures less than 15 or 20 mm of mercury (above 25 km or 82,000 ft. altitude) k decreases directly with pressure. It also decreases with decreasing temperature because of the decreasing molecular velocity.

The dynamic vicosity, μ, of air increases with rising temperature and is nearly independent of pressure.

The specific heat at constant pressure, C_p measured in watt-sec/lb-°C is independent of pressure, and increases very slightly with temperature.

The density of moist air can be computed from equation 9-7:

$$\rho = 0.0807 \left[\frac{273}{T+273}\right]\left[\frac{p-0.378e}{14.7}\right] \qquad (9\text{-}7)$$

Where ρ = lb/cu.ft.
T = °C
p = p.s.i.
e = partial vapor pressure of the water content

The water vapor also absorbs heat, however, and has about four times the specific heat of dry air. For most purposes the effect of moisture content can be neglected, and air density computed from Equation 9-8, particularly since an effort is usually made to use reasonably dry air in electronic equipment.

$$\rho = 0.0807 \left[\frac{273}{T+273}\right]\left[\frac{p}{14.7}\right] \qquad (9\text{-}8)$$

Examination of Equations 9-5 and 9-6 in terms of the thermal properties of air shows that the heat transfer coefficient h_c, decreases as the 0.8 power of the pressure ratio, decreases slightly with increasing temperature, and increases slightly with increasing moisture content. For example, h_c decreases about 40% between one atmosphere and one-half atmosphere (18,000 ft. altitude). It increases 18% between 10°C (50°F) and 120°C (250°F).

9.1.2 Weight Rate of Flow. The amount of thermal energy absorbed by air is given by the following equation:

$$q = mc_p \, \Delta t_b = 7.62 \, m \, \Delta t_b \qquad (9\text{-}9)(\text{D.E.})$$

Where:

m = the weight rate of flow of air in lb/min
c_p = the specific heat in watt-min./(lb)(°C)
Δt_b = the air temperature rise in °C
q = the heat dissipated in watts

For air, m is given by:

$$m = Q\rho \qquad (9\text{-}10)$$

Where Q = the volumetric flow rate in cu.ft./min (Cfm)
ρ = the air density in lb./cu.ft.

The flow rate in cfm is equal to the average air velocity times the net cross-sectional area normal to the directional flow, or

$$Q = \frac{VA_{c_{net}}}{144} \text{ cfm} \qquad (9\text{-}11)(\text{D.E.})$$

Where:

V is the velocity of the flow (ft./min.)
A_c is the net cross-sectional area normal to the net direction of flow (in.^{2}).

The density of air is given by the following equation:

$$\rho = 1.5 \frac{p}{t+273} \qquad (9\text{-}12)$$

As stated earlier, the value of h_c, the convection coefficient depends upon several factors. Some factors (properties of the fluid, velocity of flow) appear in the dimensionless parameters (Nu, Re), while others (configuration, spacing of components, type of flow) determine the magnitude of the constants. Forced convection is, therefore, subdivided into flow inside of ducts and tubes of various cross-sections (circular, rectangular, annular) and flow outside of components of various cross-sections. The flow outside of components can be subdivided further into flow parallel and normal (crossflow) to the axis of the components. Components in crossflow cooling may be arranged either staggered or in-line.

9.1.3 Relationship between free and forced convection. There exists an interrelation between free and forced convection. The equation for free convection may be expressed as

$$Nu = C''(GR)^{\frac{n}{2}}\,(Pr)^{a} \qquad (9\text{-}13)$$

Comparison with equation 9-5 indicates that the square root of the Grashof number can be considered a special case of the Reynolds number. This theoretical result is in satisfactory agreement with the results of experiments.

9.1.4 Application to electronic equipment thermal design. Basic convective heat transfer theory and data on idealized shapes have been found to be applicable, with some modifications, to electronic design. The problem of cooling electronic equipment is made complex by the hot spots on the surfaces of the heat sources resulting from an uneven distribution of the internally generated heat. In crossflow cooling, the variation of hot spot temperatures along the surfaces of heat-producing parts is affected by the basic configuration and the part spacings. The small size of electronic heat-producing parts result in "end-effects" which tend to increase heat transfer coefficients, thus decreasing the thermal resistance. The heat transfer coefficients for forced-air cooled high temperature parts such as electron tubes are also higher than the common convective coefficients, because the former include radiation coefficients, while the latter are corrected for pure convection.

9.1.5 Flow within tubes or pipes. Figure 30 shows an axial section of a round pipe. The wall temperature, t_w, is constant at any section normal to the axis. Fully turbulent flow causes thorough mixing and results in a nearly constant bulk temperature, t_b, in the bulk of teh fluid. In the thin laminar film adjacent to the wall there is a steep and nearly uniform temperature gradient so that the film temperature, t_f, may be taken as an average value.

$$t_f = \frac{t_w + t_b}{2} \qquad (9\text{-}14)$$

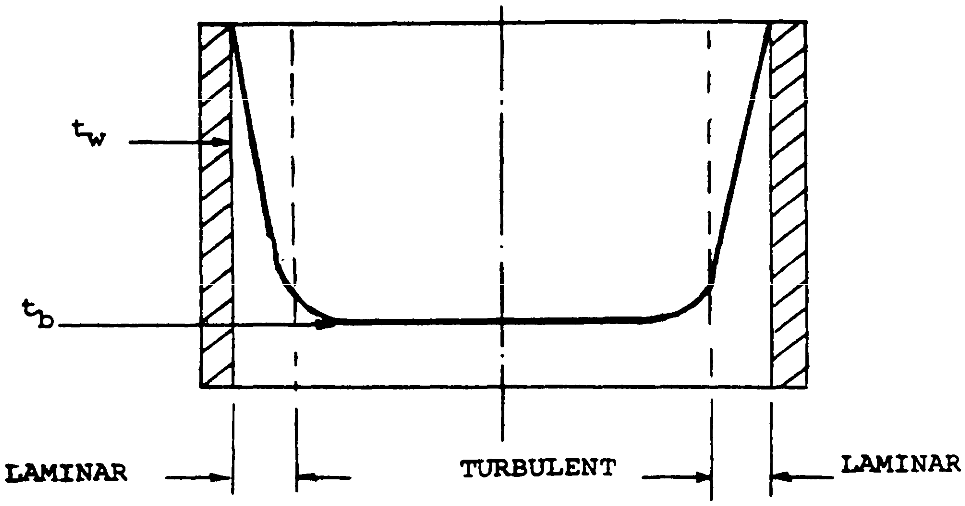


FIGURE 30. Fluid Temperature Distribution for Flow within a Round Pipe

If heat is to be exchanged, the temperatures must of course vary axially along the pipe. Since the heat transfer coefficient h_c is a function of fluid properties which vary with temperature, h_c will have a changing value along the pipe. It is necessary to evaluate h_c locally, compute the heat flow in a differential element, and integrate over the pipe length. Formal integration of the functions is impossible, however, so the pipe is treated by sections short enough so that temperature changes are relatively small and average values may be used.

For Reynolds numbers greater than 10,000 (fully turbulent flow), L/D ratio of the tube greater than 60, and with fluids having viscosities less than twice that of water, two equations have been found to be applicable.

$$h = 0.023\,(c_p)_b G_{max} \left(\frac{c_p\,\mu}{k}\right)_f^{-0.67} \left(\frac{DG_{max}}{\mu\,f}\right)^{-0.2} \qquad \begin{matrix}(9\text{-}15)\\(\text{D.E.})\end{matrix}$$

Equation 9-15 is based on film temperature.

$$h = 0.023 \frac{k_b}{D} \left(\frac{DG_{max}}{\mu_b}\right)^{0.8} \cdot \left(\frac{c_p \mu}{k}\right)_b^{0.4} \qquad \text{(9-16) (D.E.)}$$

Equation 9-16 is based on bulk temperature.

The subscripts b and f indicate values at bulk and film temperatures respectively. G_{max} indicates the largest value of mass velocity, normally occurring at the point of minimum cross-section. D is the inside diameter of the tube or pipe.

Using Pr = 0.69 and c_p = 7.62 watt-min/lb.-°C for air these equations become

$$h = 0.224 \mu_f^{0.2} \cdot \frac{G_{max}^{0.8}}{D^{0.2}} \qquad \text{(9-17)}$$

$$h = 0.0198 \frac{k_b}{\mu_b^{0.8}} \cdot \frac{G_{max}^{0.8}}{D^{0.2}} \qquad \text{(9-18) (D.E.)}$$

9.1.5.1 <u>Water cooling considerations</u>. Heat exchangers used in the design of forced-air cooled equipment often are of air-to-water type. The water temperature rise is often limited to 5°C maximum (from 35°C to 40°C). Because of this small temperature range, the physical properties of water can be assumed to be constant. Based upon 38°C, and Equation 9-16, both heat transfer coefficient equations take the following simplified form.

$$h = 0.036 \frac{G_{max}^{0.8}}{D_e^{0.2}} \qquad \text{(9-19) (D.E.)}$$

For ducts of non-circular cross-section, the equivalent diameter D_e for use in these equations is defined by:

$$D_e = \frac{4 \times \text{cross-sectional area}}{\text{cross-sectional perimeter}} \qquad \text{(9-20)}$$

9.1.6 Flow of gases parallel to plane surfaces, wires, and cylinders. The film coefficient for turbulent air flowing parallel to smooth plane surfaces is:

$$h = .055 \frac{k}{D} \left(\frac{DV \rho}{\mu} \right)^{0.75} \qquad \text{(9-21) (D.E.)}$$

Here D is the length of the surface and is limited to two feet; even if the length of the surface is greater, use two feet.

The heat transferred from a cylinder to a fluid which flows parallel to its axis will be little different from that between a plane plate and a fluid in parallel flow provided that the diameter of the cylinder is large. For cylinders of small diameter the eddies occurring in turbulent flow may be great compared with the cylinder diameter and the flow will become more similar to that perpendicular to the axis.

Heat transfer coefficients for turbulent flow of air parallel to flat plates and cylinders are expressed by:

$$\frac{hD}{K_{fluid}} = Nu = 0.0280 \ (Re)^{0.8} \text{ for } Pr = 0.71 \qquad \text{(9-22) (D.E.)}$$

or

$$Nu = 0.0272 \ (Re)^{0.8} \text{ for } Pr = 0.69 \qquad \text{(9-23) (D.E.)}$$

For laminar flow of air the heat transfer coefficients are related to other variables by:

$$Nu = 0.592 \ (Re)^{0.5} \text{ for } Pr = 0.71 \qquad \text{(9-24)}$$

or

$$Nu = 0.586 \ (Re)^{0.5} \text{ for } Pr - 0.69 \qquad \text{(9-25)}$$

9.1.7 Cylindrical shapes. For forced convection from cylindrical shaped objects, such as electron tubes, with air flow upward and parallel to the axis, and surrounded by a reflective flow baffle, a general equation is:

$$Nu = 0.313 \ (Re)^{0.6} \qquad \text{(9-26)}$$

The region tested was for Reynolds Numbers, Re, between 130 and 8,000. Experiments with air flow horizontal and parallel to the axis of subminiature tubes, mounted inside insulated white cardboard indicate that

$$Nu = 0.337\ (Re)^{0.55} \qquad (9\text{-}27)$$

The region studied here was for Reynolds Numbers, Re_b, between 130 and 5,000.

The Nusselt and Reynolds numbers were based upon the tube diameter. The inlet air temperature was used in Equations 9-26 and 9-27 for the evaluation of the properties of air.

9.1.7.1 Crossflow over cylindrical shapes. The equation for the film coefficient across a single wire or cylinder takes the form:

$$\frac{hD}{k_f} = C\left(\frac{DV\rho}{\mu_f}\right)^n \qquad \begin{matrix}(9\text{-}28)\\(D.E.)\end{matrix}$$

The constants C and n are dependent on the magnitude of the Reynolds number and are given in Table VIII. The air properties should be determined at the mean of the arriving air and surface temperatures (film temperature). The convective coefficient h is based on the difference in these temperatures. The arriving air velocity is the reference velocity.

TABLE VIII. Constants for Use in Equation (9-28) for Round Cylinders

(From References 6 and 43)

Reynolds Number $Re = \frac{VD\rho}{\mu'}$	C	n
0.4-4	0.891	0.330
4 -40	0.821	0.385
40 - 4,000	0.615	0.466
4,000 -40,000	0.174	0.618
40,000 -400,000	0.0239	0.805

9.1.8 Flow of air across cylinders of non-circular cross-section. The equation for the film coefficient for air flowing normal to the axis of cylinders of non-circular cross-section, is similar in form to that for cylinders of circular cross-section. The diameter of a cylinder with exposed surface equal to that of the non-circular cylinder should be used as the characteristic length in Equation 9-28. The constants C and N depend on the orientation of the cylinder with respect to the direction of air flow, and are given in Table IX.

Note that the constant C for the flow parallel to a diagonal of square cross-section is approximately twice that for flow parallel to a side. In general, the non-circular profiles offer better heat transfer than the circular ones.

The influence of mechanical disturbances of the flow at the surface may be seen from Table X. Constants C and n for air flow normal to cylinders with fins and grooves are given in Table X.

As much as 50% increase in the average coefficient of heat transfer was obtained by some experimenters when the turbulence was increased by passing the air through a grid. Similar effects were found with spheres.

9.1.9 Flow of air over spheres. For air flow over spheres, the film coefficient is:

$$\frac{hd}{k_f} = 0.37 \left(\frac{DG}{\mu_f} \right)^{0.6} \tag{9-29}$$

for Reynolds numbers, Re_f, between 17 and 70,000.

9.1.10 Flow of air across banks of cylinders. The relationship between Nusselt number and Reynolds number ($Nu = CRe^n$) for crossflow cooling of banks of cylinders, depends upon the number of rows and upon the longitudinal and transverse spacings. These factors affect the turbulence of the air stream, and, therefore, the heat dissipation by convection.

Note:

A bank of cylinders is a two-dimensional array, consisting of longitudinal rows and transverse rows. A longitudinal row consists of several cylinders in a straight line parallel to the direction of flow, whereas a transverse row consists of several cylinders in a straight line perpendicular to the direction of flow.

9.1.11 Correlation of data. The correlation of data for the flow of air across banks of cylinders of circular cross-section takes the same form as Equation 9-28. The Reynolds number $DV\rho/\mu$, may be written as DG/μ, where G is the mass velocity in pounds per hour-sq.ft. of flow cross-section. The value of G in the correlation is that obtained at the narrowest cross-section between the cylinders whether or not the minimum area occurs in the transverse or diagonal openings between the cylinders. Thus Equation 9-30 is used in the following form:

$$\frac{hD}{k_f} = \left(\frac{DG}{\mu_f} \right)^{n} \tag{9-30}$$

TABLE IX. Constants for Use in Equation 9-28 for Cylinders of Non-Circular Cross-Section

(From References 6 and 43)

Cross-Section (investigator A - Reiher B - Hilpert)	Description of Cross-Section	Description of Flow	Re Range	C	n
(A)	square	perpendicular to front face	2,500 - 8,000	0.160	0.699
(B)	square	perpendicular to front face	5,000 - 100,000	0.092	0.675
(A)	square	parallel to diagonal	2,500 - 7,500	0.261	0.624
(B)	square	parallel to diagonal	5,000 - 100,000	0.222	0.588
(B)	hexagonal	parallel to diagonal	5,000 - 100,000	0.138	0.638
(B)	hexagonal	perpendicular to face	5,000 - 19,500	0.144	0.638
(B)	hexagonal	perpendicular to face	19,500 - 100,000	0.035	0.782
(A)	thin, flat plate	perpendicular to flat	4,000 - 15,000	0.205	0.731
(A)	elliptical	parallel to major axis	2,500 - 15,000	0.224	0.612
(A)	elliptical	parallel to minor axis	3,000 - 15,000	0.085	0.804

TABLE X. Variations of "C" and "n"

Surface Disturbance	Re	C	n
None	400 - 6000	0.350	0.560
Longitudinal fin 0.1D thick, on front of cylinder	1000 - 4000	0.248	0.603
12 longitudinal grooves 0.07D wide	3500 - 7000	0.082	0.747
Same, with burrs	3000 - 6000	0.0368	0.86

The constants C and n depend on the distance between the cylinders the diameter of the cylinder and the configuration, i.e., staggered or in-line (see Figure 31).

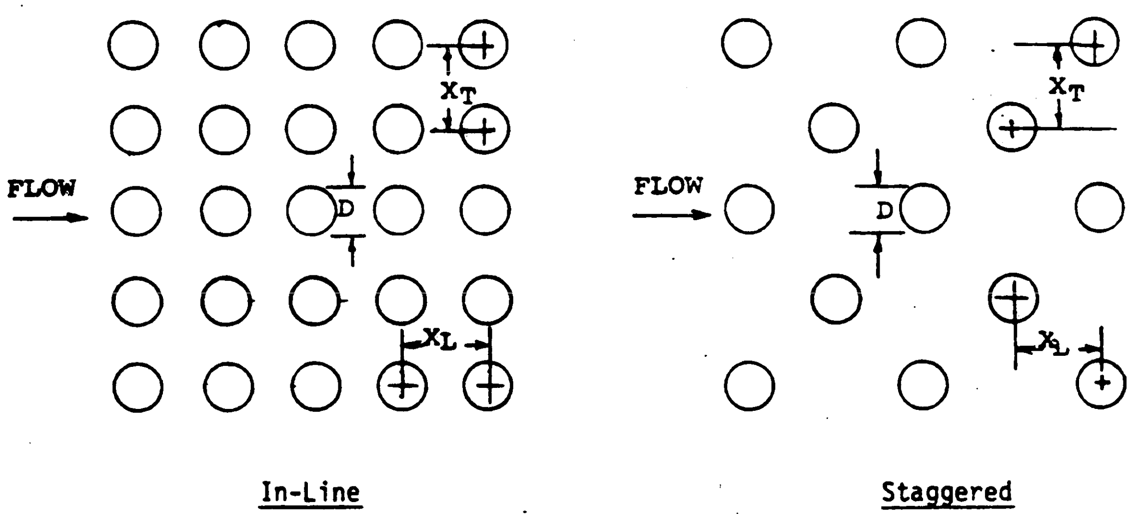


FIGURE 31. Banks of Cylinders In-Line and Staggered

For forced-air cooling of banks of cylinders, the values of C and n in Equation 9-30 depend upon the following parameters:

$$S_T = \frac{X_T}{D} \; ; \; S_L = \frac{X_L}{D} \qquad (9\text{-}30a)$$

where

S_T and S_L are configuration factors dependent upon cylinder spacing and diameter.
X_T is the transverse center to center distance between the heat sources in inches.
X_L is the longitudinal center to center distance between the heat sources in inches.
D is the heat source diameter in inches.

9.1.12 Effect of the number of transverse rows of cylinders on heat transfer coefficient. When calculations for flow across fewer than four transverse rows are based upon the data for flow across more than four transverse rows, appreciable error may result. The cylinder (part) surface temperature could run higher than predicted since there is a decrease in the heat transfer coefficient. For 2 to 4 transverse rows, calculations should be based upon data for a single transverse row which will yield conservative results; or calculations based upon the data for more than four transverse rows should be reduced by an appropriate factor as shown in Table XI.

In general, increasing the number of transverse rows of cylinders or pipes in the direction of air flow produces increased turbulence toward the rear of the bank. Hence, the heat transfer coefficients are greater for the cylinders in the rear. References 26 and 43 give values of C and n obtained by Grimmison to be used in Equation 9-30 for banks of pipes ten transverse rows deep. These values, however, are not directly applicable to electronic components and are replaced by the data obtained by Robinson (Reference 46) for cylinders with configurations and heat dissipations similar to those of electronic parts.

TABLE XI. Ratio of Mean Film Heat Transfer Coefficient for n Transverse Rows Deep to that for 10 Transverse Rows Deep

n	1	2	3	4	5	6	7	8	9	10
Ratio for Staggered Rows	—	.73	.82	.88	.91	.94	.96	.98	.99	1.0
Ratio for In-Line Rows	.64	.80	.87	.90	.92	.94	.96	.98	.99	1.0

Robinson has found that heat transfer coefficients for in-line and staggered arrangements of prismatic heat sources are the same for the third to the second last transverse row and slightly lower for the first, second, and last transverse rows (for cylindrical parts the first three transverse rows have lower heat transfer coefficients). Turbulators could be placed in front of the parts with low heat transfer coefficients to increase turbulence, which in turn will increase the rate of heat transfer.

9.1.13 Cylindrical heat sources. Robinson expresses the Nu_f; Re_f relationship in the form

$$Nu_f = CF\ Re_f^{\ n} \tag{9-31}$$

Where:

F is a configuration factor, dependent upon the cylinder geometry.

TABLE XII. Values of C and n for Use with Equation 9-31

(Re_f) ave.	C	n
1,000- 6,000	0.409	0.531
6,000- 30,000	0.212	0.606
30,000-100,000	0.139	0.806

By applying the factor F, all data were correlated with McAdams (Ref 26) data for cross-flow cooling of single cylinders in a free air stream. Values of C and n as determined by Robinson are given in Table XII, and expressions for F are given below.

Note:
These values give a curve of Nu vs. Re which falls on McAdams curve for a single cylinder in a free air stream.

1. For single cylinder in free stream: $F = 1$

2. For single cylinder in duct $F = \left(1 + \sqrt{\frac{1}{S_T}}\right)$

3. For in-line cylinders in a duct:

$$\left(1 + \frac{1}{S_T}\right) \cdot \left(1 + \frac{1}{S_L} - \frac{0.872}{S_L^{\ 2}}\right) \cdot \left(\frac{1.81}{S_T^{\ 2}} - \frac{1.46}{S_T} + 0.318\right) \cdot \left(R_{e_f}\ \left(0.526 - \frac{0.354}{S_T}\right)\right)$$

4. Staggered cylinders in a duct:

$$F = \left(1 + \frac{1}{S_T}\right) \cdot \left[1 + \left(\frac{f_1}{S_L} - \frac{f_2}{S_L}\right) R_{e_f}^{\;0.13}\right]$$

where:

$$f_1 = \frac{15.50}{S_T^2} - \frac{16.80}{S_T} + 4.15$$

$$f_2 = \frac{14.15}{S_T^2} - \frac{15.53}{S_T} + 3.69$$

S_T and S_L are as defined by Equation 9-30a. Experiments performed on the crossflow cooling of banks of electronic parts are in general agreement with those of other investigators. For example, the investigation of crossflow cooling of type 5902 subminiature tubes mounted in an insulated white cardboard duct showed that

$$Nu_f = 0.15\ (Re_f)^{0.697} \qquad (9\text{-}32)$$

for $S_T = 2.49$ and $S_1 = 1.33$. The region tested was for Reynolds numbers Re_f, between 800 and 6000. When the equation is plotted, the curve coincides with Robinson's curve for $S_T = 2.333$ and $S_L = 1.250$.

The following equation was determined for crossflow cooling of type 805 transmitting tubes

$$Nu_f = 0.067\ (Re_f)^{0.778} \qquad (9\text{-}33)$$

for $S_T = 1.57$ and $S_1 = 1.46$. The region tested was for Reynolds numbers Re_f, between 2500 and 15,000. This equation when plotted falls between Robinson's and McAdams' curves for the corresponding S_T and S_L values. Here higher radiation losses were incurred because of poorer insulation and the higher temperature level.

9.1.14 Prismatic heat sources. For the crossflow cooling of prismatic electronic components, with the side of the components perpendicular and parallel to the direction of flow, Robinson derived the following relationships. (For a single prism in a duct)

$$Nu = 0.446 \left[R_{e_o} \cdot \frac{1}{\frac{1}{6} + \frac{5A_n}{6A_o}} \right]^{0.57} \qquad \text{(9-34) (D.E.)}$$

(For Reynolds number R_{e_o} between 2500 and 8000.)

where

R_{e_o} = the Reynolds number based upon total duct flow area and prism side dimension

A_o = the total duct flow area

A_n = the net duct flow area (A_o less electronic part cross-sectional area)

In the free air stream and with the characteristic dimension $D_e = \frac{4d}{\pi}$ (diameter of a cylinder of equal surface area) the above equation reduces to:

$$Nu = 0.495 \, (Re)^{0.57} \qquad \text{(9-35)}$$

Comparing Equation 9-35 with Jakob's (Ref. 6) data, the maximum difference is only 12 percent of the given range of Reynolds numbers.

For the in-line arrangement an equation was not determined experimentally because of the non-linearity of several of the measured curves and differences in their slopes. (A plot is presented by Robinson in Ref. 27, page 47.)

For staggered arrangement the equation takes the form

$$Nu = 0.446 \left[1+0.639 \left(\frac{S_T}{S_{T_{Max}}} \right) \left(\frac{D}{S_L} \right)^{0.172} \right] \left[R_e \frac{1}{\frac{1}{6} + \frac{5A_n}{6A_o}} \right] \qquad \text{(9-36)}$$

where:

$S_{T_{Max}}$ refers to maximum value of S_T when there are different values in the same configuration.

9.2 Fan fundamentals.

9.2.1 Fan types and limitations.

9.2.1.1 Definitions. A fan is a machine for applying power to a gaseous fluid. For the purposes of this handbook, a fan is an air-moving device.

Fans are classified as either blowers or exhausters. A blower is a fan used to force air under pressure across the surface of configuration to be cooled. The resistance to the air flow is imposed primarily upon the blower outlet. An exhauster is a fan used to withdraw air under suction across the surface or configuration to be cooled. The resistance to air flow is imposed primarily upon the exhauster inlet.

Blowers are recommended for the forced-air cooling of electronic equipment. MIL-B-2307A - "Blowers, Miniature, For Cooling Electronic Equipment (10 to 500 CFM)" is applicable to most of the blowers used in cooling Naval electronics equipment.

9.2.1.2 General. Centrifugal and axial-flow fans or blowers are the two basic types. The type to be selected for a specific cooling problem is dependent on several factors, such as air flow and pressure requirements, efficiency, speed, space, the air ducting system, noise and fan characteristics.

Temperature rise of air in the fan is due to the increase of kinetic energy. Since it is also necessary to cool the motor driving the fan, there is a temperature rise due to the motor cooling. Usually this rise is small. In practice there are other factors, such as weight, size, noise rotational speed, and power which must be considered.

Weight must be considered in aircraft applications and even in shipboard equipment. Fan size is usually limited by space. For very large external blade diameters, blade tip speeds will be high, possibly exceeding the safe design limits. High rotational speeds can shorten bearing and brush life. If the incorrect size and type of fans are selected for a particular system to be cooled, inefficient operation can cause excessive power consumption.

9.2.1.3 Centrifugal fans. A typical centrifugal fan is shown in Figure 34. The three important parts are: the housing, containing the air inlet and outlet; the rotor containing the fan blades or vanes; and the external driving motor. The air enters the housing normal to the side through a single of double inlet, i.e., at one or both sides of the center of the rotor, and is discharged in a direction perpendicular to the axis of rotation.

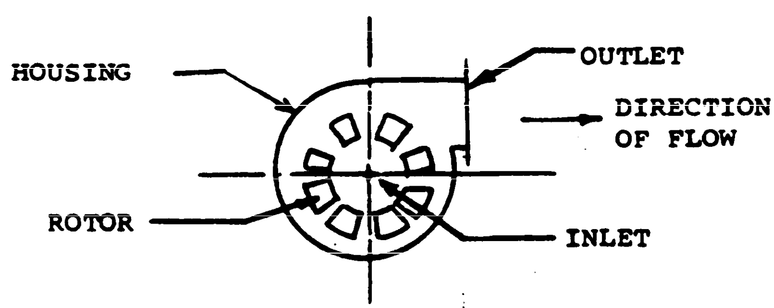


FIGURE 32. Typical Centrifugal Fan

The centrifugal fan is generally used where high pressures are required. Centrifugal fans can have a single or double inlet. For a given rotor speed and diameter, a double-inlet fan of the same type has greater air capacity and roughly the same pressure-producing ability as the single inlet unit. Centrifugal fans, capable of producing high air-pressure, have a special rotor construction to permit operation at the high peripheral blade speeds necessary for high pressure generation.

Centrifugal fans may be divided into three groups according to the shape of the blades: (1) forward-curved blades, (2) radial blades, (3) backward-curved blades.

Forward-curved blades curve away from the radial direction in the direction of rotor rotation. Backward-curved blades curve opposite to the direction of rotation. For a given rotor speed and diameter, the forward-curved blades have the greatest pressure-producing ability and the backward-curved blades the smallest. Hence, when fan size is a limiting factor, fans with forward-curved blades should be used. Fans with forward-curved blades have, however, relatively poor stability of operation and considerably less flexibility of control in aircraft applications. Fans with backward-curved blades do not have the above disadvantages, and fans with radial blades are also considered satisfactory in this respect. In mechanical strength, the radial type centrifugal fan is superior to the other two types. Considering the advantages and disadvantages of the three types of centrifugal fans, the radial and forward-curved blade fans are best suited for cooling of electronic equipment. In most applications the forward-curved blade fan would prove the best because of its greater pressure producing ability for a given speed and size. However, care must be exercised to avoid the possibility of over-loading the drive motor, because of the inadvertant reduction in flow resistance. The radial type fan should be employed when appreciable pressure is required from a reasonably small unit. The drive and the load must have fairly constant control characteristics.

9.2.1.4 Axial-flow fans. Axial-flow fans are generally one of two types: the so-called propeller type, shown in Figure 35, and a more efficient design shown in Figure 36 commonly considered to be more truly an axial-flow fan.

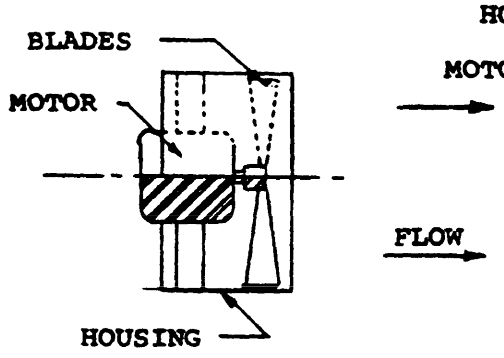


FIGURE 33. Propeller Type Fan

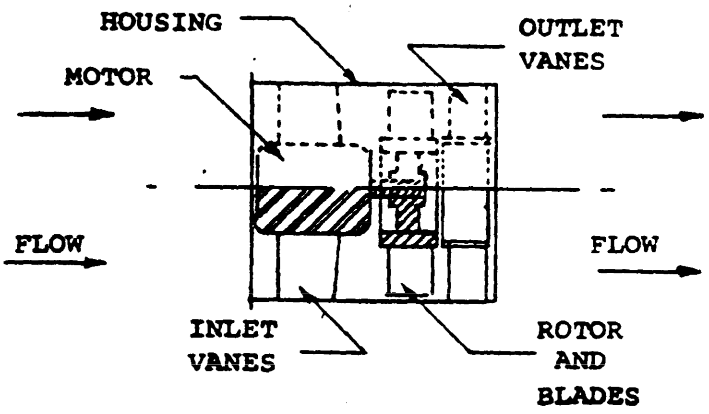


FIGURE 34. Axial-Flow Fan

There are several types of construction, but all are typified by the straight-through flow of air. The axial-flow fan is a more efficient design and is a higher pressure fan than the propeller type.

The following apply to fans with 60 hz motors:

Propeller type fans are employed usually as circulating devices, while axial-flow fans are applicable to cooling of electronic equipments requiring high volume air flow at moderate to low system resistances. When appreciable increase in static pressure is required, several rotors are usually placed in series. Axial fans of this type are thus called multi-stage fans.

Axial fans are suitable for cooling of electronic equipment having low flow resistance and constant volumetric requirements and pressure performance. Because of their configuration, axial-flow fans can be installed in air ducts without change in flow direction.

.With 400 to 800 hz motors, the performance of axial flow fans is greatly increased due to the increased motor speed. With 60 hz, the motor speeds are generally limited to 3600 RPM and the tip velocity of the fan blades is low. (Universal series wound motors cannot be used because of electrical noise and brush problems.) The 400 hz motors permit the operation of small (2 to 3 inch diameter) axial flow fans at 15,000 to 20,000 RPM with resultant high pressure performance comparable to large centrifugal blowers. These small high speed fans produce significant acoustical noise.

9.2.1.5 Comparison of centrifugal and axial-flow fans. Centrifugal fans are better suited to the cooling of electronic equipment than axial-flow fans when only 60 hz is available because of their favorable proportions and control. However, the externally mounted motor requires cooling and occupies valuable space. Axial-flow fans are less desirable because of their inferior control characteristics, and the complicated multistage configurations needed to produce appreciable air pressure. Single stage units are no more complicated than centrifugal fans, but they deliver air at very low pressures. Because of their higher operating efficiency and greater volumetric capacity for comparable diameters, axial-flow fans deserve consideration. When designing for the consumption of minimum power, they are applicable to the cooling of large units, especially those dissipating 10 kw and over.

Since the major disadvantage of the axial-flow fans with 60 hz motors is the low pressure head, high performance one-and-two stage axial-flow fans were developed recently with increased pressure producing capabilities. Axial-flow fan design becomes difficult aerodynamically if high pressure rise is required at low flow rate. High pressure corresponds to high blade rotational speed (see laws of performance) and, since the motor speed is limited by the line frequency, the blade rotational speed can be increased only by increasing blade diameter. When low flow rates are required, the radial blade length decreases very rapidly with the increase in blade diameter (see Figure 37).

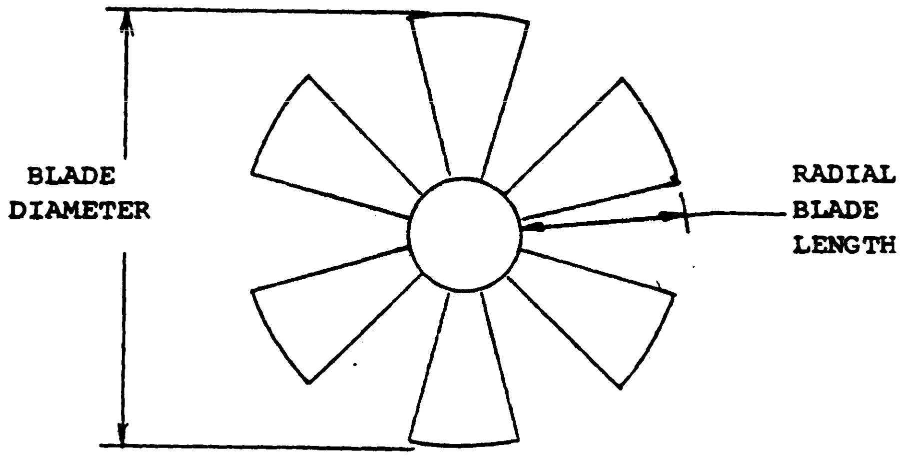


FIGURE 35. Typical Fan Rotor with Radial Blades

A point can be reached where the increased losses (tip clearance, wall boundary layer) due to short blade lengths counteract the theoretical pressure increase due to the higher blade speed at the larger diameters.

9.2.2 Laws of fan performance.

9.2.2.1 Theory. Before the laws of fan performance can be presented, the quantities involved will be defined:

The static pressure (p_s) of a fluid is the compressive pressure existing in the fluid, and is the measure of the potential energy of the fluid.

The velocity pressure (p_v) of a fluid is the pressure corresponding to the average velocity determined from the volume of air flow and the cross-sectional area. Disregarding the units,

$$p_v = \frac{1}{2} \frac{\rho}{g} V^2 \qquad (9\text{-}37)$$

or, the velocity pressure is the measure of the kinetic energy of the fluid.

The total pressure (p_t) of the fluid is the sum of the velocity pressure and static pressure and it is a measure of the total energy of the fluid.

The volumetric flow rate, v, of a fan must be calculated at outlet conditions, i.e., temperature, pressure, area, and velocity. The average velocity of the air at any point in the duct is determined from the velocity pressure readines, according to the following formula:

$$V = 1096.2 \sqrt{\frac{P''_v}{\rho}} \tag{9-38}$$

where:

ρ = the air density in lbs/cu.ft.
P''_v = the velocity pressure in inches of water
V = the average velocity of air in ft./min.

The air power (P_{air})* is the fan power output, and it is the power supplied to air by the fan as mechanical work.

9.2.2.2 Air horsepower for centrifugal fans. When the motor is mounted external to the ductwork, the power supplied to air is equal to air horsepower and may be expressed by the relationship:

$$P_{air} = P_{in} \times e_M \times e_{BL} \tag{9-39}$$

where:

P_{in} = the driving motor electrical power input
P_{air} = the power dissipated in air
e_M, and e_{BL} are motor and blower efficiencies, respectively
from the energy equation:

$$P \propto Hm \tag{9-40}$$

where:

H = the head, ft. of fluid
m = the weight rate of flow $\frac{\text{lbs.}}{\text{min.}}$ (also known as mass flow rate)

Bernoulli's equation gives for the total pressure p_t, of a fluid:

$$p_t = \frac{\rho V^2}{2g} + p_s + \rho z \tag{9-41}$$

where:
p_s = the static pressure (See Chapter 15)
z = the vertical height of the point at which pressure is given (with respect to some reference plane)

* Air Horsepower in watts

Since head is pressure divided by density, the difference in head between outlet and inlet is:

$$H = \frac{p_{s_o}}{\rho_o} - \frac{p_{s_i}}{\rho_i} + \frac{V_o^2 - V_i^2}{2g} + z_o - z_i \qquad (9\text{-}42)$$

where:

Subscripts o and i indicate outlet and inlet, respectively.

For fans and blowers z_o and z_i are practically equal. If the inlet and outlet ducts are of the same size and there is little compression; that is, if

$$\frac{\rho_i - \rho_o}{\rho_i} < 7\%$$

then

$$H = \frac{p_{s_o} - p_{s_i}}{\rho} \qquad (9\text{-}43)$$

The air power is then:

$$P_{air} = \left(\frac{p_{s_o} - p_{s_i}}{\rho_o}\right) m = \left(p_{s_o} - p_{s_i}\right) Q_o$$

When the dimensional units are considered:

$$P_{air} = 0.116 \left(\frac{p_{s_o} - p_{s_i}}{\rho_o}\right) m = 0.116 \left(p_{s_o} - p_{s_i}\right) Q_o$$

or

$$P_{air} = 0.116 \, \Delta p_s \, Q_o \qquad \begin{matrix}(9\text{-}44)\\(D.E.)\end{matrix}$$

where:

P_{air} is in watts

$\left(p_{s_o} - p_{s_i}\right) = \Delta p_s$ is static pressure rise in the fan.

Q_o is the outlet volumetric flow rate $= \frac{m}{\rho}$

9.2.2.3 Air horsepower for centrifugal fans with motor overblow, and for axial-fans. When the fan motors are mounted axially with the fan or when motor overblow ducts are used (as in the case of cooling motors for centrifugal fans) the total power supplied to the motor may be considered to be dissipated in the air. If the density change of the air entering and leaving the motor-blower system is less than 7%, then:

$$P_{air} = P_{input} \tag{9-45 (D.E.)}$$

Since some of the heat developed by the motor will be lost by conduction through the motor base and by radiation and free convection from the casing, Equation 9-45 will give conservative results.

9.2.2.4 Air temperature rise due to P_{air}. The power dissipated in air (P_{air}) must be added to the power dissipated by electronic parts when designing the heat exchangers in closed systems. Since, in the temperature and pressure ranges generally used in forced-air cooling, air may be assumed to be a perfect gas, the power dissipated in air can be expressed in terms of the weight rate of flow and air temperature rise as:

$$P_{air} = q = 7.62\ m\ \Delta t_b \tag{9-46 (D.E.)}$$

where:

P_{air} = the power dissipated in air, watts
q = the same as P_{air}
m = the weight rate of flow of air, $\frac{\text{lbs.}}{\text{min.}}$
Δt_b = the temperature rise of air, °C

$$(\Delta t_b = t_{b_o} - t_{b_i})$$

From Equation 9-46, the increase in air temperature can be calculated as:

$$t_{b_o} = t_{b_i} + \frac{P_{air}}{7.62m} \tag{9-47 (D.E.)}$$

It should be noted, however, that only the P_{air} as calculated by Equation 9-44 is the useful energy.

9.2.2.5 Similarity relationships. It is exceedingly difficult to develop accurately, general analytical relationships for fans. There are, however, approximate dynamic similarity relations which help to indicate the relative performance of fans. These relations will aid in selecting fans for any particular application and will help to evaluate the adequacy of a given fan for a new application, when the performance of the fan is known.

The proportionalities of the dynamic similarity follow:

$$\Delta p \propto \rho \frac{D^2 N^2}{g} \tag{9-48}$$

$$Q \propto ND^3 \tag{9-49}$$

$$HP \propto Q \, \Delta P \propto \rho \frac{D^5 N^3}{g} \tag{9-50}$$

$$m \propto \rho Q \tag{9-51}$$

where:

N = the fan rotational speed
D = a radial dimension, usually taken as fan blade or impeller diameter.
ρ = density of fluid at inlet condition

When any of the three quantities defined above is constant, the constants naturally, do not affect the proportionality.

For cooling of electronic equipment at sea level conditions, the density of air can be assumed to be constant.

When, for a centrifugal fan, D, N, and ρ are constant, then:

$$Q \propto B$$

$$HP \propto B$$

$$m \propto Q \propto B$$

ΔP is constant

where:

B = the impeller width

m = the weight rate of flow

9.2.2.6 Fan characteristics. Figure 36 shows typical performance curves of a centrifugal fan running at constant speed. While there are various types of both centrifugal and axial-flow fans which have different characteristic curves, Figure 36 is typical. With the outlet blocked there is no air flow and usually maximum static pressure. At maximum air flow or free delivery, the static pressure developed is zero. There is some point between these two extremes where efficiency is a maximum. Fans should operate at or near this point when power considerations are important. This requires that optimum fan type, particularly for size and speed, be selected to deliver the required air flow and static pressure. In other words, the fan must be well matched to the system requirements. Due to changes in air density with altitude, fan performance will change with altitude in airborne applications. Special fans are available with constant mass flow characteristics, independent of altitude.

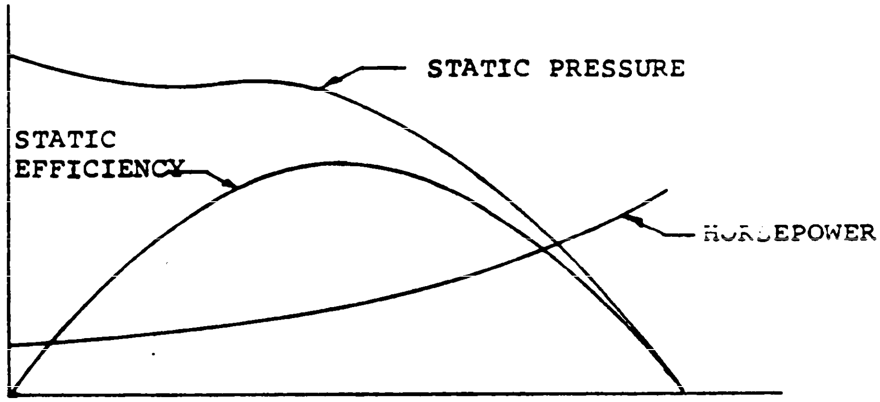


FIGURE 36. Characteristic Curve of a Centrifugal Fan at Constant Speed

9.2.2.7 Interrelation between system resistance and fan performance. In order to show the relation between the duct system resistance characteristics and the fan performance, Figure 39 is presented:

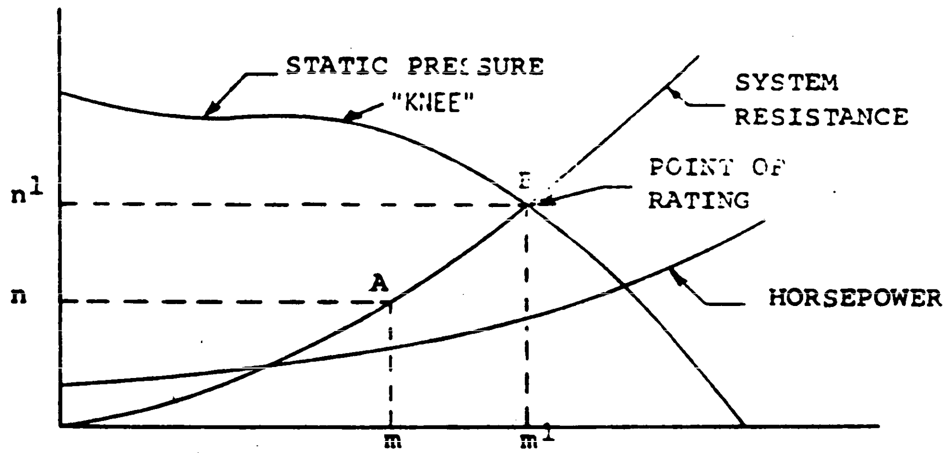


FIGURE 37. Fan and Duct System Performance

First, consider the "system" resistance curve. This is the static pressure-air flow characteristic of the forced-air cooling system. For example, point "A" shows that, at m cfm air flow through the system, the required static pressure is n inches of water at a given air density. The pressure drop varies as the square of the air flow rate so that the system is a squared curve. The system rating point is that where the system resistance curve crosses the fan static pressure curve, or, in this case, point "B". Thus, the air flow will be m' cfm at n' static pressure. It can be seen that it is important to select the fan so that its static pressure curve passes through or near the system resistance curve at the desired air flow and pressure near peak efficiency (if efficiency is important). Further, the speed and noise level must also be considered. Referring to the example in Figure 37, if point "A" were desired but a fan was used having characteristics as shown, the air flow actually delivered would correspond to that at "B" and the horsepower would be necessarily high. The above situation can be remedied by judicious selection of fans and proper control of fan performance.

9.2.3 Selection of fans. Matching a fan to the resistance of a system is sometimes complicated by operation under temperature and altitude conditions where the density ρ of the coolant, usually air, is less than standard sea level density. On the other hand, some electronic equipment is pressurized to prevent high-voltage from arcing. In this case, coolant density is higher than standard density, and the pressurized coolant is circulated through a heat exchanger. Resistance and fan delivery data are most often obtained at standard conditions, where ρ = .075 pounds/ft.^{3} for air.

It is extremely important to understand that the designer must now begin to think in terms of coolant weight flow rather than volume flow. Whatever the environment may be, the temperature difference for a given heat dissipation between the surface of a hot body and the coolant, neglecting variations in radiation heat transfer, is constant at constant weight flow. With that in mind, thermal analysis at conditions other than standard need not cause any anxiety.

Since system resistance varies directly as the square of the coolant weight flow, W, it is convenient to use a log-log plot of weight flow vs. resistance to match a fan with a system. Thus,

$$\Delta P_{system} = CW^2 \qquad \underline{\text{at constant } \rho}$$

$$\log \Delta P = 2 \log W + \log C, \text{ where } C \text{ is a constant.} \qquad (9\text{-}52)$$

Plotted on log-log paper, this is a straight line with slope = 2.

W is usually measured in pounds per minute, ΔP in inches of water. Note that the system resistance also varies inversely with density,

$$\Delta P_{system} = C_2/\rho \qquad \underline{\text{at constant } W}$$

where C_2 is a constant,

while the ability of the fan to overcome resistance varies directly with density at constant volume.

Thus,

$$\text{system } \Delta P_2 = \Delta P_1 \left(\rho_1/\rho_2\right) \quad \begin{array}{l}\text{at constant W} \\ (\Delta P = \text{system pressure drop})\end{array} \quad \begin{array}{l}(9\text{-}53) \\ (\text{D.E.})\end{array}$$

$$\text{fan } \Delta P_2 = \Delta P_1 \left(\rho_2/\rho_1\right) \quad \begin{array}{l}\text{at constant volume} \\ (\Delta P = \text{fan pressure rise})\end{array} \quad \begin{array}{l}(9\text{-}54) \\ (\text{D.E.})\end{array}$$

Figure 38 demonstrates the use of the log-log plot where coolant density during worst case equipment operation is less than standard density. One resistance point 1, determined by test or calculation, is plotted. A straight line with slope = 2 is drawn through point 1. This line describes system resistance vs. coolant flow at sea level, or standard conditions. From equation 9-53 find

$$\Delta P_2 = \Delta P_1 \left(\rho_1/\rho_2\right) = 3\Delta P_1 \text{ (equipment)}$$

and draw in the altitude resistance line with slope = 2 through point 2.

Convert the manufacturer's pressure rise (ΔP) vs. volume flow curve for the fan under consideration to pressure rise (ΔP) vs. weight flow. Convert this curve to a log-log curve and plot it on your log-log paper using as many points as necessary to get a clear picture. Convert the fan curve from sea level to altitude using equation 9-54. Plot it on the log-log sheet.

Now enter the required coolant weight flow, determined by calculation or experiment, as a vertical line. Actual weight flow under altitude operating conditions is shown at the intersection, A, of the two altitude lines. Actual weight flow must be equal to or greater than required weight flow, and it should occur at or beyond the "knee" of the fan curve. The sample case of Figure 38 shows that the fan's weight flow delivery at altitude meets all requirements.

Although the required weight flow may be greater at sea level because of a higher ambient temperature, the fan will still probably have excess sea level capacity.*

9.2.4 Acoustical noise. Noise level is an important factor in the selection of a fan for cooling electronic equipment. In general, a fan operating with high blade-tip velocity and developing a static pressure of more than 2" of water produces noticeable amounts of noise in quiet surroundings. Static and dynamic balancing, vibration isolation, and streamlined flow design provide smoother performance with fewer vibrations, which also tend to reduce noise. Where low noise output is important, the manufacturer should be consulted. With speed fans, when the noise level is known

*For a more detailed treatment of fans, see Fan Engineering, Buffalo Forge Co.

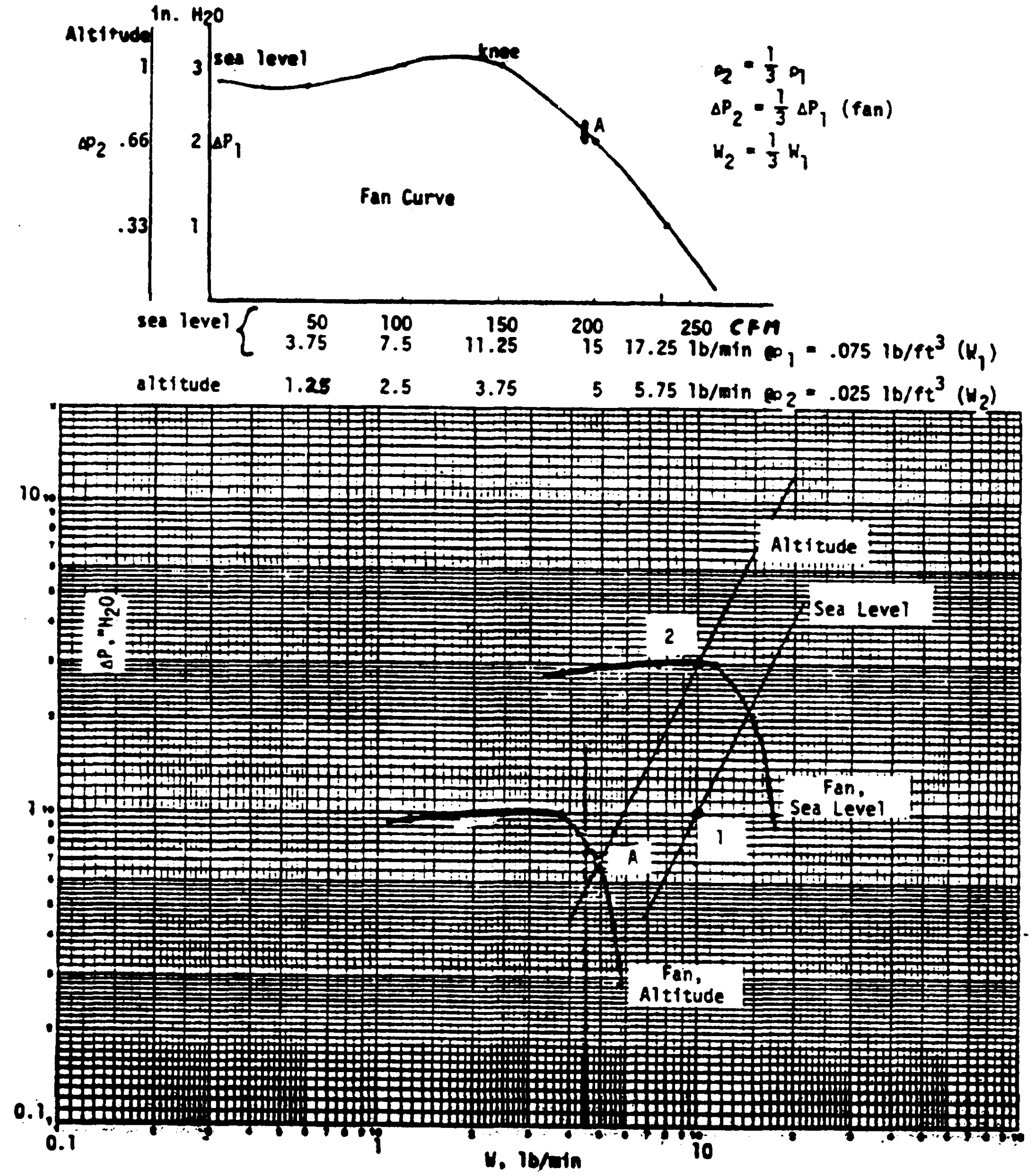


FIGURE 38. Fan/System Matching

for a particular fan speed, corrections for variations of fan speed not exceeding ± 10% may be made according to the following formula (for flat response observations):

$$\text{Db change} = 50 \log_{10} \frac{\text{RPM}_2}{\text{RPM}_1} \qquad (9\text{-}55)$$

Note:
MIL-STD-740B (SHIPS) - "Airborne and Structureborne Noise Measurements and Acceptance Criteria" is applicable to shipboard electronics.

9.3 Pressure losses and their determination.

9.3.1 Pressure drop. Fluid flow through a duct or conduit is caused by a difference in energy along the conduit, the flow being from the higher energy level to the lower energy level. In steady flow, this difference in energy levels exists because there is present a resistance to the flow. This resistance can be divided into two types:

1. Viscosity, the resistance offered by a fluid to the relative motion of its parts, and
2. Friction to the flow due to contact with the inner surface of the conduit.

Head loss or pressure drop is a measure of the difference in energy. In the absolute sense, friction results in the conversion of flow energy to some other form that is of no use in sustaining the flow, i.e., heat energy.

An electrical analogy for head loss is the voltage drop across a length of wire. Conduit friction is analogous to the ohmic resistance of wires. Thus, according to Ohm's Law, head loss and voltage drop are directly proportional to the flow rate and current respectively.

The head loss or pressure drop around a given cooling system must be known in order to select the proper fluid moving device, i.e., blower. The blower has to provide sufficient pressure head to result in the desired flow rate after all losses have been overcome.

9.3.1.1 Head loss. Flow through a conduit may be either laminar or turbulent. It has been shown by experiments that, in laminar flow, head loss varies directly with the viscosity of the fluid and its velocity. However, for turbulent flow, head loss varies directly with the density of the fluid and the square of its velocity.

Generally, for most fluids and expecially gases, laminar flow takes place at extremely low velocities; hence, all commonly used equations apply to turbulent flow.

The equation most used in calculations of head loss is the Darcy-Weisbach Equation: (Ref. 47)

$$H_L = f \frac{L}{D} \frac{V^2}{2g} \qquad \begin{matrix}(9\text{-}56)\\(\text{D.E.})\end{matrix}$$

where:

H_L = the head loss - ft. of fluid
L = the length of conduit - ft.
D = the diameter of conduit - ft.
V = the velocity of the flow - ft. per second
g = 32.2 ft. per second per second - $\frac{ft.}{sec.^2}$
f = the dimensionless parameter known as the friction factor

where:

$$f = \emptyset \frac{\epsilon}{D}, Re \qquad (9\text{-}57)$$

where:

$\emptyset$ = "some function of"
ϵ = the absolute roughness of the surface (ft.)
Re = the Reynolds Number (dimensionless)

Note: There are in existence two systems of noting friction factor. One of these is four times greater than the other. However, use of the larger f is becoming predominant, and it is this factor that is used in this handbook.

In generalized flow, the friction factor has been shown to be a function of the relative roughness, $\frac{e}{D}$ and the Reynolds Number, Re. Relative roughness is the ratio of the absolute roughness of the conduit surface to the diameter of the conduit. The absolute roughness refers to the size of the small bumps and ridges which are inevitably present in commercial conduits. Excellent correlation between experiments using artificially roughened conduits and experiments using natural conduit surfaces has been obtained. (Nikuradse's tests, (Reference 47) etc.) Reynolds Number is discussed elsewhere in this handbook.

The dependency of the friction factor on the two quantities stated makes it possible to show the relationship between friction factor and Reynolds Number as a two dimensional plot, with a family of curves for each of which $\frac{e}{D}$ is constant. This plot (Figure 39) is known as the Moody Diagram. Its use will be explained in the sample problems.

For perfectly smooth conduit, the friction factor is a function of the Reynolds Number only. The following equations have been empirically validated.

(a) Hagen-Poiseulle Equation

$$f = \frac{64}{Re} \qquad (9\text{-}58)$$

(b) Blasius Equation

$$f = \frac{0.316}{(Re)^{0.25}} \qquad (9\text{-}59)$$

(c) Prandtl Equation

$$f = \frac{0.184}{(Re)^{0.2}} \qquad (9\text{-}60)$$

valid up to Re = 3,000,000

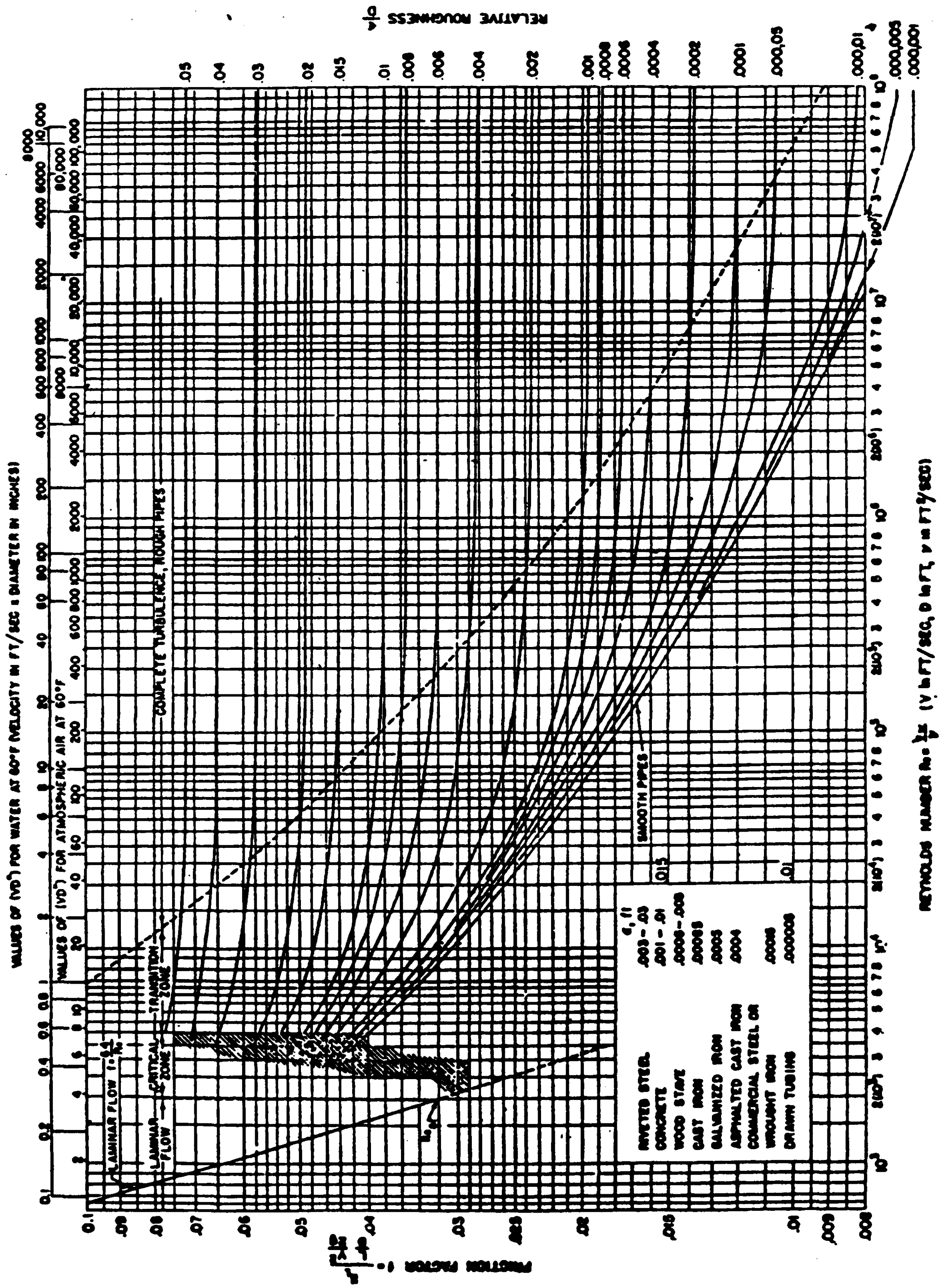


FIGURE 39. Moody Diagram

TABLE XIII. Absolute Roughness Classification of Conduit Surfaces for Selection of Friction Factor, f, in the Moody Diagram*

Commercial Surfaces (new)	Absolute Roughness-- feet
Brass, Glass, Tubing (drawn)**	0.000005
Wrought Iron, Steel	0.00015
Galvanized Iron, Steel	0.0005
Cast Iron	0.00085
Concrete, Average	0.004
Riveted, Light, Steel	0.003
Riveted, Heavy, Steel, Brick	0.03

* The values given in Table XIII are for clean conduits. Rusted or carbunculated conduit may have several times the roughness of clean conduit.

** Under brass and tubing is included all drawn material such as lead, block tin, aluminum, copper, steel, and glass. Metal tubing 1/4" O.D. or less is often finished "no plug" draw, in which case it will not be smooth inside, but wrinkled, and must be considered as rough as cast iron.

9.3.1.2 Sample Problem.

Note: This is not a practical problem, but is included to demonstrate the operations and principles involved in this type problem. (For illustrating use of Darcy-Weisbach Equation and Moody Diagram)

Problem - 1000 cfm of atmospheric air at 15°C flows through 6" diameter galvanized duct 70 feet long. What is the head loss in inches of water?

Solution:

a. Area of duct $= \frac{\pi D^2}{4} = \frac{\pi}{4} \times \left(\frac{1}{2}\right)^2 \text{ft.}^2 = \frac{\pi}{16} \text{ ft.}^2$

b. Velocity $= \frac{\text{flow rate}}{\text{area}}$, $V = \frac{Q}{A_c}$

$$V = 1000 \frac{\text{ft.}^3}{\text{min.}} \times \frac{\text{min.}}{60 \text{ sec.}} \times \frac{16}{\pi} \frac{1}{\text{ft.}^2}$$

$$= \frac{1000 \times 16}{60}$$

$$= 85. \frac{\text{ft.}}{\text{sec.}}$$

c. $\mu = 3.8 \times 10^{-7} \frac{\text{lb.-sec.}}{\text{ft.}^2}$ (from tables in Appendix)

Note: This is based on the dimensional system where one pound-force is the basic unit. To convert to the pound-mass system, μ must be multiplied by g.

Hence:

$$\mu = 3.8 \times 10^{-7} \frac{\text{lb.-sec.}}{\text{ft.}^2} \times 32.2 \frac{\text{ft.}}{\text{sec.}^2}$$

$$= 1.22 \times 10^{-5} \frac{\text{lb.}}{\text{ft.-sec.}}$$

d. ρ can be found from Equation 9-8

$$\rho = 0.0807 \left[\frac{273}{T+273}\right]\left[\frac{P}{14.7}\right]$$

$$= 0.0807 \left[\frac{273}{15+273}\right]\left[\frac{14.7}{14.7}\right]$$

$$= 0.0765 \frac{\text{lbs.}}{\text{ft.}^3}$$

e. Reynolds Number

$$Re = \frac{VD\rho}{\mu}$$

$$= 85 \frac{\text{ft.}}{\text{sec.}} \times 0.5 \text{ ft.} \times 0.0765 \frac{\text{lbs.}}{\text{ft.}^3} \times \frac{1}{1.22 \times 10^{-5}} \frac{\text{ft.-sec.}}{\text{lb.}}$$

$$= \frac{85 \times 0.5 \times 0.0765}{1.22} \times 10^5$$

$$= 2.66 \times 10^5$$

f. ϵ for galvanized surface = 0.0005 ft. (from table XIII)

$$\frac{\epsilon}{D} = \frac{0.0005 \text{ ft.}}{0.5 \text{ ft.}} = 0.001$$

g. From Moody Diagram

$Re = 2.66 \times 10^5$, $\frac{\epsilon}{D} = 0.001$

$f = 0.021$

h. $H_L = F \frac{L}{D} \frac{V^2}{2g}$

$$= 0.021 \times \frac{70 \text{ ft.}}{0.5 \text{ ft.}} \times (85)^2 \frac{\text{ft.}^2}{\text{sec.}^2} \times \frac{\text{sec.}^2}{2\text{x}32.2 \text{ ft.}}$$

$$= \frac{0.021 \times 70 \times 7200}{0.5 \times 64.4} \text{ ft. of air} \tag{9-61}$$

$$= 329.0 \text{ ft. of air}$$

i. $H_L = 329 \text{ ft. of air} \times 0.0765 \frac{\text{lbs.}}{\text{ft.}^3} \text{ of air} \times \frac{\text{ft.}^3}{62.4 \text{ lbs.}} \text{ of water}$

$$= \frac{329 \times 0.0765}{62.4} \text{ ft. of water}$$

$$= 0.403 \text{ ft. of water}$$

j. $H_L = 0.403 \text{ ft. of water} \times \frac{12 \text{ in.}}{\text{ft.}}$

$= 4.85$ inches of water ANSWER

¶. Note that since the fluid is air at 15°C, the VD" scale at the top of the Moody Diagram can be utilized. Instead of calculating the Reynolds Number, the VD" product is used.

$$VD" = 85 \frac{\text{ft.}}{\text{sec.}} \times 6" = 510$$

$$\frac{\epsilon}{D} = 0.001$$

then f = 0.021

Note that this is the same f as that previously determined. This method provides somewhat of a shortcut if the air is at standard atmospheric pressure (14.7 PSI) and 15°C (60°F).

9.3.2 Types of additional losses.

9.3.2.1 Losses. The head loss due to curves, elbows, meters, and valves may be expressed in either of two ways: (1) as a constant times the velocity head, or (2) as being equal to the loss in a certain additional length of straight conduit. These values are known as loss coefficients and equivalent lengths, respectively.

The loss coefficient is a constant for a given type fitting and is identified by the symbol K.

$$H_L = K \frac{V^2}{2g} \tag{9-62)(D.E.}$$

where:

H_L = the head loss, in feet of fluid flowing

K = the loss coefficient, or the sum of several loss coefficients

$\frac{V^2}{2g}$ = the velocity head, feet of fluid flowing

TABLE XIV. Loss Coefficients for Some Common Fittings* (From Ref. 44)

Fitting	K
Close Return Bend (180°bend)	2.2
Standard Tee	1.8
Standard Elbow	0.9
Medium Sweep Elbow	0.75
Long Sweep Elbow	0.60
45°Elbow	0.42

*The loss coefficients presented in this table are the resultant average of several experiments conducted by various sources. Truly reliable loss coefficients have not been fully established.

Through use of a simple relationship, the loss coefficient of a fitting can be converted to its equivalent length, i.e., the additional length of straight conduit whose head loss is the same as that of the fitting:

$$L_e = \frac{KD}{f} \qquad (9\text{-}63)(D.E.)$$

Then the head loss equation becomes:

$$H_L = f \frac{L_e}{D} \frac{V^2}{2g} \qquad (9\text{-}64)(D.E.)$$

where:

L_e = the equivalent length, feet
K = the loss coefficient
D = the diameter of conduit, feet
f = the friction factor

Generally, "minor" losses may be neglected in those situations where they comprise less than 5% of the total head losses due to friction.* The friction factor, at best, is subject to approximately 5% error and it is impractical to compute values to more than two significant figures.

*In a multichassis equipment of rack-type construction having a central blower system, "minor" losses are likely to be significant and should not be neglected.

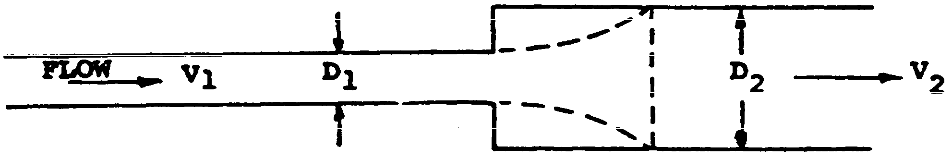


FIGURE 40. Sudden Expansion

9.3.2.2 Changes in duct size. In duct systems a sudden change in cross-sectional area is identified as either a sudden expansion or sudden contraction.

9.3.2.3 Sudden expansion. When the flow enters a large cross-section from a small cross-section, (see Figure 40) the following equation is applicable:

$$H_e = K \frac{V_1^2}{2g} \qquad (9\text{-}65)(\text{D.E.})$$

where:

H_e = the loss in feet of fluid due to expansion

$$K = \left[1 - \left(\frac{D_1}{D_2}\right)^2\right]^2$$

if section 2 is a reservoir then

$$\frac{D_1}{D_2} = \frac{D_1}{\infty} = 0$$

and K = 1.0

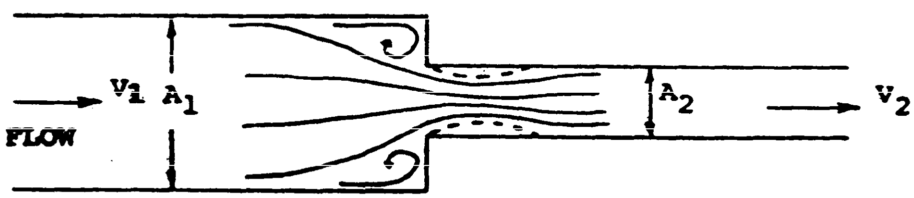


FIGURE 41. Sudden Contraction

9.3.2.4 Sudden contraction. With a sudden contraction, there is a contraction of the flow and a coefficient of contraction must be incorporated into the loss coefficient. The values presented are in close agreement with several references. (See Figure 41)

$$H_c = K \frac{V_2^2}{2g} \qquad (9\text{-}66)(\text{D.E.})$$

Where:

H_c = the loss in feet of fluid due to contraction

K_2 = the loss coefficient, from Table XV

$\frac{V_2}{2g}$ = the velocity head

TABLE XV. Loss Coefficient For Sudden Contractions
(From Ref. 45)

Area Ratio	$\frac{A_2}{A_1}$	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
Loss Coeff.	K	0.46	0.41	0.36	0.30	0.24	0.18	0.12	0.06	0.02	0

Note that the velocities used in the equations for head loss due to sudden changes of cross-section are the velocities which exist in the duct having the smaller diameter.

9.3.2.5 Gradual contraction. The loss in a gradual contraction, as shown in Figure 42, is generally computed as:

$$H_L = 0.05 \frac{V_2^2}{2g} \qquad (9\text{-}67)(\text{D.E.})$$

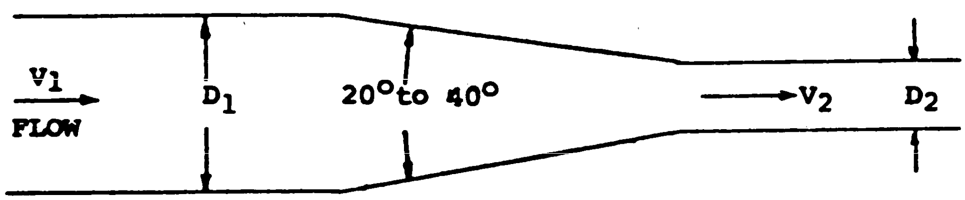


FIGURE 42. Gradual Contraction

The loss coefficient, i.e., K = 0.05, is considered constant over the range of angles shown.

9.3.2.6 Gradual expansion. The case of the gradual expansion is completely different from that of gradual contraction because the loss coefficient is directly dependent on the angle subtended by the sides of the expansion cone (see Figures 43 and 44).

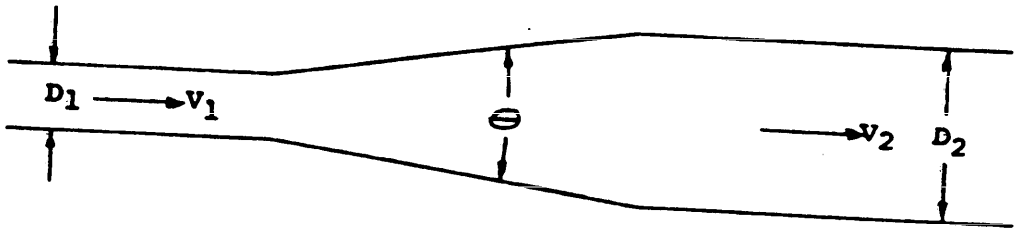


FIGURE 43. Gradual Conical Expansion

The equation is:

$$H_L = K\frac{(V_1 - V_2)^2}{2g} \qquad (9\text{-}68)(D.E.)$$

Where K is shown below

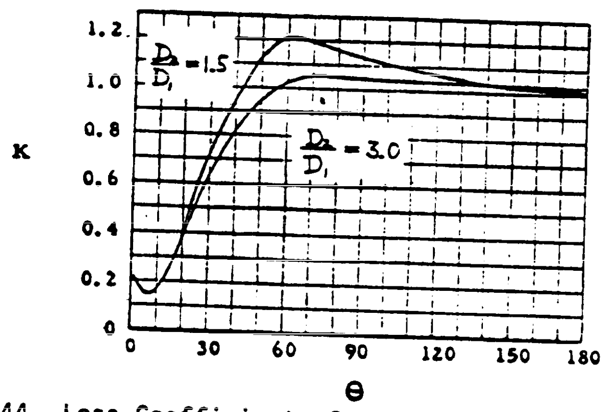


FIGURE 44. Loss Coefficients for Gradual Conical Expansions

9.3.2.7 Ducts in series. Two or more ducts of different sizes and roughnesses are in series, when connected so that the flow goes through one, then another, etc. For this situation it is necessary to determine the losses due to (1) the length of duct one, (2) the expansion or contraction, (3) the length of duct two, etc. A summation of the individual losses gives the total head loss.

9.3.2.8 Ducts in parallel. If two or more ducts are connected as shown in Figure 45 the system is classified as a parallel duct system.

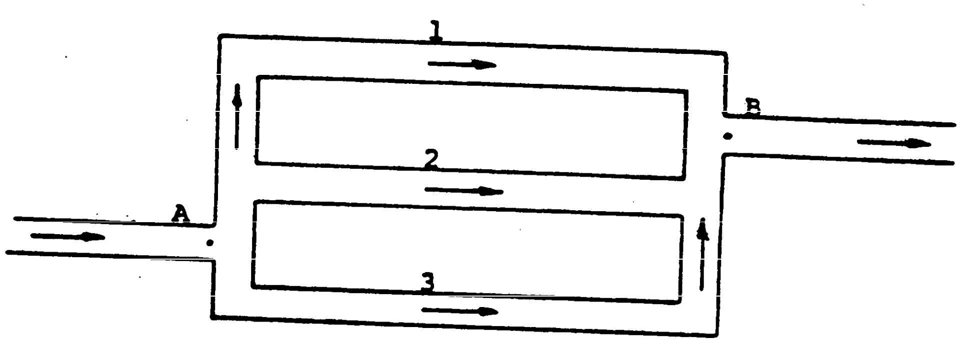


FIGURE 45. Ducts in Parallel

At equilibrium, i.e., the steady state, the following equations are valid:

$$Q_{total} = Q_1 + Q_2 + Q_3$$

$$H_{AB_1} = H_{AB_2} = H_{AB_3}$$

Or the total flow rate is the sum of the individual flow rates and the pressure drop or head loss is the same, regardless of the branch, between junctions.

9.3.3 Hydraulic radius (non-circular cross-section). In applications where it is necessary to calculate the flow through non-circular and annular shapes, a special parameter has been devised. This term is defined as the hydraulic radius.

$$HR = \frac{\text{Cross-sectional area}}{\text{Wetted parimeter}} \qquad (9\text{-}69)(\text{D.E.})$$

For a circular conduit flowing full, i.e., with only one fluid occupying the total cross-section, the hydraulic radius becomes:

$$HR = \frac{\text{Diameter}}{4}$$

If 4(HR) is substituted for D, the Reynolds Number becomes,

$$Re = V \frac{(4HR)}{\mu} \rho$$

and the head loss equation can be rewritten as

$$H_L = f \frac{L}{(4HR)} \frac{V^2}{2g}$$

Use of the hydraulic radius provides acceptably accurate results for square, oval, rectangular, triangular, and similar type ducts. Use of equation 9-69 for ordinary shapes gives results which are in fair agreement with experimental data if the flow is turbulent. Use of the hydraulic radius for laminar flow may give very inaccurate results. (Reference 47)

9.3.4 Compressible flow. When a gas flows in a duct, head loss may be calculated by using the "incompressible flow approximation," provided that the loss is less than 10% of the initial absolute static pressure. This procedure is based on the assumption that the density of the fluid remains constant. As a result, the velocity of the fluid is constant and pressure forces on the body of fluid in the duct will be used only in overcoming friction forces.

Generally, the initial density of the gas is used (i.e., conditions as of the upstream tap). Then the Darcy-Weisbach equation becomes:

$$H_L = 144 \frac{(P_1 - P_2)}{\rho} = f \frac{L}{D} \frac{V^2}{2g} \qquad (9\text{-}70)(D.E.)$$

Where:

P_1, P_2 = the initial and final pressures, psia
H_L = the head loss, feet of fluid flowing
ρ = the density of fluid, lbs./ft.^{3}
L = the length of duct, feet
D = the diameter of duct, feet
V = the the velocity of flow, ft./sec.
g = the acceleration due to gravity, 32.2 ft./sec.^{2}
f = the friction factor

If the pressure drop exceeds 10% of the upstream absolute static pressure, then true compressible flow equations must be used. These are beyond the scope of this handbook, but can be found in most fluid mechanics texts or mechanical engineering handbooks (see References 26, 45, & 47).

9.3.5 Basic types of problems. There are three basic types of problems that are encountered in head loss calculations. They are:

a. Given the flow rate, length, diameter, viscosity, and roughness, find the head loss. This is generally the most common type and is particularly applicable to electronic cooling systems where the flow rate is determined by the amount of cooling required.

b. Given head loss, length, diameter, viscosity, and roughness, find the flow rate. This usually would not be the case in an electronic cooling system, since the flow rate is determined by the amount of heat to be dissipated.

c. Given head loss, flow rate, length, viscosity, and roughness, find the diameter. This type of problem is probable inasmuch as a given blower might be available, and the duct size may depend on the permissible pressure drop (since pressure drop is inversely proportional to diameter).

9.3.5.1 Sample problem. Atmospheric air at 32°C is to flow at the rate of 300 cfm, through a straight run of galvanized circular duct, 130 feet long. Find the diameter of the duct if the permissible head loss is 0.35 inches water.

Solution:

The head loss equation states

$$H_L = f \frac{L}{D} \frac{V^2}{2g}$$

but V is not obtainable.

However, since

$$Q = VA = \frac{V\pi D^2}{4}$$

$$V = \frac{4Q}{\pi D^2}$$

and,

$$H_L = f\frac{L}{D}\left(\frac{4Q}{\pi D^2}\right)^2 \frac{1}{2g}$$

$$= f\frac{L}{D}\ \frac{16Q^2}{\pi^2 D^4 2g}$$

$$= \frac{8fLQ^2}{\pi^2\ D^5 g}$$

or,

$$D^5 = \frac{8LQ^2}{\pi^2 H_L g}\ (f) = C_1 f$$

Where C_1 is constant for any given configuration and,

$$C_1 = \frac{8LQ^2}{\pi^2 H_L g}$$

C_1 can now be calculated, but first change H_L = 0.35 inches of water to equivalent feet of air.

$$\rho \text{ air} = 0.0807 \left[\frac{273}{32 + 273}\right]\left[\frac{14.7}{14.7}\right]$$

$$= \frac{0.0807 \times 273}{305}$$

$$= 0.0723\ \frac{\text{lbs.}}{\text{ft.}^3}$$

$$H_L = 0.35\text{" water} \times \frac{\text{ft.}}{12\text{ in.}} \times \frac{62.4\text{ lbs.}}{\text{ft.}^3\text{ water}} \times \frac{\text{ft.}^3\text{ air}}{0.0723\text{ lbs.}}$$

$$= \frac{0.35 \times 62.4}{12 \times 0.0723}$$

= 25.2 feet air

Now calculate C_1

$$C_1 = \frac{8LQ^2}{\pi^2 H_L g}$$

$$= 8 \times 130 \text{ ft.} \times \left(\frac{5 \text{ ft.}^3}{\text{sec.}}\right)^2 \times \frac{1}{\pi^2} \times \frac{1}{25.2 \text{ ft. air}} \times \frac{\text{sec.}^2}{32.2 \text{ ft.}}$$

$$= \frac{8 \times 130 \times 25}{9.9 \times 25.2 \times 32.2}$$

$$= 3.24 \text{ ft.}^5$$

then

$$D^5 = 3.24 \text{ f}$$

$$Re = \frac{VD\rho}{\mu} \quad \text{but } V = \frac{4Q}{\pi D^2}$$

then

$$Re = \frac{D\rho}{\mu} \times \frac{4Q}{\pi D^2} = \frac{4\rho Q}{\mu\pi D} = \frac{C_2}{D} \qquad (9\text{-}71)$$

Where C_2 is a constant for a given configuration and

$$C_2 = \frac{4\rho Q}{\mu\pi}$$

Find C_2

$$C_2 = \frac{4\rho Q}{\mu\pi}$$

$$\mu = 4 \times 10^{-7} \frac{\text{lb.-sec.}}{\text{ft.}^2} \qquad \text{(from tables in appendix)}$$

then

$$C_2 = 4 \times 0.0723 \frac{\text{lb.}}{\text{ft.}^3} \times \frac{5 \text{ ft.}^3}{\text{sec.}} \times \frac{\text{ft.}^2}{4 \times 10^{-7} \text{lb.-sec.}} \times \frac{\text{sec.}^2}{32.2 \text{ ft.}} \times \frac{1}{\pi}$$

$$= \frac{4 \times 0.0723 \times 5 \times 10^7}{4 \times 32.2 \times \pi}$$

$$= 0.00357 \times 10^7 = 35{,}700.$$

then

$$Re = \frac{C_2}{D} = \frac{35700}{D}$$

When the above constants have been determined the following procedure is to be followed:

(1) Assume a value for f by inspection of the Moody Diagram.

(2) Solve equation 9-70 for D.

(3) Solve equation 9-71 for Re.

(4) Find the relative roughness $\frac{e}{D}$.

(5) Using Re and $\frac{\epsilon}{D}$, find new f from Moody Diagram.

(6) Using new f, repeat steps 1 - 5.

(7) When the value of f does not change, all equations are satisfied and problem is solved.

Now assume the value of f

$$\text{i.e., } f = 0.02$$

Solve for D

$$D^5 = C_1 f = 3.24f$$

$$= 3.24 \times 0.02 = 0.0648$$

$$D = \sqrt[5]{0.0648}$$

$$= 0.5785 \text{ ft.}$$

Now solve for Re

$$Re = \frac{C_2}{D} = \frac{35700}{0.5785} = 61{,}700$$

$$\frac{e}{D} = \frac{0.0005}{0.5785} = 0.00086$$

Then from Moody Diagram

$$Re = 61{,}700 \quad \frac{e}{D} = 0.00086$$

$$f = 0.023$$

Calculate new D

$$D^5 = C_1 f = 3.24 \times 0.023$$

$$D = \sqrt[5]{.07452}$$

$$= .595 \text{ ft.}$$

Calculate new Re

$$Re = \frac{35700}{D} = \frac{35700}{.595} = 60{,}000$$

and

$$\frac{\epsilon}{D} = \frac{0.0005}{0.595} = 0.00084$$

Then from Moody Diagram

$$Re = 60{,}000 \quad \frac{\epsilon}{D} = 0.00084$$

$$f = 0.023$$

Note that f does not change.

Note: It is generally the case that the solution for f converges very rapidly.

Then

$$D = 0.595 \text{ ft.} = 7.14 \text{ inches}$$

If this exact size is not available, the next larger size must be selected.

9.4 Forced air cooling design.

9.4.1 General considerations. Forced air cooling designs require careful assessment and consideration of the pertinent factors. Improper designs can lead to difficulties far worse than those which could be encountered with natural methods of heat transfer. Forced convection, at small flow rates, can provide thermal resistances of the order of half those obtained with free convection and radiation.

The design of forced air cooling systems for electronic equipment should be based mainly upon the attainment of safe operation temperatures at the heat sources and minimization of the energy required to move the air through the cooling system, i.e., cooling power. Two other factors of importance are the volume and weight of the cooling systems. Heat flow within and from heat producing electronic parts is discussed in detail in chapter 17 of this handbook. In general, most electronic parts, developed a decade or more ago, were designed for cooling by natural means and their external geometry is not ideally suited for forced air cooling. Only a few parts, such as certain transmitting tubes and finned rectifiers, are specifically designed for cooling by forced air. Other parts, especially medium and high power dissipation semi-conductor devices are designed for conduction into heat sinks or similar configurations with heat flow paths having low thermal resistances to the ultimate sink. Such heat sinks and cold plate chassis, etc., can be effectively cooled using forced air.

Parallel flow is generally recommended for cooling of large heat sources having high unit heat dissipations. Crossflow, though slightly less efficient, should generally be employed for small heat sources, since the duct work is simpler and less space is required. Under the same conditions, the surface temperatures will be slightly higher than for parallel flow, but if the power dissipated by the heat sources is low, the cooling air temperature rise will be small and the difference in cooling effectiveness for parallel and crossflow cooling will not be appreciable. In crossflow, a staggered arrangement increases turbulence, thus, lowering the thermal resistance and improving cooling. Crossflow-in-line arrangement is simpler and cheaper, but does not cool as effectively as a staggered arrangement.

In airborne applications, the effects of varying air density and pressure with altitude can be accommodated by using weight rate of air flow (also known as mass flow rate) in lbs./min. rather than volume flow rate in cfm. Most on board airborne equipment is currently cooled with relatively cool pressurized cabin air. Usually, the equipments are cooled with the air exhausting from the cabin. In other instances, jet engine bleed air or cooling air produced by "refrigeration or air conditioning equipment" is used. Section 9.8 of this chapter discusses this in detail. It is important to note that the air used to cool airborne equipment can become moisture laiden and should not pass directly over electronic parts and circuits. Indirect forced air cooling with cold plate type heat exchangers is recommended. The heat can be transferred from the heat sources to the cold plates by conduction or circulation of internal dry air within the equipment cabinet.

In the design of an equipment for direct forced air cooling, the following are to be considered:

(a) Determination of inlet cooling air pressure and temperature.

(b) Selection of part arrangements and spacings based on electronic and space requirements, as well as cooling power considerations.

(c) Determination of the maximum allowable temperature level of each part consistent with the reliability requirements.

(d) Determination of the required Reynolds Number.

(e) Calculation of the weight rates of flow over each part or group of parts based upon physical dimensions of the systems and the prescribed Reynolds Number.

(f) Determination of the overall pressure losses and required cooling power of the system.

9.4.2 Semiconductors.

9.4.2.1 Small transistors and diodes. Low power dissipation transistors and diodes can be directly forced air cooled usually without the addition of heat sinks or extended surface area devices such as fins. The external surfaces are sufficiently large to obtain the necessary thermal resistances which are usually in the °C/milliwatt range. The procedures outlined in section 9.1 of this chapter can be used to determine the required air flow and resultant thermal resistances.

Card mounted low power dissipation semiconductors are treated in section 9.4.7 of this chapter.

9.4.2.2 Power transistors and diodes. Semiconductor devices dissipating significant power do not have external surfaces sufficiently large for effective direct forced air cooling. Extended surface area devices, such as heat sinks, must be used in order to provide adequately large surface areas for heat transfer to the air. The heat dissipated in the semiconductors must be initially transferred by conduction into the sink. Thus, the conduction cooling techniques detailed in chapter 8 must be adhered to. Design techniques for flat plate and finned sinks and cold plates are presented in section 9.4.8 of this chapter.

High power rectifier diodes and SCR's are often supplied with integral finned heat sinks for forced air cooling. The manufacturers supply pertinent thermal data for the cooling of each configuration. The designer can select his operating points and determine the required air flow rate and pressure from the curves provided. It is important to note that the required air flow must be over and through the fins and not just in their general vicinity. This requires that the air flow be ducted to and over the finned sinks.

9.4.2.3 Integrated circuits. Integrated circuits in cylindrical metal cases dissipating low powers can be treated the same as small transistors. However, integrated circuits dissipating appreciable power, especially the dual-in-line types, generally require special finned heat sinks, unless adequate conduction is provided into the card on which they are mounted. In this instance the card surface becomes a flat plate heat sink. An assortment of small clip-on finned sinks for dual in line IC's are available from the various heat sink manufacturers. These sinks have staggered fingers (turbulators) and thermal resistance vs. air flow data are available from the manufacturers. The sinks should be mounted on the card integrally with the IC.

9.4.2.4 LSI devices. Large scale integrated circuits under development range in size and power from enhanced integrated circuits to single packages of the order of three to four inches in diameter mounted on a five inch square plate or substrate and dissipating 25 watts or more. The thermal design for forced air cooling of the lower dissipation LSI devices can be the same as for conventional integrated circuits. The larger LSI devices such as that mentioned above can be directly cooled with forced air provided at least one flat surface (preferably two) is exposed to relatively high velocity ducted air. Section 9.4.7, design example of printed circuit cards dissipating high power, presents techniques for the pertinent thermal design.

9.4.2.5 Microwave semiconductor conductor devices. The thermal resistances associated with these devices are critical as explained in paragraph 8.2.1.5 and in chapter 17. The cooling air flow requirements of the device manufacturers must be met and preferably exceeded, since usually the minimum requirements are listed.

9.4.2.6 Hybrid semiconductor devices. Large sized hybrid microelectronic devices containing up to 25 chips are finding increasing usage in modern avionic equipment. These higher dissipation hybrids may require thermally conductive adhesives between the package and P.C. boards and conductive thermal paths to P.C. board guide rails to maintain internal chip temperatures at values consistent with high reliability.

9.4.3 Electron tubes.

9.4.3.1 Conventional tubes. Conventional receiving and medium power glass transmitting tubes can be effectively cooled with forced air. Parallel flow of air over the base and upward over the envelope opposite the anode is ideal, especially if the air flow is ducted by a "chimney" type shield. Crossflow cooling is less effective than parallel flow, but can provide adequate cooling particularly when the tubes are staggered rather than in line. Section 9.1 presents design methods applicable to electron tubes. The tubes should be considered as cylinders having a diameter equivalent to the average diameter of the envelope.

9.4.3.2 Glass transmitting tubes. Many glass transmitting tubes are designed specifically for forced air cooling. Other glass transmitting tubes are designed for cooling by natural convection and radiation. These latter tubes, when forced air cooled usually provide improved equipment reliability since their temperatures are reduced and their rejected heat is not transferred by natural means into the other electronic parts in the equipment.

Parallel flow cooling with the air ducted over the base and upward over the envelope opposite the anode is recommended for glass transmitting tubes. Chimneys should always be used. Air flow data are provided by the tube manufacturers. These requirements are usually the minimum and should be exceeded by at least ten percent.

9.4.3.3 External anode tubes. Special high frequency and high powered transmitting tubes with finned external anode coolers intended for forced air cooling require special design considerations. External anodes are in general thermally desirable. Such anodes eliminate the largest resistance to heat heat transfer which normally exists in a tube, that is, the thermal resistance offered by the vacuum separating the anode from the glass bulb. Temperatures of external anodes will be considerably lower than those of the anodes in conventional tubes, thus, the internal control elements can be cooler or can operate at higher unit heat dissipations.

External anode tubes are generally more sensitive than conventional tubes to temperature gradients on their outer surfaces. This limitation is primarily due to the small dimensions of the glass or ceramic seals. The thermal, electrical, and mechanical stresses in the seals can be severe unless uniform cooling techniques are used. Tubes such as the 4X150A and 4X250A must be forced air cooled even when only the heaters are energized.

In general, the base seals are the most sensitive to temperature, with the plate-screen of plate-control grid seals running a close second. Seal temperatures on tubes such as the 4X150A are limited to 175°C maximum. However, it has been found that longer and more reliable service can be obtained when the seals are maintained in the vicinity of 100°C, with 125°C being the maximum reliable seal temperature. The limitation of the anode temperature is frequently the ceramic seal temperature or the melting point of the solder used to attach the fins and cooler to the anode.

The tube manufacturers' ratings for air flow over external anode tubes have been found to be somewhat marginal, especially at the seals. It is generally recommended that the nominal air flow ratings be exceeded by 20% for reliable operation. Further, some manufacturers rate tube air flow in CFM. This preferably should be a weight rating of flow rather than a volumetric rating. The manufacturer's pressure drop data for air flow over or through external anodes have been found to be reasonably accurate, the largest variations being at the special sockets for cooling these tubes by forced air.

Parallel flow should be used with tubes such as the 4X150A, whereas crossflow should be used with tubes such as the 2C39. With parallel flow it is recommended that the air first be directed to the base and the seals of the tube, thence flowing over the anode. Where possible, "chimneys" should be used at plate-screen seals to maximize turbulence at the seal surface. This flow direction will produce the lowest seal temperatures which can be obtained in a given situation. If the air is first passed over the anode, the seals are in effect cooled with "second hand" heated air. The air temperature rise through the anode will be high since most of the heat produced is dissipated at the anode.

Certain forced air cooled microwave tubes have external anodes which are inadequate in fin size and area for effective forced air cooling. This problem can be alleviated significantly through the application of clip-on turbulators attached to the anode fins. Section 9.4.6 discusses turbulators in detail.

9.4.4 Resistors.

9.4.4.1 Stacked resistor assemblies. Automatically produced assemblies of resistors usually need more than natural convection to provide adequate cooling. The printed wiring or plastic decks upon which the resistors are mounted form barriers against natural convection, and conduction through the plastic is essentially negligible.

Forced air cooling is the most economical means of attaining a reliable operating temperature for composition carbon and similar resistors under average operating conditions. For severe applications, conduction through metal clamps over the resistors offers the lowest thermal resistance.

9.4.4.2 Banks of resistors. It is often necessary or convenient to locate several resistors in close proximity to each other. When it is necessary to group heat generating elements such as resistors, the spacing between them should be as great as possible. If resistors are to be mounted on a vertical deck or chassis, as is commonly done to conserve space, the resistors' axes should be vertical. If the resistors are not of convenient length, then they may be mounted with their axes horizontal. However, with the axes horizontal, the resistor bodies should be staggered so that they are not directly above one another.

Typical data and comparison curves are presented herein which are based upon the following resistor arrangements and spacings:

elements in horizontal position; S/D = 2.0
elements in vertical position; S/D = 2.0
elements in vertical position; S/D - 1.5

where:
S is the distance between resistors
D is the resistor diameter

Generally, data from such configurations are correlated by means of equation 9-72.

$$\frac{hD}{k_f} = \emptyset \left(\frac{DV\rho}{\mu_f}\right) \tag{9-72}$$

This equation was found to be valid within experimental limits between Reynolds Numbers of 0.2 to 250,000.

Figure 46 shows data for single cylinders. The experimental points lie generally above the predicted curve, indicating better heat transfer in the bank than would be expected from single cylinders. This improvement in heat transfer may be due to improved turbulence obtained in a bank of staggered resistors.

Other configurations of resistors can be treated in the same manner as electron tubes (cylinders).

9.4.5 Reactors. Transformers and inductors used in electronic equipment are normally not designed for forced air cooling. However, special high power transformers, of the order of 10KVA or more, have been designed for "air blast cooling." These transformers have open cores and open windings with the conductors supported on widely separated spacers arranged so that the cooling air passes freely over the individual conductors. The conductors are self-supporting between the spacers; and thus, this technique is amenable only to transformers with large rigid conductors. The design techniques in section 9.1 can be used to determine the conductor spacings and the related forced air system design details.

The internal thermal resistances of transformers and reactors used in electronics are relatively large and the external surface areas are relatively small with respect to the dissipated power, (see chapter 17). Such transformers should be provided low thermal resistance conduction heat flow paths to finned or flat plate heat sinks, when forced air cooled. The applicable forced air cooling thermal design data are presented in this section.

9.4.6. Turbulators. On a local basis turbulent rather than laminar air flow is desirable because turbulence results in a much thinner boundary layer and reduced thermal resistance to the air. Designs of forced air cooled equipment should if possible assure turbulence near the surfaces of heat producing

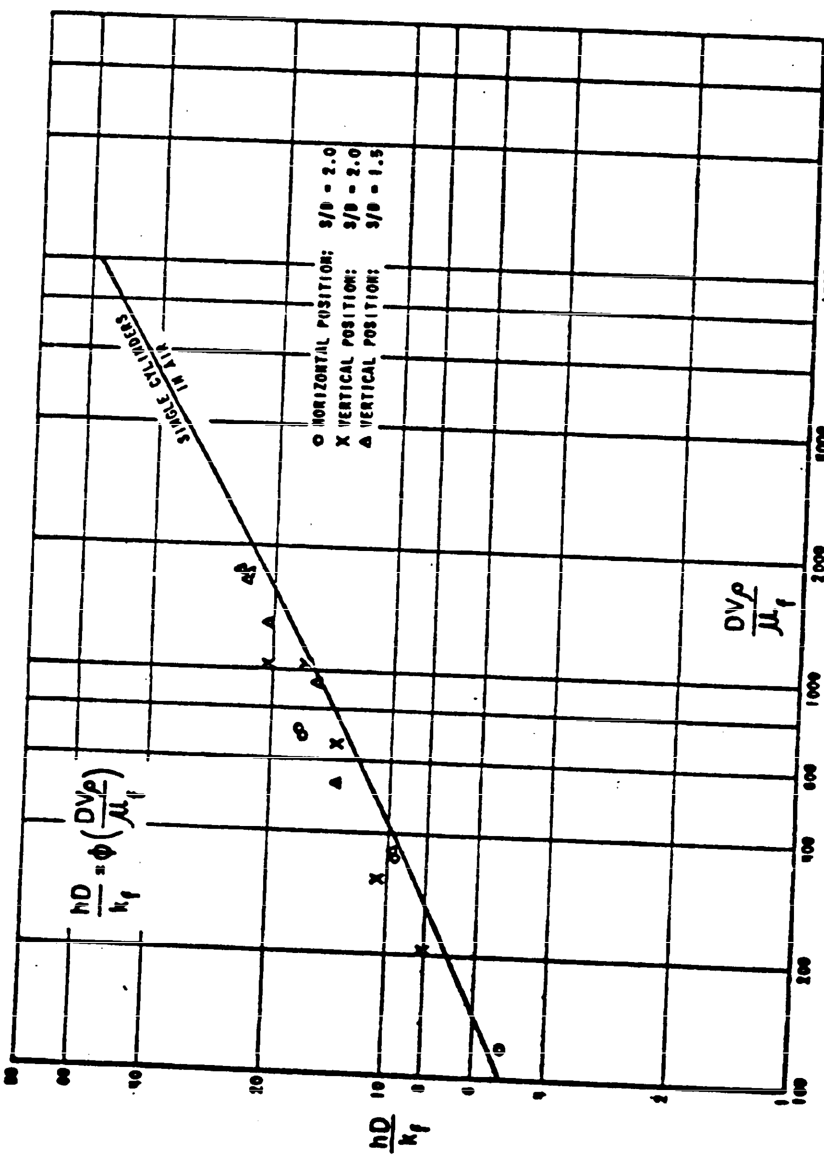


FIGURE 46. Correlation of Heat Transfer by Forced Convection From a Bank of Resistors

parts. This is generally accomplished by placing heat sources near each other or near duct walls so that the clearance spaces are small. A rough check on the design can be made by computing the Reynolds Number. It should be larger than 4000, although a Reynolds Number down to 2000 is acceptable. However, between 2000 and 4000, there exists a possibility of incomplete turbulence, possibly resulting in marginal design. Therefore, it may be better to use a Reynolds Number of 4000 as the minimum design quantity. The upper limit of Reynolds Number is dictated by pressure drop and blower power required.

Air flow patterns can be observed by injecting smoke at several points along the flow path. If, at any point, low turbulence exists, it will be evident by a concentration of smoke occurring at that point. Corrective steps, described below, may then be taken. If components of different size and shape are placed in a series in a duct, the turbulence will vary and may become undesirably low in some places. When such spots are found, the turbulence may be increased by introducing "turbulators" into the air stream. A sharp-edged plate set normal to the stream is an effective turbulator, the action of which is sketched in Figure 47. Both upstream and downstream turbulators are shown. It is evident that the turbulator throttles the flow and increases the required blower power by increasing the number of obstacles in the flow path.

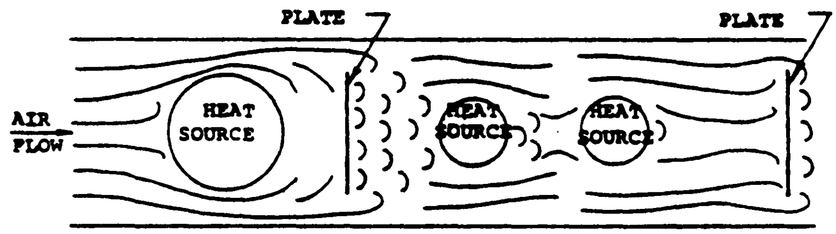


FIGURE 47. Sharp-Edged Plate Turbulator

Another useful device for increasing turbulence is a transverse row of small rods placed in front of a heat source as shown in Figure 48. The laminar flow is broken up, thus, increasing the heat-transfer coefficient.

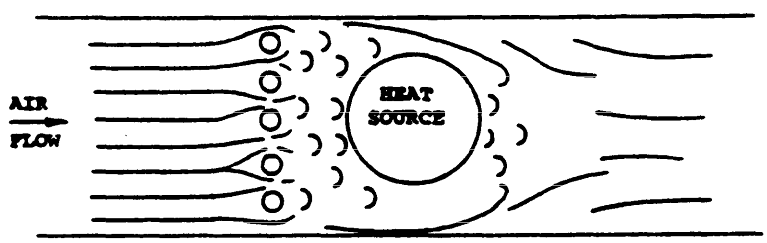


FIGURE 48. Rod Type Turbulator

No definite design rules are given for turbulators. If hot spots are discovered and a study of the air stream indicates low turbulence, various types and locations of turbulators should be tried until an effective arrangement has been found. Sketches of air flow similar to Figure 47 and 48 will frequently be helpful.

The nozzle arrangements described in the discussion of parallel flow cooling may also be thought of as turbulators. Nozzles and orifice plates may be found useful in eliminating local hot spots.

9.4.7 Flat heat sinks and P.C. cards. Circuitry employing single or multilayer boards with forced air cooling on one or both sides requires estimates of the thermal resistance between heat source surface and the heat transfer surface to the air. The printed layers of conductor a few thousandths of an inch thick together with the plated through holes and the solder form an extremely complex series parallel network which sometimes defies exact calculation.

For boards made of insulating material, the network can be treated as in Figure 49. The separate items are estimated from dimensions. It is reasonable to consider the printed conductor layer as covering 60 to 80% of the board surface. Interface mounting resistances of the parts comprise transistor mountings, heat sinked power resistors, and transformers chiefly. Reasonable values can be determined by the methods given in paragraph 9.3.

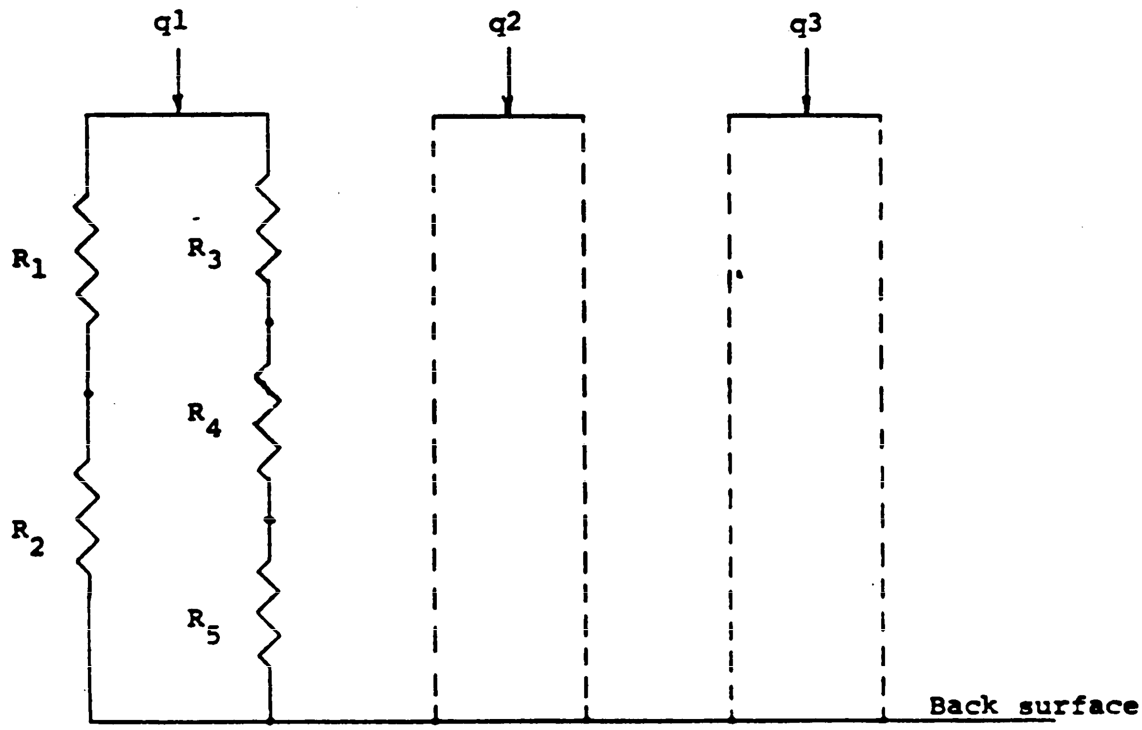
Cooling air is usually applied to the back of the board. When the mounting side is also cooled, the heat transfer surface area of that side should be taken as the actual surfaces of the parts, at the part case temperature.

Conduction through the leads and pin connectors to the structural supports, and radiation from the surfaces, are negligible for this type of construction.

A superior type of construction makes use of metal circuit boards. Each side is covered with a thin insulating layer on which conducting paths are printed, joined by plated through holes. The thermal circuit can be treated as in Figure 50. The conducting layer on each side tends to form a node and maintain a uniform surface temperature. Board structure comprising R_2, R_3, and R_4 has a very low thermal resistance. With a typical board carrying a considerable number of heat producing parts it is not necessary to solve the complete circuit.

Commercial producers of metal circuit boards can supply reliable thermal data for a large number of configurations. However, the designer must frequently make a thermal analysis of a new design. The following design examples illustrate practical procedures.

9.4.7.1 Design example P.C. card of insulating material. Ninety six flat pack dual transistors are mounted on a 2 x 4 inch card. Thermal resistance is 500°C/W junction to ambient air with natural cooling and 125°C/W junction to case. Each measures 0.25 x 0.15 inch. Each has 6 leads 0.012 inch diameter and 1/4 inch long. The board is 0.10 inch thick, with a thermal resistivity of 1200°C/watt-in., and is 70% covered by copper 0.005 inch thick. There are 6 plated through holes 0.02 inch diameter for each flat pack.



R_1 Leads

All parts in parallel

R_2 Plated through holes

R_3 Part surface contact

R_4 Insulating board material

R_5 Conductor layer, in series if more than one

q Part dissipation

FIGURE 49. Thermal Circuit for Insulating Card

The card is plugged in by 36 copper pins 0.04 inch diameter and 0.5 inch long.

Develop a thermal circuit, junction to environment, and a forced air cooling method. There are four principal temperature nodes:

T_j junction temperature
T_c case temperature
T_s card surface temperature
T_o sink temperature

The cards are to be spaced 1/4 inch apart with 40°C air moving at 500 ft./min. in the 4 inch direction. Forced air cooling is applied to only the rear side of the board.

Eight separate thermal resistances are identified:

R1 junction to case
R2 case to card contact
R3 card body
R4 holes with wire and solder
R5 printed copper layer
R6 leads from flatpack to card
R7 cooling of card surface
R8 plugs and wiring system. (assume no heat transfer through this path)

R1, R2, R4, and R6 are associated with a single flat pack. Since there are 96 equal thermal resistances in parallel in the thermal network, the overall equivalent resistance for the total card for these resistances is 1/96 the resistance for an individual transistor. R3, R7, and R8 are for the entire card.

R1 is given as 125°C/W/96 = 1.3°C/W

R2 is difficult to compute since the contact pressure and the surface factor are unknown. Manufacturers give these data for power transistors but not for this type of case.

For an approximate value, consider R2 a layer of air 0.005 inch thick.

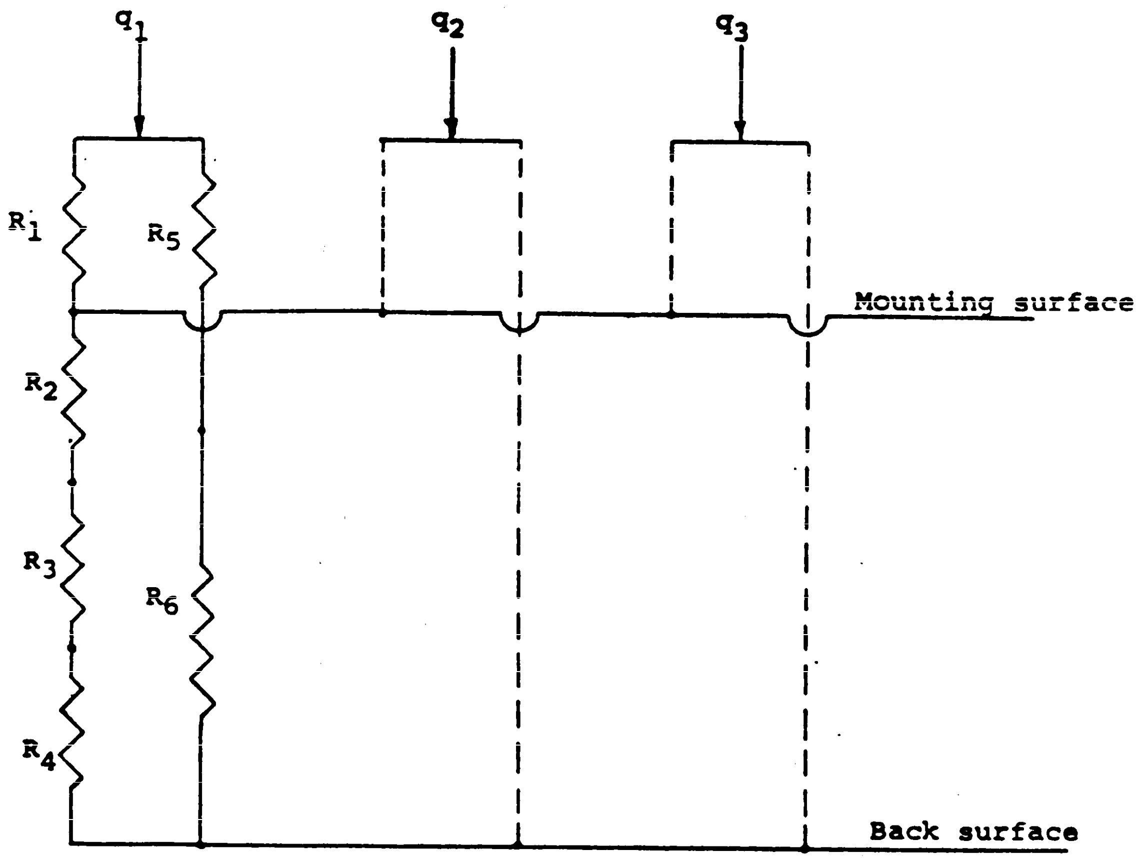
$$R2 = 0.005/(8 \times 10^{-4})\ (0.15)\ (0.25)/96$$
$$= 167°C/W/96 = 1.74°C/W$$

R3 is estimated by the formula

$$R = \frac{L}{KA}$$

where:

L = 0.1 inch
K = 1/1200 w-in./°C
A = average of case area and 1/96 of card area x 96



R_1 Part surface contact

R_2, R_4 Insulating layer

R_3 Metal board

R_5 Leads

R_6 Plated holes

FIGURE 50. Thermal Circuit for Metal Card

$$A = \frac{(0.25)\ (0.15) + (2)\ (4)/96}{2} \times 96 = 5.8$$

$$R3 = \frac{(0.1)\ (1200)}{5.8} = 20.7°C/W$$

For one hole considered as solid copper 0.02 in diameter

$$R = \frac{0.1}{(10)\ (314 \times 10^{-6})} = 32°C/W$$

Since there are 6 x 96 holes,

$$R4 = \frac{32}{(6)\ (96)} = 0.005°C/W$$

For the printed copper layer 0.005 inches thick, with A = (0.70) (2) (4) = 5.6 sq. in. and K = 10 w-in./°C,

$$R5 = \frac{0.005}{(10)\ (5.6)} = 8.9 \times 10^{5}\ °C/W \quad \text{(negligible)}$$

For one lead, R = 0.25/(10)A where $A\pi r^2 = \pi(.006)^2 = 0.000113$. Since there are 6 x 96 leads,

$$R6 = \frac{(0.25)}{(10)\ (.000113)\ (6)\ (96)} = 0.384\ °C/W$$

$$R7 = \frac{T_s - T_o}{hA(T_s - T_o)} = \frac{1}{hA} = \frac{1}{8h}$$

where h = w/sq. in.-°C depending on the forced air cooling rate.

With cards spaced 1/4 inch apart and 40°C air moving at 500 ft./min. in the 4 inch direction, h is computed by equation -77.

$$D_e = \frac{(4)\ (2)\ (0.25)}{(2)\ (2 + 0.25)} = 0.444 \text{ in.}$$

$$D_e/L = 0.111$$

$$Re = \frac{(41.1 \times 10^{-6})\ (500)\ (12/60)\ (0.444)}{1.07 \times 10^{-6}}$$

$$= 1705$$

$$h = \frac{6.87 \times 10^{-4}}{0.444} \left[3.65 + \frac{(0.0688)\ (0.111)\ (1705)\ (0.7)}{1 + 0.04\ (133)^{2/3}}\right]$$

$$= 0.00155 \quad 3.65 + \frac{(0.0668)\ (132.5)}{1 + (0.04)\ (26.06)}$$

$$= 0.0124 \text{ w/sq. in. -°C}$$

$$R7 = \frac{1}{(8)\ (.0124)} = 10.1\text{°C/W for 40°C air at 500 ft./min.}$$

Approximate circuit values, shown in Figure 51, are reduced to include all 96 flatpacks. By series parallel combinations the value of P from junction to sink reduces to 10.53°C/W.

9.4.8 Heat sinks. This section describes the thermal design of forced air cooled heat sinks including flatplate sinks, P.C. cards as single fin sinks, finned sinks, ducted finned sinks, and cold plates.

A wide variety of similar sinks are available commercially together with complete thermal data provided by the manufacturers. It is recommended that when such commercial sinks are used that the claimed thermal performance be experimentally evaluated to make sure that the performance is that anticipated.

9.4.8.1 Design example of metal PCB dissipating high power. An electronic equipment package comprising transformers high power transistors and PCB's mounted on four metal plates has a total steady state power dissipation of 500 watts. Stringent space and weight requirements result in a maximum unit heat concentration of 0.6 watts/cu. in. and unit heat dissipation as high as 1.4 watts/sq. in. All solid state parts are silicon and junction temperatures must be less than 150°C.

The cooling air supply is restricted to 6 lb./min./KW at 27 to 47°C and 2.0 inches of water pressure. No auxiliary fans are permitted. The package dimensions and geometry are fixed by non thermal considerations, and do not permit the inclusion of fins for extended heat transfer surfaces. Also, it is required to prevent direct contact between the cooling air and the electronic parts.

Design the forced air cooling system.

This package is to be cooled by high velocity air flowing through ducts formed by shrouding the P.C. boards, forming three ducts shown in Figure 52. The cooling air will pass over the back of the boards to avoid contact with electronic parts. All parts are thermally bonded to the plates and located so as to distribute the heat load as uniformly as possible consistent with circuit requirements. The plates are spaced to form ducts 1/8 inch wide to develop the required high velocity.

Since the power transistors dissipating 32 watts each are the most concentrated heat sources, a maximum allowable plate surface temperature was determined as follows:

The junction to case thermal resistance is 0.8°C/W. Estimating 0.6°C/W for interface resistance, at 32 watts dissipation, for a maximum junction temperature of 150°C, the maximum allowable plate surface temperature is 150°-(32)(1.4) = 105°C.

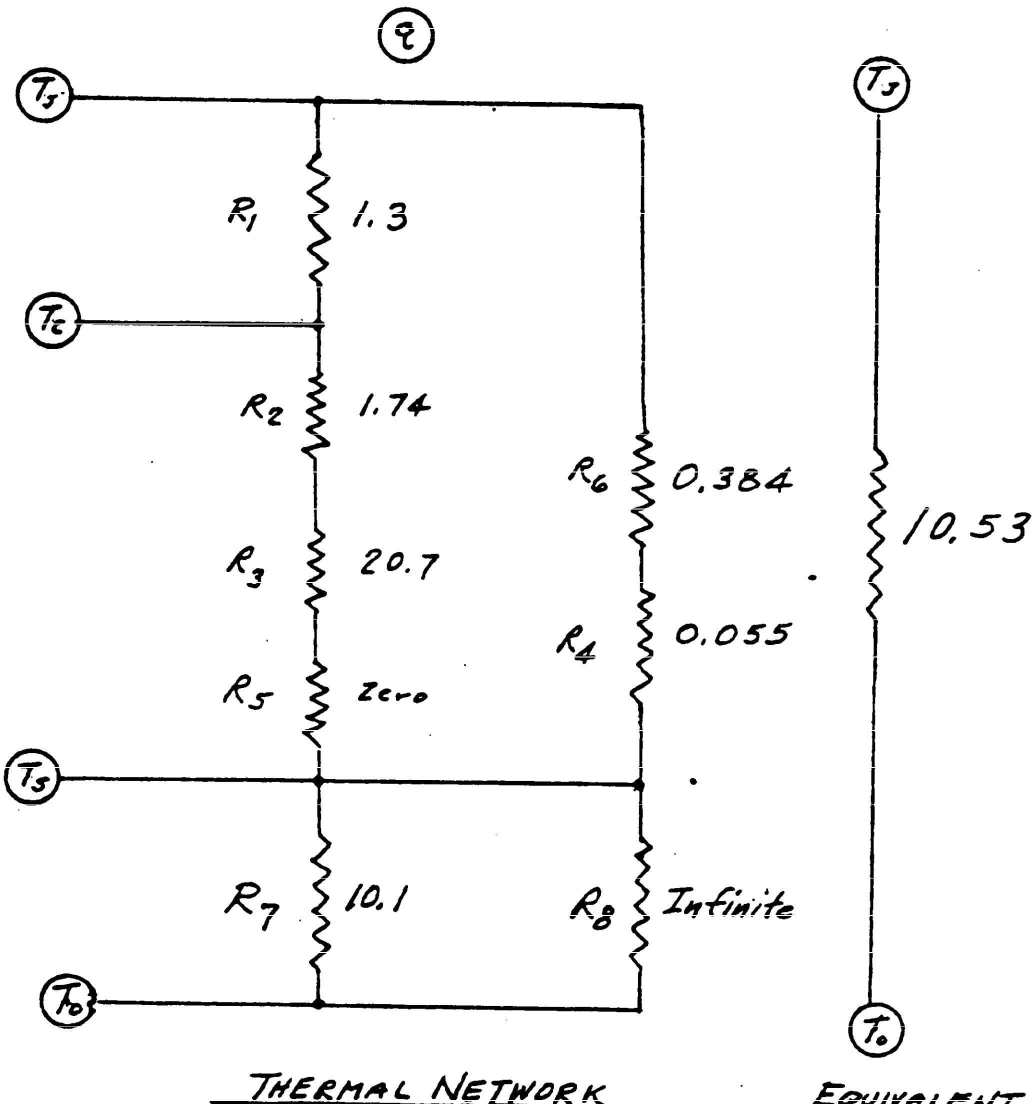


FIGURE 51. Thermal Circuit of P.C. Card

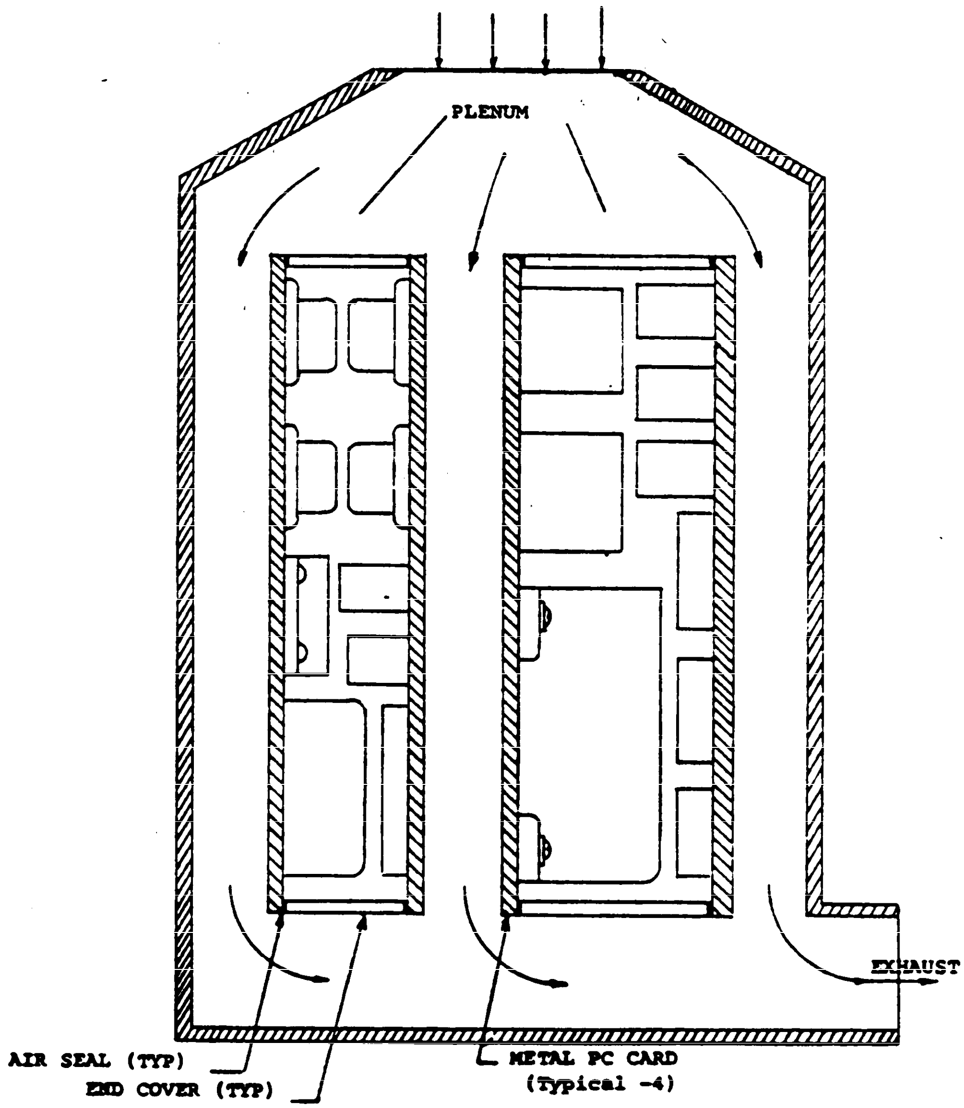


FIGURE 52. Sketch of Configuration for Cooling High Power P.C. Cards

Using 100°C plate temperature and 47°C air (worst case) calculations were made of heat transfer coefficient, air pressure drop, air flow, and heat dissipation at 100-47 = 53°C temperature difference between air and plate, for a series of arbitrarily chosen Reynolds Numbers. These calculations are given in Table XVI and plotted in Figure 53.

TABLE XVI. Predicted Thermal Performance Finless Cold Plates

Re	h w/ ^{2}in.-°C	Watts @ΔT=53°C	Total pressure in of H_2O	Air flow lb./min.
1000	0.037	94	0.042	0.186
2000	.0511	130	.109	.372
4000	.0596	151	.729	.745
6000	.0718	182	1.84	1.11
8000	.079	201	2.80	1.49

The electronic package was next constructed and tested. Table XVII presents the measured performance.

Table XVII lists the design power dissipation per plate and per duct. From the curves of Figure 53 the air flow and pressure drop for each duct were determined. The air temperature rise and exit air temperature for each duct were then calculated and are also given in Table XVII.

Total air flow is 2.67 lb./min. which is less than the 3 lb./min. available for 500 watt heat load.

Ducts 1 and 3 must be throttled to limit the air flow to that required. The full pressure available would force 1.2 lb./min. through each duct.

TABLE XVII. Finless Cold Plate Measured Performance With 74°C Air and 100°C Plate Temperature

Plate #	Duct #	Dissipation Watts	Air Flow lb./min.	Pressure in water	Exit Air T °C
1		150			
2		100			
3		80			
4		170			
	1	150	0.65	0.52	77
	2	180	1.1	1.7	69
	3	170	.92	1.27	71
		500			

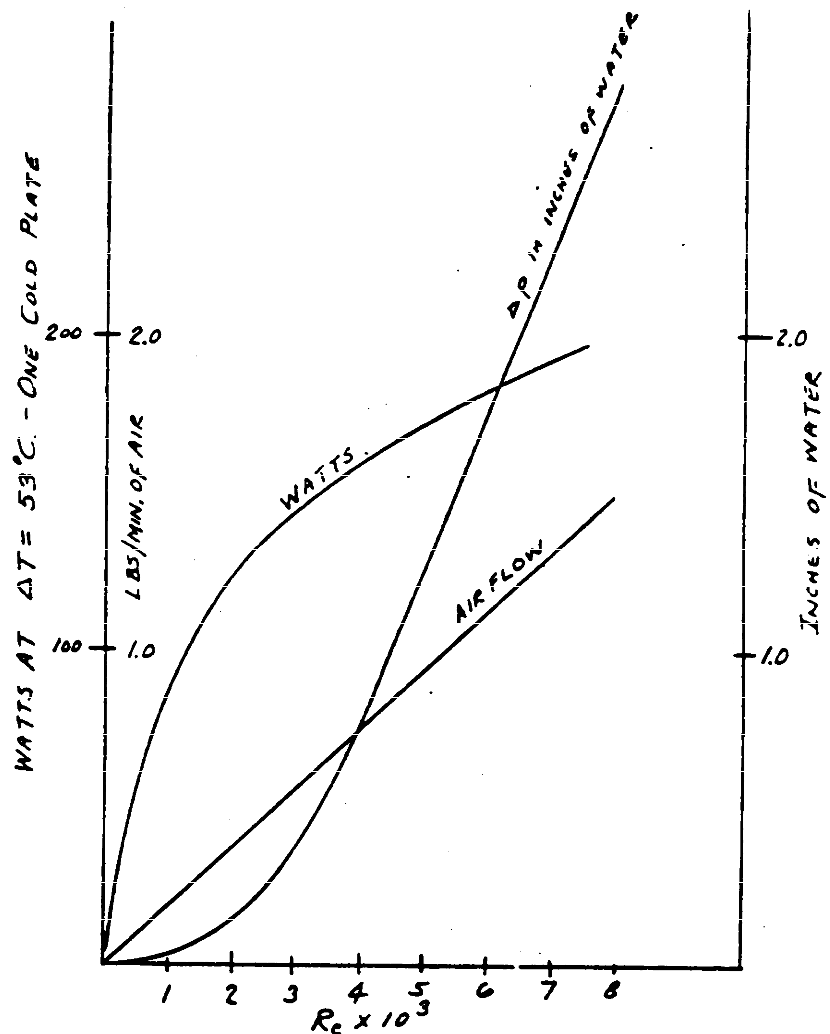


FIGURE 53. Computed Performance with Air at 47°C

Pressure drop estimates are always uncertain due to the effects of surface bends, eddies, and other minor losses of head. This design afforded a comfortable margin between 1.7 in. estimated and 2.0 in. available.

The mounting plates for high dissipation elements must be designed to give the assumed interface resistance of 0.6°C/W. Using anodized aluminum plates of thickness t inches, the heat flow will spresd quite uniformly over the 6 x 8 inch surface.

$$R = \frac{t}{KA} = \frac{t}{(4)\ (48)} = 0.0019t$$

A thickness of 0.093 inch was chosen for mechanical reasons. Since this resulted in a very low R value no analysis of the spreading resistance was deemed necessary. The anodized surface 0.003 inch thick offers a resistance

$$R = \frac{0.003}{(0.42)\ (48)} = 1.5 \times 10^{-4}\text{°C/W}$$

The contact resistance of a properly mounted power transistor is from 0.2 to 0.4°C/W. The assumed interface resistance of 0.6°C/W is therefore, conservatively high.

9.4.8.2 Finned heat sinks (not ducted). When a smooth convective heat transfer surface does not have sufficient area to dissipate a given heat load the area can be increased by the additions of fins. Several configurations of fins are shown in Figure 54. The rectangular configuration is a common one of manageable complexity.

It is helpful to see graphically what effects are produced by fin parameter changes. To understand curve A (heated surface, no fin), Figure 55, consider the following:

$h = kW^n$, where K is a constant
W is coolant weight flow

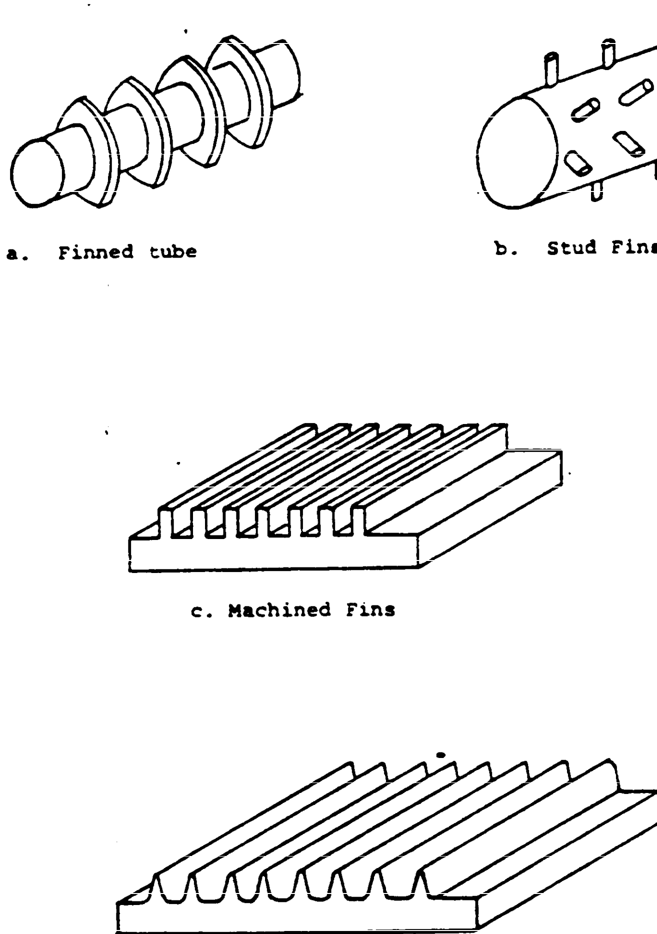
$Q = hA\Delta T$, where Q is heat flow
A is surface area
ΔT is surface temperature-coolant temperature

If Q and A are constant

$$\Delta T = Ch^{-1} = C_2 W^{-n}, \text{ where C and } C_2 \text{ are constants}$$

$$\text{Log } \Delta T = \text{Log } C_2 - n \text{ Log } W \qquad (9\text{-}73)$$

Plotted on log-log paper, this is a straight line with slope -n and y-intercept $\Delta T = C_2$. Values of n for turbulent flow vary from about 0.5 (flow perpendicular to a single cylinder) to about 0.8 (flow parallel to a plane surface), depending on surface shape or configuration.



a. Finned tube

b. Stud Fins

c. Machined Fins

d. Extruded or Rolled Fins

FIGURE 54. Fin Configurations

Curve B, Figure 55, shows the effect of adding a fin.

$$Q = hA\eta\Delta T, \text{ where } \eta, \text{ "fin efficiency"} = \frac{\text{average fin temp. - coolant temp.}}{\text{fin base temp. - coolant temp.}}$$

ΔT = fin base temp - coolant temp.

If Q and A are constant,

$$\Delta T = Ch^{-1}\ \eta^{-1} = C_3\ W^{-s} \qquad (9\text{-}74)$$

where:
-s is the new slope, <-n
C_3 is a constant

Careful analysis of the heat transfer equation for straight fins of rectangular cross section reveals that the minimum slope -s = -n/2, (curve C), where n is 0.75, and that slope -s is affected by changes in fin dimensions as well as by changes in coolant flow. At constant heat dissipation, constant coolant flow K^1 (see Figure 55), and constant fin thickness and conductivity, changing fin length b will cause ΔT to move down from curve A towards curve C along K'. Note that the slope decreases from curve A to B to C. Curve C represents a lower limit of performance beyond which ΔT no longer decreases with increasing fin length. This limit is shown differently in Figure 56 (abscissa approximately 3).

Another interesting effect of fin dimensions and coolant flow is shown on Figure 55. As the coolant flow increases over a fin, a limiting point "p" is reached where the curve for the fin crosses the no-fin curve. At this point the fin becomes useless. At higher coolant flows the fin actually acts as an insulator. Figure 56 also illustrates this phenomenon in different form. Evaporation cooling, with high h, is likely to be inhibited by a fin, which becomes useless when $h \geq 2k_{FIN}/t$.

Maximizing the effectiveness η of a fin is not a good design goal. A thick, stubby fin whose temperature all over nearly equals its root temperature is very effective; its η is nearly 1. But it is of little value, since it adds little to the heat transfer from the hot body because of its small area. The designer must weigh the space available, the cost of fins, coolant flow available, and the temperature sensitivity of the hot body to determine each individual point of diminishing returns.

Figure 57 shows an example of a fin design problem which illustrates the value of the log-log plot described above. One segment of a flat surface with n aluminum fins, carrying 1/n times the total heat to be dissipated (Q) is analyzed. What air flow rate and what fin length should be selected? Follow this procedure:

At any reasonable air flow $V\rho$, like 3 lb./ft.^{2} sec.

$$\text{find } h_{plate} = \frac{.055^{*}_{air}}{L}\left(\frac{V\rho L}{g\mu}\right)^{.75} = 10.88 \text{ BTU/hr. ft.}^{2}\,^{\circ}\text{F}$$

*constant used for plane surfaces

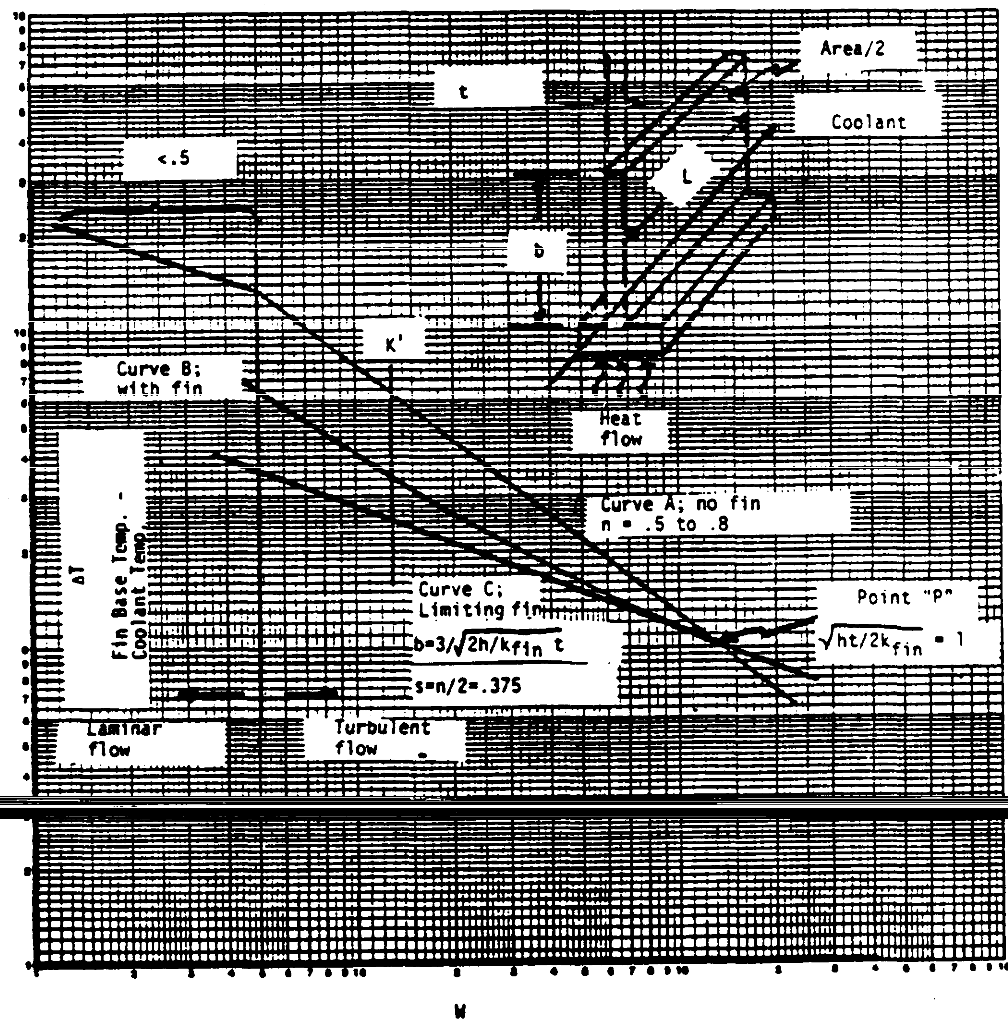


FIGURE 55. Coolant Weight Flow vs. Coolant-Surface ΔT, With and Without Straight Fin (Rectangular Cross Section). Constant Heat Flow

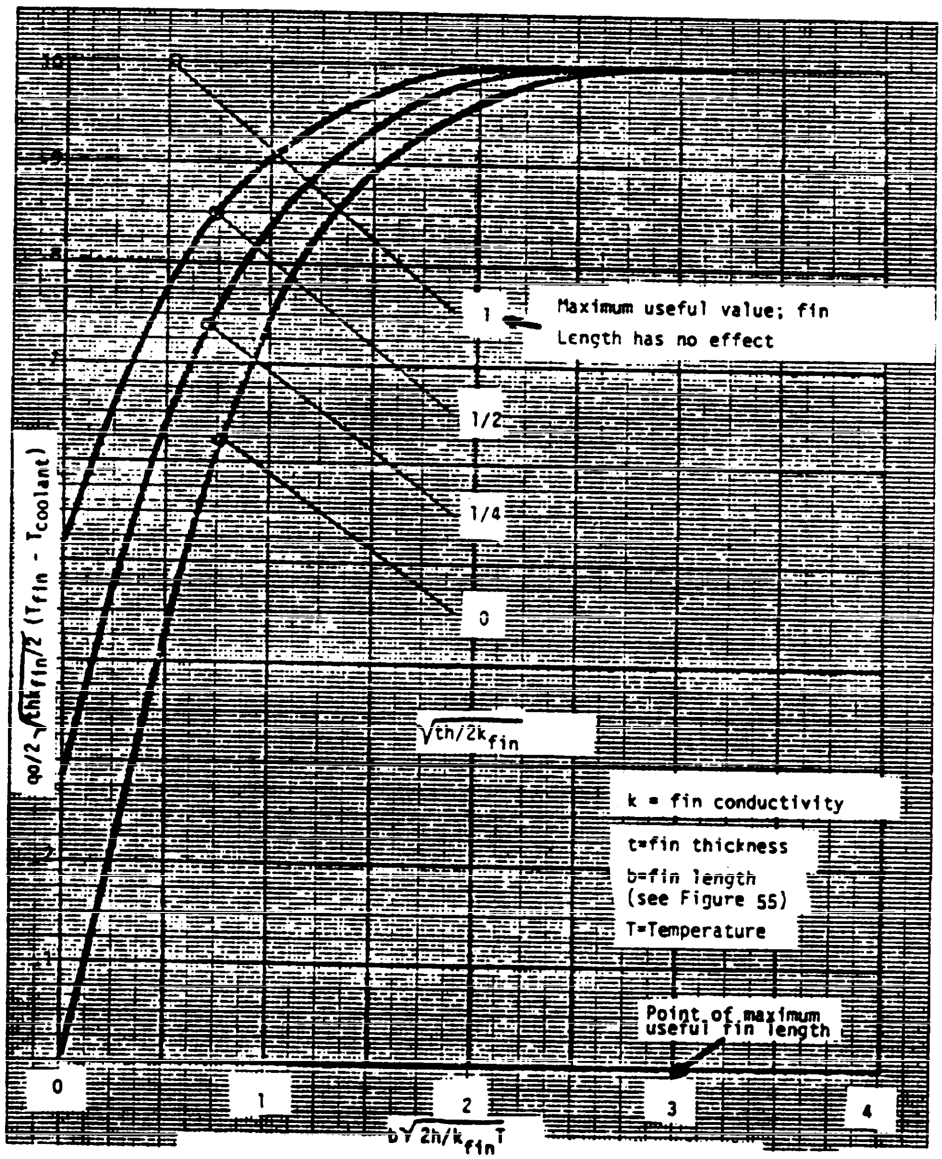


FIGURE 56. Heat Flow Through a Straight Rectangular Fin, Accounting for Fin Tip Heat Loss

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then ΔT_{plate} (without fin) = $Q/_{hA_{plate}}$ = $Q/_{.015h}$ = 38.5°F.

Draw a straight line through A with slope = -.75. This line shows the temperature difference, surface-to-air, of the non-finned surface vs. air flow rate.

Now find $h_{fin} = \frac{.028^{*} k_{air}}{L} \left(\frac{V\rho L}{g\mu}\right)^{.375}$ = 5.54 for $V\rho$ = 3, and the limiting fin length $b = 3/\sqrt{2h/k_{fin}t}$ = 1 ft. Since the ordinate of Figure 56 is 1 in the limiting case, $\Delta T = Q/2\sqrt{thk_{fin}/2}$ = 1.8°F (point C), Figure 57. Draw a line through C with slope = -.75/2 = -.375. This line shows the best that can be done by a fin of maximun useful length. Note that this maximum useful length gets shorter as $V\rho$, and therefore, h_{fin} increases. We have now bracketed the extremes between which the fin must fall.

Now by selecting some other more practical fin length, like .083 ft. (1 inch), find the abscissa and ordinate on curve 56.

Thus,

$$b\sqrt{2h/k_{fin}\,t} = .247 = \text{abscissa.}$$

Since

$$\sqrt{th/2k_{fin}} = 0.15, \text{ (use the lowest curve).}$$

The ordinate is about $.2 = Q/2\sqrt{thk_{fin}/2}\ \Delta T$.

Then,

$$\Delta T = 9.2°F \text{ (point B)}$$

The designer can now plot the curve for a 1 inch fin with a few points, knowing that it will approach the limiting curve. Calculating points at $V\rho$ = 10, 50, 150, and 1000 pretty well tells the story. At $V\rho$ = 3 lb./$_{sec.}$ $_2$ft.^{2}, for instance, cross sectional flow area A' = .03b = (.03)(.083) = .0025 ft.^{2}, W = A'$V\rho$ = .0075 lb./sec. = 6 ft.^{3}/min. air flow at standard conditions. Then 6 times n fins = total air flow required in ft.^{3}/min.

9.4.8.3 Fin spacing. For fully developed turbulence, where $\frac{V\rho L}{q\mu}$, the Reynolds number, is $\geq 10^5$, the boundary layer thickness is $d = .376\ Re^{-1/2}$. Fins should be at least 2d apart to avoid interference with each other.

9.4.8.4 Ducted finned sinks and cold plates. Shrouded fins provide a much more satisfactory design for forced air cooling than open type fins. As shown in Figure 58 this design actually forms closed ducts. Heat transfer in closed ducts has been thoroughly studied and is accurately predictable. Also, the spillage of air out of the open duct is avoided. Forced air cooled cold plates with internal air ducts are efficient and effective.

*Constant used if 2 sides of flat plate are exposed.

Q = 2 watts = 6.82 BTU/hr.
t = .01 ft.
L = .5 ft.
area $A_{plate\ segment}$ = 3tL = .015 ft.^{2}
area $A'_{flow\ cross\ sect.}$ = 3tb

k_{fin} = 125 BTU/hr. ft.^{2} °F per ft.
k_{air} = 1.57 x 10^{-2} BTU/hr. ft.^{2} °F per ft.
μ_{air} = 4 x 10^{-7} lb.-sec./ft.^{2}
g = 32.2 ft./sec.^{2}

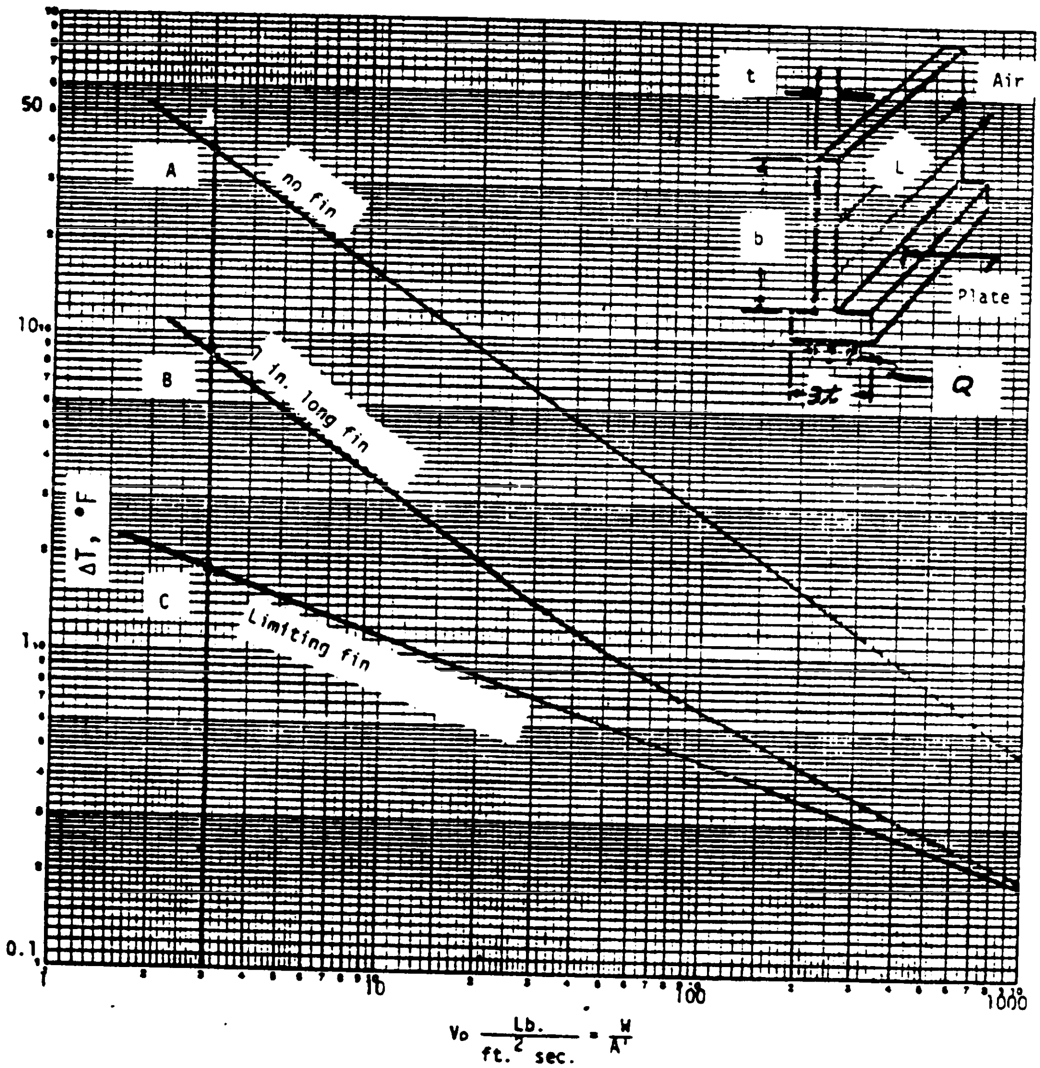


FIGURE 57. Fin Design Example

The thickness of the metal web between ducts should be sufficient to keep the temperature drop small, about 1 or 2°C. The thickness of the shrouding plate is not important since this plate conveys only a small fraction of the total heat. Equation 9-75 (from Giedt, Reference 9) has been found to be sufficiently accurate for computing the average heat transfer coefficient for ducted fins.

$$h_{av} = \frac{k}{d} \left[3.65 + \frac{0.0668\ (d/L)\ Re\ Pr}{1 + 0.04\ \left[(d/L)\ Re\ Pr\right]^{2/3}} \right] \qquad (9\text{-}75)(D.E.)$$

where k = thermal conductivity of the air

$$d = \frac{4ab}{2(a+b)} = \frac{4 \times \text{duct cross section}}{\text{duct parimeter}}$$

l = length in direction of flow

Re = Reynolds Number = $\frac{\rho v d}{\mu}$

v = fluid velocity

Pr = Prandtl Number = $\frac{C_p \mu}{k}$ = 0.7 for air under normal conditions

This equation is applicable when the duct wall is at a substantially uniform temperature and for Re less than 2000 corresponding to laminar flow.

Jakob (Reference 6) gives a somewhat different equation for the average heat transfer coefficient for round and rectangular ducts.

$$h_{av} = 1.615\ \emptyset^{1/3} k/d \qquad (9\text{-}76)$$

where $\emptyset = \frac{4}{\pi}\ \frac{c_p G}{KL}$

G = mass flow rate

Equation 9-76 fits experimental data very well when $\emptyset$ is larger than 10.

For ducts of height 10 or more times width no one equation covers such a wide range of $\emptyset$. Jakob gives two equations for flat ducts.

$$h_{av} = 1.85\ \emptyset^{1/3}\ k/d \text{ for } \emptyset > 2000$$
$$= 0.5\ \emptyset\ k/d \text{ for } \emptyset < 6 \qquad (9\text{-}77)$$

For $\emptyset$ between 6 and 2000, values of $\frac{h_a d}{k}$ may be determined from graphical data, such as that given in Figure 22-7, Reference 6. The use of these equations is illustrated by the following examples.

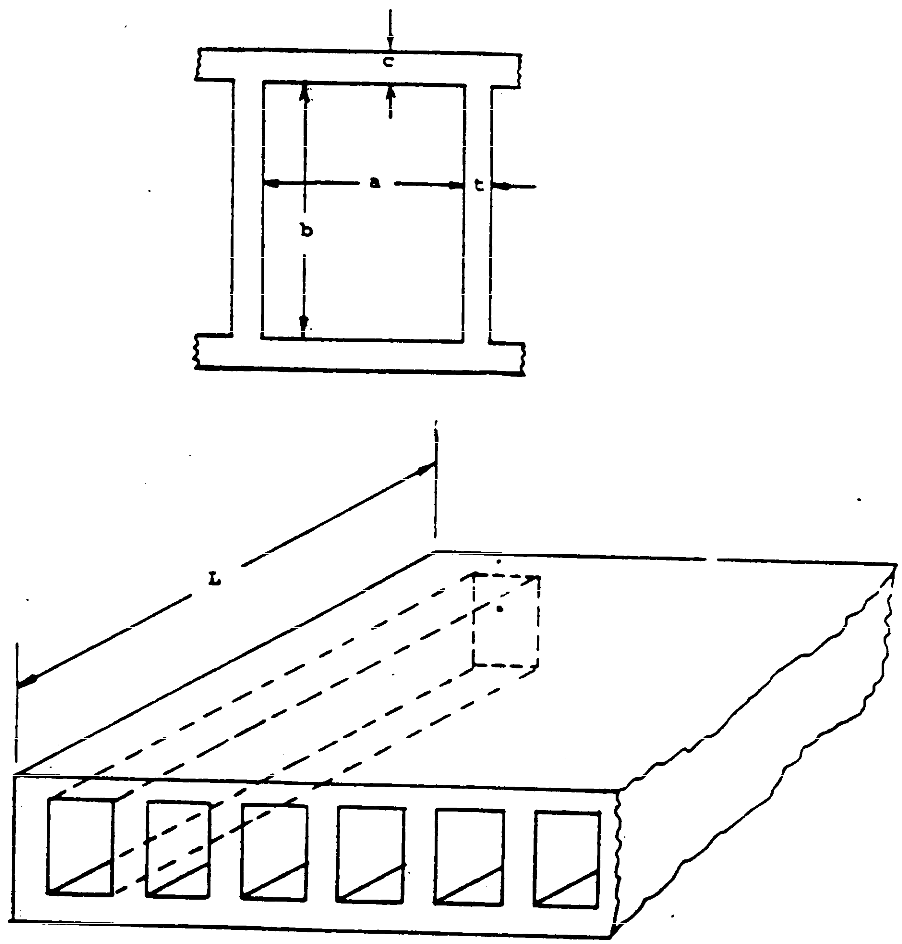


FIGURE 58. Ducted or Shrouded Fins in Cold Plate

Example:

A ducted fin design 12 inches long has 58 passages measuring 0.1 x 0.5 inches. The heat load is 100 watts. Air supply is 20 cfm at 60°C (140°F) entering temperature. The fin surface temperature is designed to be 70°C (158°F) or less.

Determine whether this surface temperature requirement can be met with the specified configuration and flow rate.

Find air temperature rise.

$$WC_p \, \Delta T = q$$

$$\frac{20 \times 1728}{60} \; \frac{\text{cu. in.}}{\text{sec.}} \times \frac{0.0653}{1728} \; \frac{\text{lbm.}}{\text{cu. in.}} \times 0.24 \times 1899 \; \frac{\text{w - sec.}}{\text{lbm. - °C}} \times \Delta T = 100$$

$\Delta T = 10°C$
Average air T = 65°C
Average ΔT surface - air = 5°C

Find heat transfer rate.

$$d = \frac{(4)(0.1)(0.5)}{2(0.1 + 0.5)} = 0.167 \text{ in.}$$

Assume Pr = 0.7, k=0.016 BTU/ft.-hr.-°F = (0.016)(.044) = 0.000705 w/in.-°C

$$\rho = 0.07/1728 = 4.05 \times 10^{-5} \text{ lbm/cu. in.}$$

$$\mu = 0.047 \times 23.3 \times 10^{-6} = 1.09 \times 10^{-6} \text{ lbm/sec.-in.}$$

$$Re = \frac{(4.05 \times 10^{-5})(0.167)\ V}{1.09 \times 10^{-6}} = 6.2\ V \text{ (in./sec.)}$$

Duct cross section area = (58)(0.1)(0.5) = 2.9 sq. in.

$$V = \frac{20 \text{ cfm} \times 1728}{60 \times 2.9} = 199 \text{ in./sec.}$$

$$Re = (6.2)(199) = 1230$$

This assures Laminar flow.

$$D/L = \frac{0.167}{12} = 0.0139$$

$$h_{av} = \frac{0.000705}{0.167} \quad 3.65 + \frac{(.0668)(.0139)(1230)(0.7)}{1 + (0.04)(5.25)}$$

$$= 0.0184 \text{ w/sq. in.}$$

The heat transfer area is (58)(12)(1.2) = 835 sq. in.

$$q = (835)(.0184)(5) = 77 \text{ watts}$$

To obtain the watt removal, the average surface-air temperature difference must be 6.5°C (11.7°F), instead of 5°C originally estimated.

$$\frac{(Ts - 60) + (Ts - 70)}{2} = 6.5$$

$$Ts = \frac{13 + 130}{2} = 71.5°C$$

Alternatively, the air flow rate could be increased to $(20)\frac{(13)}{10} = 26$ cfm to maintain a 70°C fin surface temperature with a dissipation of 100 w.

A prototype of this design was built and tested. The measured surface temperature was 11.5°C above ambient at 20 cfm, 100w instaed of the 10°C desired, verifying the calculations.

The metal thickness required is estimated as follows:

no. of webs = 58 x 2 + 11 = 117
web area = 12 x thickness (t)
web length = 0.5 in.
heat flow per web = $\frac{100}{117}$ = 0.86 watt
k for aluminum = (100)(.044)
= 4.4 w/in. - °C

$$\text{drop in T} = \Delta T \text{ in web} = \frac{ql}{kA} = \frac{(0.86)(0.5)}{(4.4)(12t)} = \frac{0.00814}{t}$$

$$\text{For 1°C drop, } t = \frac{0.00814}{1} = 0.00814 \text{ in.}$$

The actual thermal resistance of the web 0.02 in. thick was

$$\frac{0.5}{(4.4)(.24)} = 0.47°C/w,$$ and the drop in T was 0.4°C.

The base plate was 12 x 7 in. and 0.19 in. thick. Thermal resistance is

$$\frac{0.19}{(4.4)(84)} = 0.00515°C/W$$

Drop in T for 100w = 0.515°C

Ducted fins are subject to the same limitations due to boundary layer as open fins. Reynolds number should be less than 2000 when equation 9-77 was used. Between Re = 2000 and Re = 10,000 an unstable flow regime exists, characterized by the formation of a thick layer of effectively stationary air which inhibits heat transfer. If Re is larger than 2000 it should be increased to more than 10,000 so that turbulence is full developed. This increases heat transfer but usually requires excessive air pressure and velocity resulting in undesirable noise and vibration. Also the additional blower power required usually adds to the heat load.

9.4.8.5 Jet impingement cooling of heat sinks. One method for producing relatively large forced-convection heat transfer coefficients on a surface by air (or other gas) is the use of jets impinging on the surface.

As the air jet approaches close to the surface it turns by an angle of 90°C and thereby becomes what is called a "wall jet" (after this 90°C turn).

As the wall jets from two adjacent impinging jets approach each other their interference forces the flow to separate from the surface and form a stream, often of relatively low velocity, flowing past the impinging jets to reach the exit where the gas is removed. This flow may be called the "spent flow." This spent flow tends, however, to deflect the impinging jets somewhat from their initial direction, and can thereby reduce the average convective heat transfer coefficient, and make it non-uniform from the region around one jet compared to the region around another jet nearer the exit.

Figure 59 shows the configurations of the orifice plate and heat transfer surface and the dimensional symbols. Figure 60 shows a common method of supplying the coolant gas by means of a plenum chamber and a fan blower.

Either round holes or slots may be used. The significant parameters are: shape and size of holes, air velocity, distance from orifice to cold plate, spacing of holes, curvature of the orifice plate (if not flat), and the thermal properties of the gas.

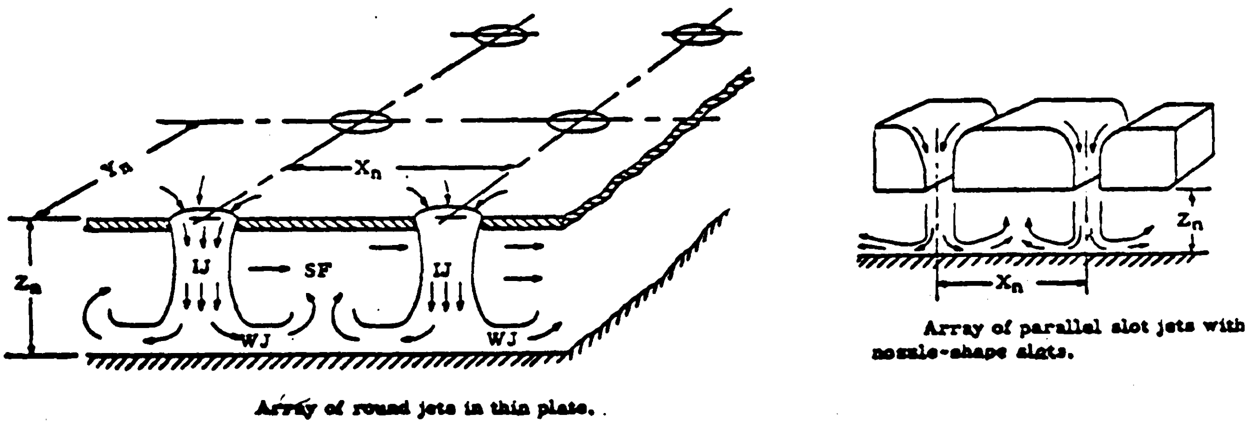


FIGURE 59. Jet Plate Details

IJ = jet flow
WJ = wall jet flow
SF = spent flow

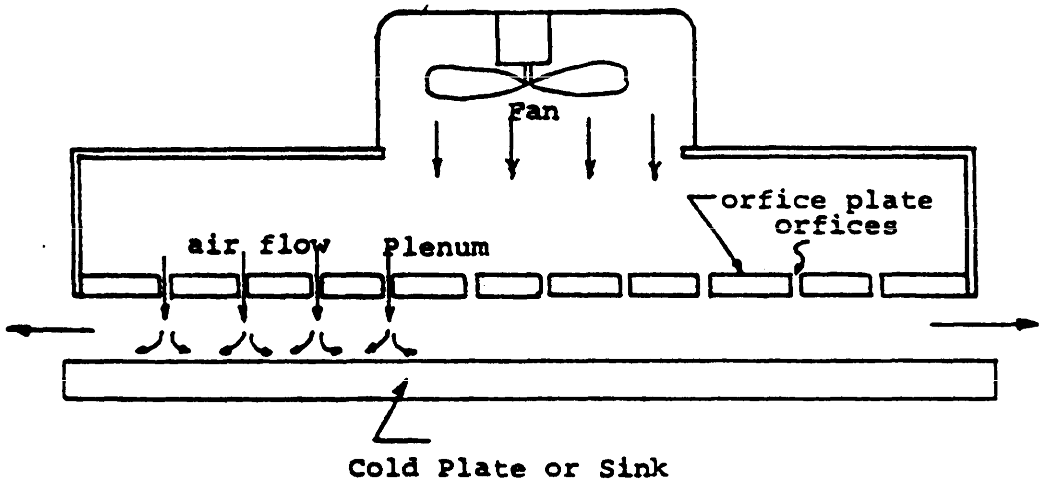


FIGURE 60. General Configuration for Jet Cooling

Jet cooling is applicable to flat surfaces such as cold plates, heat exchangers, or any panel on which numerous heat sources are mounted. The surface may be smooth or finned and may have moderate curvature. The exhaust air at low velocity can further be used for cooling a load of low heat concentration.

Several empirical correlation equations have been developed. Equation 9-78 is applicable to square arrays of round holes in thin plates.

$$h_{av} = \frac{k}{D_o} \phi_1 \phi_2 Re_o^{\,m} \left[\frac{Z_n}{D_o}\right]^{0.091} Pr_o^{\,1/3} \qquad (9\text{-}78)$$

where:

h_{av} = heat transfer coefficient based on temperatures of entering gas and heat transfer surface
D_o = hole diameter

Where the subscript "o" means the value at entering gas temperature,

ϕ_1 = a constant depending on X_n/D_o and Re_o.
ϕ_2 = a constant to account for the effect of the spent flow, evaluated for the parameter

$$(\pi/4)(\text{no. of rows} - 1)\ D_o/Y_n.$$

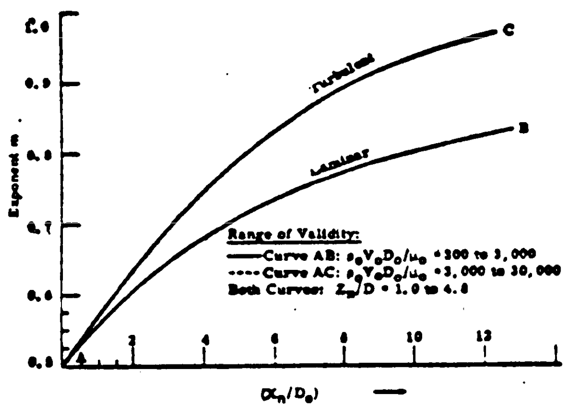
$Re_o = \rho_o D_o V_o/\mu_o$

m = a constant depending on X_n/D_o and Re_o.

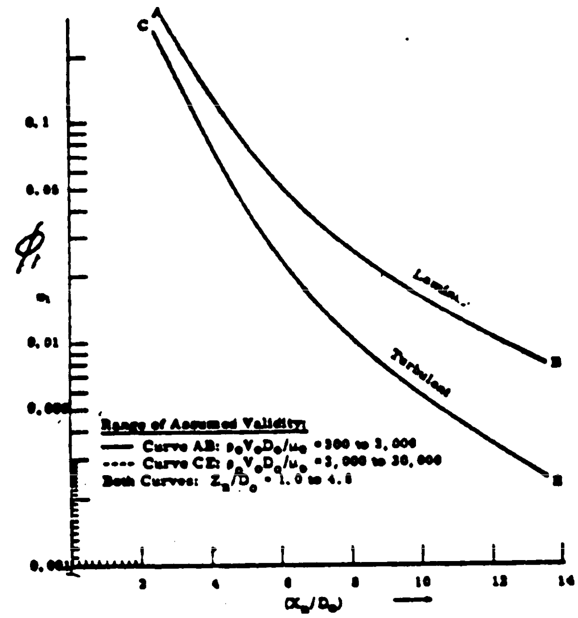
The functions $\emptyset_1$, $\emptyset_1$, and m are plotted in Figure 61.

The ratio Z_n/D_o should have a value between 1.0 and 5.

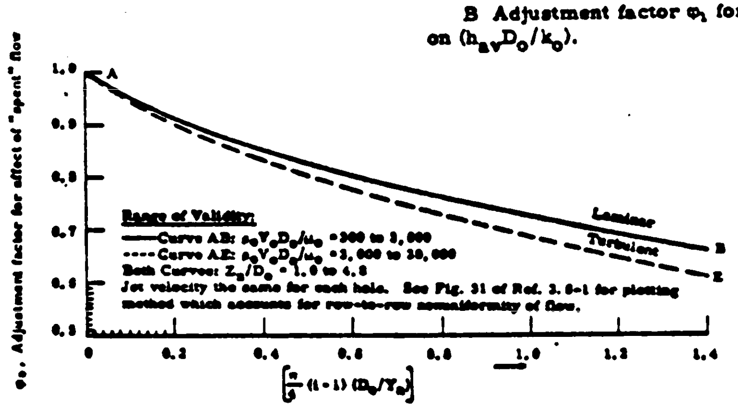
The hole diameter should be between 0.01 and 0.08 inch, which is the range used in developing the correlation equation. The parameter for the function $\emptyset_2$ should have a value between 0.2 and 1.4. The ratio X_n/D_o should not exceed 14. The effect of Re_o is accounted for as shown in Figure 61 where the value of 3000 indicates the transition from laminar to turbulence flow.



A Exponent of Reynolds number.



B Adjustment factor φ_1 for effect of (X_n/D_o) on $(h_{av}D_o/k_o)$.



C Adjustment factor φ_2, for effect of "spent" flow

FIGURE 61. Exponent and Factors for Equation 9-78

The shape of the hole entrance affects the discharge coefficient. Holes in thick plates (tube jets) and trumpet shaped, streamline holes will provide increased air flow but no good correlation formula has been developed to take this into account. It is difficult to make holes less than 0.1 inch diameter with truly square edges. Equation 9-78 may give somewhat pessimistic results because the small holes used had rounded edges.

For further details of jet impingement cooling see Reference 49.

9.4.8.5.1 Determination of required pressure for jet air cooling. The pressure drop between the plenum and the peripheral discharge area can be treated as the sum of three drops; that for the orifice, the 90° bend, and a sudden expansion. The orifice drop is given by equation 9-79.

$$V_o = \frac{Q}{A_o} = \frac{c_d}{\sqrt{1 - (D_o/D_1)^4}} \cdot \sqrt{2\ g\ H} \qquad (9\text{-}79)$$

where:

Q = volume flow rate

D_1 = hydraulic diameter of area feeding one orifice

For the values of Re and D_o/D_1 usually used the discharge coefficient is 0.6. This reduces to equation 9-80.

$$H_o = \frac{V_o^2 \left[1 - (D_o/D_1)^4\right]}{0.72g} \qquad (9\text{-}80)$$

where:

H_o = in. of the gas at entrance temperature

V_o = average velocity, in./sec.

$$D_1 = \frac{4X_n\ Y_n}{2(x_n + Y_n)} \text{ in.}$$

The drop for a 90° bend is given by equation 9-81.

$$H_b = 0.9\,\frac{V_o^2}{2g} \qquad (9\text{-}81)$$

The drop for the expansion is given by equation 9-82.

$$H_e = \left[1 - \frac{D_2}{D_3}^2\right]^2 \frac{V_o^2}{2g} \qquad (9\text{-}82)$$

where:

$$D_2^{\,2} = \frac{4}{\pi} \times \text{total hole area} = N D_o^{\,2}$$

$$D_3 = Z_n \times \text{plate circumference}$$

Equations 9-81 and 9-82 are pessimistic. The minor head losses will generally be less than these equations give.

This estimate should be checked experimentally since it is impossible to compute the effects of flow mixing and surface roughness.

The following design examples illustrate the use of the above equations.

Design Example 1.

An orifice plate 8 x 11 in. has uniformly spaced holes 0.10 in. diameter spaced 0.5 in. on centers. It is spaced 0.5 in. from a parallel flat cold plate. The air flow is 26 cfm at 40°C. Find the thermal resistance and the air pressure required.

$$N = \text{no. holes} = \left[\frac{8}{.5} - 1\right] \left[\frac{11}{.5} - 1\right] = (15)(21) = 315$$

$$\text{Hole area} = 315 \left(\frac{0.1}{2}\right)^2 \pi = 2.47 \text{ sq. in.}$$

$$V_o = \frac{(26)(1728)}{(60)(2.47)} = 302 \text{ in./sec.}$$

$$\rho = 41.2 \times 10^{-6} \text{ lbm/cu. in.}$$

$$\mu = 1.07 \times 10^{-6} \text{ lbm/sec. in.}$$

$$K = 6.86 \times 10^{-4} \text{ w-sec./in.-°C}$$

$$Re_o = \frac{(41.2 \times 10^{-6})(302)(0.10)}{1.07 \times 10^{-6}} = 1164$$

The flow is laminar.

$X_n/D_o = \frac{0.5}{0.1} = 5$ From Figure 61 $m = 0.7$ and $\phi_1 = 0.08$

$\frac{\pi}{4}$ (i-1) $D_o/Y_n = 2.35$ and $\phi_2 = 0.60$

From equation 9-78 with $Pr^{1/3} = 0.888$

$$h_{av} = \frac{6.86 \times 10^{-4}}{0.1} (0.08)(0.60)(1164)^{0.7} \left(\frac{0.5}{0.1}\right)^{0.091} (0.888)$$

$$= 0.0474 \text{ w/sq. in.-°C}$$

$$g/\Delta T = (0.0474)(8)(11) = 4.17 \text{ w/°C}$$

$$R = \frac{1}{4.17} = 0.240 \text{ °C/W}$$

A commercially produced jet impingement heat exchanger of these dimensions is claimed to develop a thermal resistance of 0.25°C/W

Estimate of pressure drop

From equation 9-80

$$H_o = \frac{(302)^2 \left[1 - \frac{0.1}{0.5}\right]^4}{(0.72)(32.2 \times 12)}$$

$$= 330 \text{ in. of air}$$

From equation 9-81

$$H_b = 0.9 \frac{(302)^2}{(2)(32)(12)}$$

$$= 107. \text{ in. of air}$$

From equation 9-82

$$D_2 = 2\sqrt{\frac{4(2.47)}{\pi}} = 3.54 \text{ in.}$$

$$D_3 = (0.5)(2)(8 + 11) = 19 \text{ in.}$$

$$H_e = (1-0.0347)^2 \left[\frac{(302)^2}{(2)(32.2)(12)}\right]$$

$$= 110 \text{ in. of air}$$

Total drop = 547 in. of air

$$= (547)\ \frac{0.0711}{62.4} = 0.623 \text{ in. of water}$$

The commercial heat exchanger cited above is advertised to require a pressure of 0.5 in. of water.

Design Example 2.

If in Example 1 the hole size is reduced to 0.05 in. and the distance Z_n is reduced to 0.25 in. so that Z_n/D_o ratio is the same value. Find thermal resistance and pressure required.

$$N = 315$$

$$\text{Hole area} = 315\left(\frac{0.05}{2}\right)^2 \pi = 0.618 \text{ sq. in.}$$

$$V = \frac{(26)(1728)}{(60)(0.618)} = 1210 \text{ in./sec.}$$

Re = 2333; The flow is laminar

$$X_n/D_o = 10$$

From Figure 61, m = 0, and $\phi_1 = 0.15$

$$\frac{\pi}{4}\ (i-1)\ D_o/Y_n = 1.18$$

From Figure 61, $\phi_2 = 0.71$

From equation 9-78 with $Pr^{1/3} = 0.888$

$$h_{av} = \frac{6.86 \times 10^{-4}}{0.05}\ (0.015)(0.71)(2333)^{0.9}\left(\frac{.25}{.05}\right)^{0.091}\ (0.888)$$

$$= 0.161 \text{ w/sq. in.-°C}$$

$$\frac{q}{\Delta T} = (0.161)(8)(11) = 14.2 \text{ w/°C} \qquad R = \frac{1}{14.2} = .0704 \text{ °C/W}$$

Pressure drop from equation 9-80

$$H_o = \frac{(1210)^2\left[1 - \frac{.05}{.5}\right]^4}{(0.72)(32.2)(12)} = 5262 \text{ in. air}$$

From equation 9-81

$$H_b = 0.9\ \frac{(1210)^2}{(2)(32.2)(12)} = 1705 \text{ in. air}$$

From equation 9-82

$$D_2 = 2\sqrt{\frac{(4)(.618)}{\pi}} = 1.77 \text{ in.}$$

$$D_3 = (0.25)(2)(8 + 11) = 9.5 \text{ in.}$$

$$H_e = \left[1 - \frac{1.77}{9.5}^2\right]^2 \left[\frac{1210^2}{(2)(32.2)(12)}\right] = 1765 \text{ in. air}$$

Total drop = 8732 in. air, or 9.95 in. of water.

The decreased thermal resistance is obtained at the expense of excessive blower pressure. It is more economical to use larger orifices and fins on the cold plate, which increase the heat transfer area.

Design Example 3.

Design a jet impingement cooler to remove 200 watts from a cold plate measuring 8 x 8 in. Plate temperature must be 80°C or less. Incoming air temperature is 40°C. Exit air temperature must be 70°C or less. The pressure must be less than 2 inches of water. The required air flow is fixed by heat rate and temperatures. With the exit air at 70°C,

$$200 \text{ w} = Q(\text{cu. in./sec.}) \, \rho C_p \, (\Delta T_A)$$

$$Q = \frac{200}{(41.2 \times 10^{-6})(455)(30)}$$

$$= 355 \text{ cu. in./sec.} = 12.3 \text{ cfm}$$

The pressure requirement constrains the value of V_o. Since most of the heat loss is in the orifice, from equation 9-80

$$H_o = 2(\text{in. of water}) \times \frac{62.4}{0.0711} = 1755 \text{ (in. of air)}$$

$$1755 = \frac{V_o^2 \left[1 - (D_o/D_1)\right]^4}{(0.72)(32)(12)}$$

D_o/D_1 can be neglected since only an approximate value is required.

$$V_o = \left[(1755)(276.5)\right]^{1/2}$$

$$= 696 \text{ in./sec.}$$

It must be less than this. Use 600 in./sec. Therefore, 355 cu. in./sec. = $600A_{hole}$

$$A_{hole} = \frac{355}{600} = 0.592 = N\,(D_o/2)^2\pi$$

$$ND_o^2 = \frac{(0.592)(4)}{\pi} = 0.753$$

Using a uniform square array of holes,

$$N = (8/x_n)^2 \text{ approximately}$$

$$ND_o^2 = 64\ D_o^2/X_n^2$$

$$(D_o/X_n)^2 = 0.0118$$

$$X_n/D_o = 9.2$$

Now estimate the required heat transfer coefficient for a smooth cold plate.

$$200\ w = h_{av}\ (8 \times 8)(T_s - T_o)$$

$$h_{av} = \frac{200}{(64)(80-40)} = \frac{5}{64} = 0.078 \text{ w/sq. in.-°C}$$

The problem is to select values of X_n and D_o which will yield this high value of h_{av}. For a velocity, V_o of 500 inches per second,

$$Re = \frac{(4.12 \times 10^{-6})(600)(D_o)}{1.07 \times 10^{-6}} = 23100\ D_o$$

Summing the requirements:

$$Q = 355 \text{ cu. in./sec. or more}$$
$$V_o = 600 \text{ in./sec. or less}$$
$$ND_o^2 = 0.753 \text{ or more}$$
$$X_n/D_o = 9.2$$

Since a square pattern is assumed, N must be the square of an integer number, i, where i is the number of rows.

The solution may be obtained by assuming values of i, and proceeding as follows:

$$N = i^2$$

$$D_o = \sqrt{\frac{0.753}{N}}$$

$$X_n = 9.2\ D_o$$

$$Re = 23100\ D_o$$

Determine m and $\emptyset_1$ from Figures 61 (a), and (b) for X_n/D_o = 9.2.

m = 0.78, $\emptyset_1$ = 0.02 for laminar flow (Re < 3000)

m = 0.91, $\emptyset_1$ = 0.008 for turbulent flow (Re > 3000)

$\alpha = \frac{\pi}{4}\,(i-1)\,(D_o/X_n)$

Determine $\emptyset_2$ from Figure 61 (c)

Evaluate (from equation 9-78)

$$h_{Av}/(Z_n/D_o)^{0.091} = \frac{6.86 \times .10^{-4}}{D_o}\ \emptyset_1\ \emptyset_2\ (Re)^m\ (0.896)$$

Where 0.896 is obtained from $Pr^{1/3}$

Note that the value of Z_n has not yet been defined, but that $(Z_n/D_o)^{0.091}$ will be close to unity, and will not vary significantly with D_o.

TABLE XVIII. Values of i Solutions

i	N	D_o	X_n	Re	m	ϕ_1	α	ϕ_2	$h_{Av}/(Z_n D_o)^{0.091}$
6	36	0.145	1.33	3350	0.91	0.008	0.426	.82	.038
8	64	0.108	0.998	2506	0.78	0.02	0.595	.81	.0413
9	81	0.0964	0.887	2227	0.78	0.02	0.683	.80	.0416
10	100	0.0868	0.798	2005	0.78	0.02	0.769	.79	.0421
11	121	0.0789	0.7226	1822	0.78	0.02	0.854	.77	.0419
12	144	0.0723	0.665	1670	0.78	0.02	0.939	.75	.0416
15	225	0.0576	0.532	1336	0.78	0.02	1.19	.71	.0416
20	400	0.0434	0.400	1002	0.78	0.02	1.62	.63	.0391

The maximum heat transfer rate occurs with 100 holes in a 10 by 10 pattern, with a hole diameter of .087 inches. The maximum heat transfer coefficient is less than the required coefficient of 0.078 w/sq. in.°C, so that it will be necessary to extend the plate area by means of fins or pins. The value of Z_n is of minor importance, but should be large enough to accommodate the extended surface elements. Assume Z_n = 0.5 in.

Then, $h_{Av} = (0.0421)\left(\dfrac{0.5}{.0868}\right)^{0.091} = 0.494$ w/sq. in.-°C

The surface area must be extended by a factor of $\dfrac{0.078}{0.0494} = 1.58$.

Calculate the pressure drop by equation 9-80, 9-81, and 9-82.

$$H_o = \frac{V_o^2}{.72g}\left[1 - (D_o/D_1)\right]^4 = \frac{600^2}{(0.72)(32.2)(12)}\left[1 - \left(\frac{.0868}{0.1}\right)\right]^4 = 559 \text{ in. air}$$

$$H_b = 0.9\,\frac{V_o^2}{2g} = (0.9)\,\frac{(600)^2}{(2)(32.2)(12)} = 419 \text{ in. air}$$

$$H_e = \left[1 - \left(\frac{D_2}{D_3}\right)^2\right]^2 \frac{V_o^2}{2g} = \left[1 - \frac{(100)(.0868)^2}{(.5)(4)(8)}\right]^2 \cdot \frac{600^2}{(2)(32.2)(12)} = 423 \text{ in.air}$$

Total pressure drop = 1401 in. air = 1.60 in. water.

There will be an additional pressure drop due to the flow over the extended surfaces, but the margin of 2.0 - 1.6 = 0.4 inches of water will be sufficient to accommodate this loss.

If pins are used for extended surface area, and the pins are nested between holes, there will be 9 x 9 = 81 pins, each 0.5 inches long. Assuming the surface area is to be extended by a factor of 1.75 (1.58 required), the pin diameter is determined by:

$$D_p = \frac{A_e(.75)}{N\pi Z_n} = \frac{(8)(8)(.75)}{(81)(\pi)(0.5)} = 0.377 \text{ inches}$$

Use 3/8 diameter pins

The system requirements can therefore, be met with a jet impingement plate assembly with the following parameters:

Jet plane: 100 holes in a 10 x 10 pattern, hole diameter = 0.086 inches (no. 44 drill), hole spacing = 0.79 in.

Impingement plate: 8 x 8 inches, with 81 pins, 0.375 diameter 0.5 inches long, in a 9 x 9 pattern, spaced on 0.79 inch center.

Airflow rate: 355 cu. in./sec. = 12.3 cfm ANSWER

9.4.9 Heat exchangers

9.4.9.1 Types. Heat exchangers are classified with respect to:

(1) Type of construction:

(a) Shell and tube
(b) Extended surface
(c) Flat "thermo-panel"

(2) Type of flow:

(a) Parallel flow
(b) Counterflow
(c) Cross flow (including single-pass and multi-pass, reverse flow, and cross counterflow)

(3) Types of fluids:

(a) Air to air
(b) Air to liquid
(c) Liquid to liquid

Heat exchangers are discussed in more detail in chapter 10.

9.4.9.2 Mechanism of heat transfer. In a heat exchanger, heat is transferred from one fluid to another across a solid wall. The wall itself is commonly made of material of high thermal conductivity but one or both sides may become coated with rust, scale or dirt which has low thermal conductivity. Heat transmission through the wall and the contaminating layers is treated by conduction theory, while heat transfer between the wall and the hot and cold fluids is treated by convection theory. The mechanism of heat transfer in the exchanger is thus, rather complicated, particularly since the temperatures of the hot fluid, the wall, and the cold fluid vary from point to point. For preliminary calculations, the process is simplified by the use of an average overall transfer coefficient U. U is defined as the quantity of heat transferred between the two fluids in watts per square foot of heat transfer surface per °C mean temperature difference between the hot and the cold fluid.

To ensure effective heat transfer, and hence, compact design, it is desirable that turbulence be fully developed in the fluid flow. This may be achieved by making the Reynolds number sufficiently high. Reynolds numbers of about 10,000 for air and (considering pumping power requirements) of about 2300 to 4000 for water are suggested. Sometimes "tubulators" are used, such as screens or rods at the entrance to air passages (see paragraph 9.4.6) or helically twisted partitions inside tubes carrying liquids. Staggered tube arrangements of suitable spacing increase air turbulence.

In counterflow heat exchangers, the exit temperature of the cold fluid can be higher than the exit temperature of the hot fluid. With parallel flow, the exit temperature of the cold fluid must be lower than the exit temperature of the hot fluid. In cross-flow exchangers, the temperature pattern is complex, and the average exit temperature of the cold fluid is often (but not always) lower than the average exit temperature of the hot fluid.

9.4.9.3 Altitude effects. The thermal performance of heat exchangers fundamentally depends on the coefficient of convective heat transfer, h, between the solid walls and the two fluids. The equations for forced convection show that regardless of the density of the flowing fluid, h will be substantially constant if the maximum mass velocity G_{max}, which occurs at the minimum cross section of the flow channel, remains constant. For a heat exchanger employing atmospheric air as the cold fluid, it follows that the overall coefficient of heat transfer U is not appreciably affected by altitude, provided that the mass flow rate remains constant.

Now

$$G_{max} = \rho V_{max} \quad (9\text{-}83)$$

where

G_{max} = maximum mass velocity, lb./(sec.-ft.^{2})

V_{max} = velocity at minimum cross section, ft./sec.

ρ = fluid density, lb./ft.^{3}

Therefore,

$$V_{max} = G_{max}/\rho \quad (9\text{-}84)$$

The air pressure drop through the heat exchanger, Δp, is approximately proportional to ρV^2_{max}; i.e.,

$$p = K \rho V^2_{max} = K\rho \left(G_{max}/\rho\right)^2 = KG^2_{max}/\rho \text{ (approx.)} \quad (9\text{-}85)$$

where K is a constant. Therefore, when ρ decreases at altitude while G_{max} is held constant, the pressure drop is increased, approximately in direct proportion to $1/\rho$. To avoid this increase in pressure drop, the cross sectional area of the flow passages will have to be increased, that is, designed for maximum altitude conditions. (Reference paragraph 9.2.3)

9.4.9.4 Heat exchanger selection. The usual heat exchanger selection and design problem involves the transfer of heat at a specified rate from a specified hot fluid to a specified cold fluid, with minimum pumping power and in a minimum space of specified size and shape. Weight and cost are also important considerations. Due to the large number of variables involved, heat exchanger design is usually a cut-and-try process and relies heavily on experience. The subject is discussed in more detail in chapter 10.

With regard to type of construction, it sometimes happens, as in an air-to-water heat exchanger, that the convective (film) coefficient of heat transfer is much smaller on one side than of the other. In such cases it is desirable to use a larger area on the air side, which has the lower coefficient. This is accomplished by use of extended surface in the form of fins. Since heat flows by conduction through each fin, and thus, establishes a temperature gradient, the heat transfer effectiveness becomes less toward the outer edge of the fin. However, there is a gain in heat transfer rate because the fin area is large relative to the area of the basic unfinned surface.

9.4.9.5 Detailed design. The detailed design of a heat exchanger involves numerous interdependent parameters generally requiring an iterative procedure. The following discussion outlines one method of determining size and performance and permits the specification of a heat exchanger. It is recommended that heat exchangers be purchased since the detailed internal design is best left to experts at heat exchanger manufacturers.

A sample heat exchanger design problem is presented in chapter 10. More detailed design procedures employing special charts, are presented in references 9.21 and 50.

In a typical heat exchanger, a "hot fluid" which has absorbed heat from electronic equipment has to be cooled before reentering the equipment. The cooling is accomplished by transferring heat from the hot fluid to a cold fluid. In order to calculate the total rate heat transfer it is necessary to have some mean value of the temperature differnece between the hot and cold fluids. The appropriate mean value is the logarithmic mean temperature difference given by

$$\Delta t_{lm} = \frac{\Delta t_{max} - \Delta t_{min}}{\ln\left(\Delta t_{max} / \Delta t_{min}\right)} \qquad \begin{Bmatrix}9\text{-}86\\ \text{D.E.}\end{Bmatrix}$$

where:

Δt_{lm} = logarithmic mean temperature difference, °C

Δt_{max} = maximum difference at any section between the temperatures of hot and cold fluids, °C

Δt_{min} = minimum difference at any section between the temperature of the hot and cold fluids, °C

This expression is used regardless of whether counterflow, parallel flow, or cross flow is used, though the application to cross flow is somewhat arbitrary.

The overall rate of heat transfer is then given by

$$q = UA\ \Delta T_{lm} \qquad \begin{Bmatrix}9\text{-}87\\ \text{D.E.}\end{Bmatrix}$$

where:

q = rate of heat transfer, watts

U = overall coefficient of heat transfer, $\frac{\text{watts}}{\text{ft.}^2\ °\text{C}}$

A = heat transfer area, ft.^{2}

Neglecting external heat transfer between the heat exchanger and the environment, the heat removed from the hot fluid must equal the heat absorbed by the cold fluid. It follows that

$$q = m_h c_{ph}\ \Delta t_h = m_k c_{pk}\ \Delta t_k \qquad \begin{Bmatrix}9\text{-}88\\ \text{D.E.}\end{Bmatrix}$$

where:

m_h = mass flow rate of hot fluid, lb./sec.

c_{ph} = specific heat of hot fluid, $\frac{\text{watt-sec.}}{\text{lbm - °C}}$

Δt_h = temperature drop through the exchanger of the hot fluid, °C

m_k = mass flow rate of cold fluid, lb./sec.

c_{pk} = specific heat of cold fluid, $\frac{\text{watt-sec.}}{\text{lb.-°C}}$

Δt_k = temperature rise through the exchanger of the cold fluid, °C

For the hot fluid, the mass flow rate and the temperatures entering and leaving the heat exchanger will normally be known, from the cooling requirements of the electronic equipment. In a typical case, the temperature of the cold fluid as it enters the heat exchanger will be known. The temperature of the cold fluid leaving the heat exchanger may also be known. For example, heat exchangers used in shipboard forced-air cooled equipment often are of the air-to-water type in which water temperature rise is often limited to 5°C maximum (from 35°C to 40°C). See MIL-W-21965. Then, the required mass flow rate of the cold fluid can be calculated from equation 9-88. Conversely, if the cold fluid mass flow rate is specified, equation 9-88 yields the temperature rise in the cold fluid.

With entering and leaving temperatures of both fluids known, and with the type of flow specified (counter, parallel, or cross flow), Δt_{max} and Δt_{min} will be known. Then Δt_{lm} can be calculated from equation 9-86.

A typical value of the overall heat transfer coefficient U will be known for the type of heat exchanger under consideration. The rate of heat transfer q is given by equation 9-88. The required cooling surface area can then be calculated from equation 9-87.

The flow passages for the two fluids must be designed with cross sectional areas which will produce Reynolds numbers in the recommended ranges. The Reynolds number is calculated from

$$Re = GD_e/\mu \qquad \begin{Bmatrix}9\text{-}89\\ \text{D.E.}\end{Bmatrix}$$

where:

Re = Reynolds number
G = mass velocity (lb./(sec.-ft.^{2})
D_e = equivalent diameter of flow passage, ft.
μ = viscosity of the fluid (lb./sec.-ft.)

The equivalent diameter of the flow passage is given by

$$D_e = \frac{4 \times \text{cross sectional area (ft.}^2)}{\text{cross sectional perimeter (ft.)}}$$

The mass velocity is given by

$$G = m/\ (\text{cross sectional area, ft.}^2)$$

where m is the mass flow rate (lb./sec.)

9.5 Ducting of air flow

9.5.1 General. Air ducts impose resistances to air flow which must be overcome by pressure differences resulting from the expenditure of energy in maintaining the flow. A reasonably precise estimate of the flow resistance offered by the system is essential for satisfactory duct design. The theoretical resistance can be computed from the methods and data given in section 9.3.

The drop in pressure in air transmission systems is due to friction losses and dynamic losses. Pressure increases and decreases may also be caused by changes in duct areas. The friction losses for turbulent flow are due to the friction of air against the side of the duct, and to internal friction between the air molecules. The dynamic losses are caused by changes in the direction or in the velocity of air flow, and may be caused by changes in size and shape of the cross section of the duct, by elbows, and by obstructions to flow offered by dampers. Turbulent air flow is not desirable in ducts because pressure loss due to friction is directly proportional to the degree of turbulence.

9.5.2 Basic ground rules. The following factors must be considered when designing an air duct system:

a. The air should be conveyed as directly as possible at permissible velocities.

b. Avoid sharp elbows and bends in ducts. Splitters and turning vanes should be used to minimize pressure losses.

c. Avoid both sudden enlargements and abrupt contractions. The angle of divergence of enlargements should not exceed 20 degrees. The angle of convergence in contractions should not be greater than 60 degrees.

d. For the greatest air carrying capacity, rectangular ducts should be made as close to square as possible. In rectangular ducts, aspect ratios, that is, the ratio of the larger duct cross section dimension to the smaller, greater than 6 to 1 should be avoided.

e. Make the ducts as air tight as possible. All laps should be in the direction of flow. Avoid raw edges on splitters and vanes.

f. Ducts should be made from smooth materials to minimize friction losses.

g. Provide dampers in all duct branches for final balancing.

9.5.3 Design procedures. The steps necessary in the design of an air-duct system are:

a. Lay out the most convenient, direct system to provide for proper air distribution.

b. Lay out the return system.

c. Decide on the temperature of the air that is to be supplied and determine the air requirement for each chassis to be cooled, based upon the dissipated power and predicted temperature rise.

d. Size the ducts, using the methods described.

e. Check the calculations by determining the friction loss from the fan to the discharge outlet at the end of each branch.

f. Determine the total pressure loss from the fan to the farthest outlet. Add the pressure losses for all equipment, outlet loss, and loss in the return-duct system to determine the pressure that must be produced by the fan to insure adequate flow through the system.

9.5.4 Design methods. The following illustrates the use of the basic fluid flow equations to solve for friction loss in a straight duct. The fittings, such as elbows and branch takeoffs, which cause dynamic pressure losses are converted into Equivalent Lengths of Straight Duct (ELSD) by obtaining the ELSD from Table XIX. The equivalent lengths are added to the length of straight duct before calculating friction loss. For any fluid other than air, these calculations are necessary, but for standard air with a density of 0.075 $lb./ft.^3$ and a temperature between 50 to 90°F flowing through a round galvanized duct, a friction chart has been designed to eliminate calculations. Figures 62 and 63 are the air friction charts, divided into two parts, covering air flow of 10 to 1000 cfm in 62, and 1000 to 100,000 cfm in 63. Note that these charts have four variables: air flow (cfm), air velocity (fpm), duct diameter (in.), and pressure loss (in. of H_2O per 100 ft. of straight duct). Determining any two of these four variables immediately fixes the values of the other two.

Note that the size of the ducts are given as diameters. All air ducts are first sized as round ducts; then, if rectangular ducts are to be used, their dimensions can be selected to provide equivalent air carrying capacity. Figure 64 speeds the conversion from round to rectangular ducts.

In sizing a duct system, the equal friction method is recommended. This method allows the entire air-duct network to be designed for an equal friction drop for each foot of duct length, preventing widely varying resistances between sections. The method is fast and relatively simple to use. Little balancing is required for symmetrical layouts in which all runs have about the same resistance.

The step-by-step procedure of the equal friction method follows:

1. Determine the required air flow needed at the inlet temperature desired.

2. Determine the main duct velocity (maximum 2000 ft./min.).

3. From Figures 62 and 63 determine the main duct diameter for the total air flow (cfm) and velocity (fpm) just selected. At the same time read the pressure loss per 100 ft. of straight duct. This figure will be used in sizing the remaining ducts.

4. Along the pressure-loss line, at the air flow rate carried by each section of duct, read from Figures 62 and 63 the corresponding duct diameters and air velocities. Record all information from Figures 62 and 63 in a table for ready reference.

5. Enter the duct lengths in the work table. From Table XIX obtain the ELSD factors for all fittings in the system and enter them in the work table. Calculate the ELSD of all fittings. Add the total friction lengths of all ducts.

6. Determine the actual pressure drop in inches of water for each section of duct.

7. Add the pressure drops from the fan to the end of each duct run. These pressure drops should be almost equal. If not, re-size the ducts. Do this by assuming a higher pressure loss per 100 ft. of duct in those cases in which the pressure drop must be increased. Use a lower pressure loss per 100 ft. if the drop must be decreased. Read the new duct diameters from Figures 62 and 63.

8. Recalculate the ELSD of all fittings for the new diameters.

9. Recalculate the actual pressure drop in inches of water for each section of duct. Remember to use the new pressure losses per 100 ft. that were assumed in step (7) to re-size the ducts.

10. Pressure drops through all runs should now be equal and the system should balance.

11. From Figure 64 select suitable rectangular duct equivalents for the round duct diameters.

12. The fan selected must be able to overcome the static pressure of such accessory equipment as air cleaners or filters. A small factor of safety should also be included for leakage from the system.

Element	Conditions	Eld Factor
1. Round Radius Elbow	$R/D = 0.5$ $R/D = 0.75$ $R/D = 1.0$ $R/D = 1.5$ $R/D = 2.0$	43D 23D 15D 10D 9D
2. Rectangular Radius Elbow	W/D = 0.5; R/D = 0.5, 0.75, 1.0, 1.5 W/D = 1 to 3; R/D = 0.5, 0.75, 1.0, 1.5	79D, 29D, 17D, 11D 57D, 20D, 12D, 8D
3. Rectangular Radius Elbow	No. Vanes 1; R/D = 0.5, 0.75, 1.0, 1.5 No. Vanes 2; R/D = 0.5, 0.75, 1.0, 1.5	42D, 10D, 8D, 7D 27D, 7D, 6D, 9D
4. Round Section Mitre Elbow		53D
5. Rectangular Mitre Elbow		76D
6. Rectangular Square Elbow	Single Vane Thickness. Double Vane Thickness. (Use Manufacturer's Data)	21D
7. Rectangular Tee	Consider equal to an elbow. Base loss in each elbow on duct dimension indicated.	
8. Radius Tee	Consider equal to an elbow. Base loss in each elbow on duct dimension indicated.	
9. 45° Elbow	Either rectangular or round. Vaned or Unvaned.	½ of value for similar 90° elbow.
10. Expansion	$\theta = 5°$ $\theta = 10°$ $\theta = 20°$ $\theta = 30°$ $\theta = 40°$	10D 17D 27D 36D 43D
11. Contraction	$\theta = 30°$ $\theta = 45°$ $\theta = 60°$	1D 2D 4D
12. Transition Piece		9D
13. Abrupt Entrance		30D
14. Abrupt Exit		60D
15. Bell Mouth Entrance		2D
16. Bell Mouth Exit		60D
17. Re-entrant Entrance		51D
18. Abrupt Contraction	v/V = 0.00 0.25 0.50 0.75	30D 27D 19D 11D
19. Abrupt Expansion	v/V = 0.00 0.20 0.40 0.60 0.80	60D 39D 22D 9D 2D
20. Double Elbows	Loss for both elbows L = 0 L = D Elbows Vaned	26D 19D 9D
21. Double Elbows	L = 0 L = D Elbows Vaned	38D 40D 12D
22. Double Elbows	L = 0 L = D Elbows Vaned	24D 28D 12D
23. Double Elbows	Direction of arrow Reverse direction	70D 62D
24. Pipe Running Through Duct	E/D = 0.10 0.25 0.50	12D 33D 120D
25. Bar Running Through Duct	E/D = 0.10 0.25 0.50	40D 80D 240D
26. Streamlined Covering Over Obstruction	E/D = 0.10 0.25 0.50	4D 14D 54D

TABLE XIX. Equivalent Lengths and Standard Duct Factors for Common Fittings

9.5.5 Construction. Aluminum is the most commonly used metal for duct work. It is resistant to corrosion, light in weight, easy to fabricate, and, because it is thicker than steel for the same weight, it tends to have greater rigidity. The construction of fittings and changes of shape cannot be definitely outlined, because of the various conditions peculiar to each installation, but in general, long radius elbows and graded changes in shape tend to maintain uniform velocities accompanied by decreased turbulence and lower resistance. Examples of some generally accepted duct fittings are shown in Figure 65.

9.6 Design of forced air cooled cabinets and enclosures

9.6.1 Unsealed or open cabinets. Electronic equipment cabinets which suck in cool external air, circulate it over the heat sources, and exhaust the warm air to the environment are, for the purposes of this discussion, considered to be unsealed or open. With cabinets of this type the following are recommended:

a. Locate the air intake near the base of the cabinet, but not so low that dirt and water can enter floor mounted cabinets. Also, equipment cabinets are often installed tightly side by side and the rear of the cabinet may be snugly resting against a wall or bulkhead. Therefore, air intakes at the front of cabinets should be the first choice, locationwise.

b. The air exhaust should be near the top of the cabinet, but not on top since foreign objects or liquids can fall into the electronics. The top sides of the cabinet should be the first location choice. However, hot exhaust should not be blown into the faces of personnel. Usually, upward facing louvers or diverters can be used to direct the air stream upward.

c. The air should be circulated from the bottom to the top of the cabinet as indicated above. Specific intake and exhaust openings should be used. The air should be ducted to the fan or blower intake as a minimum.

d. The air flow should be across and among the heat sources. Too often the air flow is not directed and ducted over the chassis and heat sources and flows through the path of least resistance which is usually next to the cabinet wall along the side of the chassis. Too many shipboard equipments have forced air cooling systems that are inadequate because the air blast does not come in contact with the individual heat sources. The air bypasses the parts it should be cooling.

e. As a minimum, screens should be placed over the intake and exhaust openings to prevent the entrance of foreign objects.

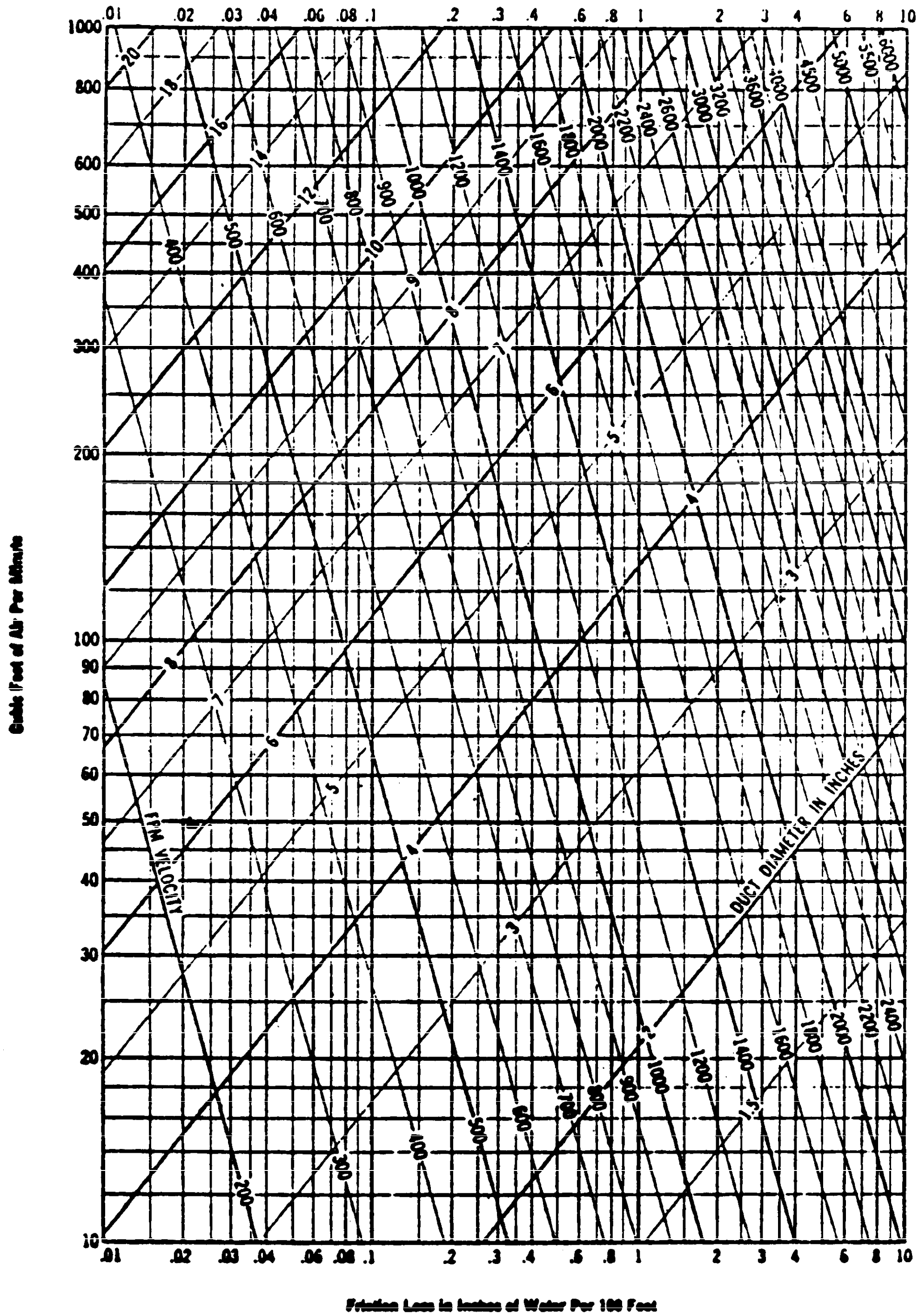


FIGURE 62. Air Friction Chart

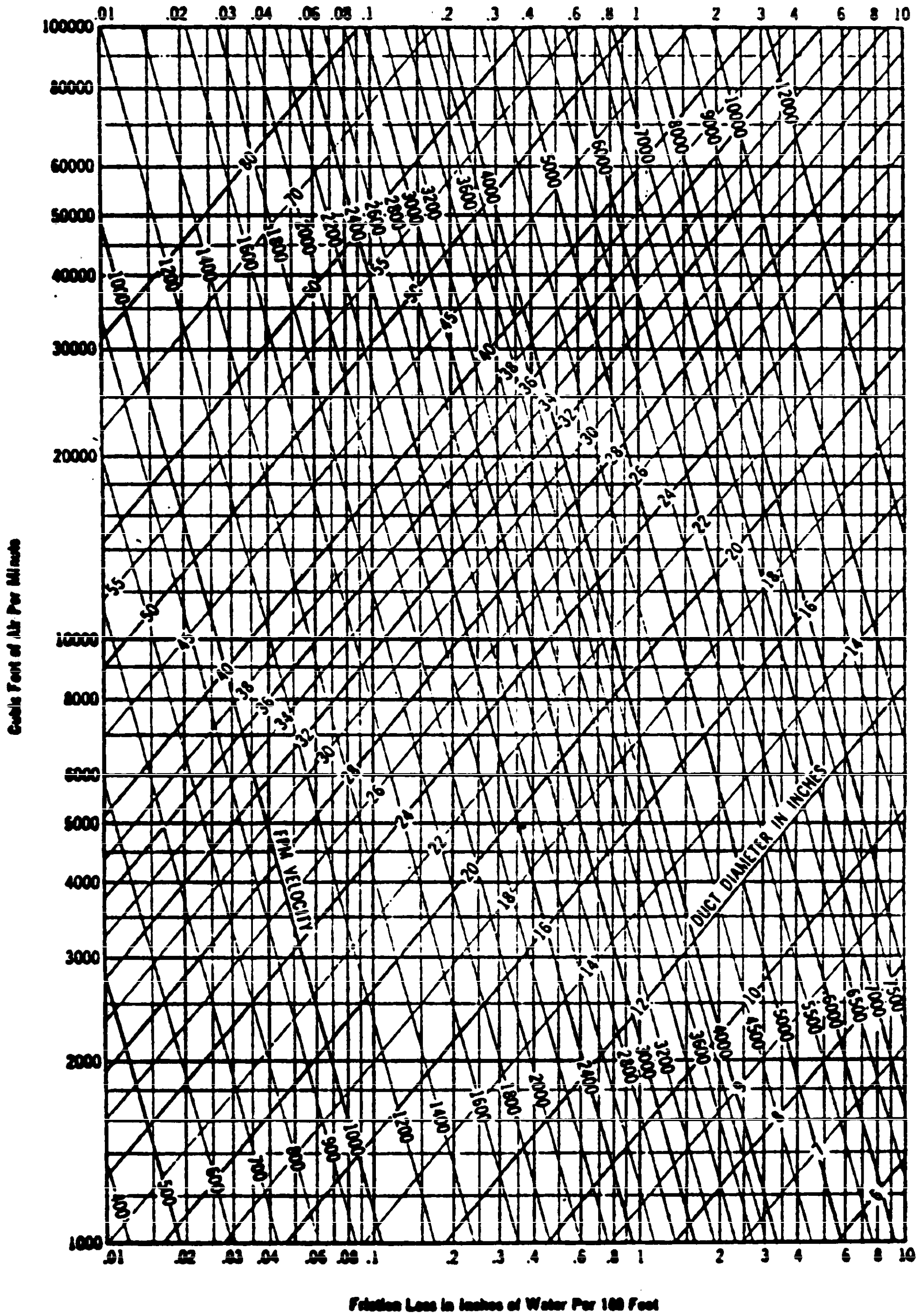


FIGURE 63. Air Friction Chart

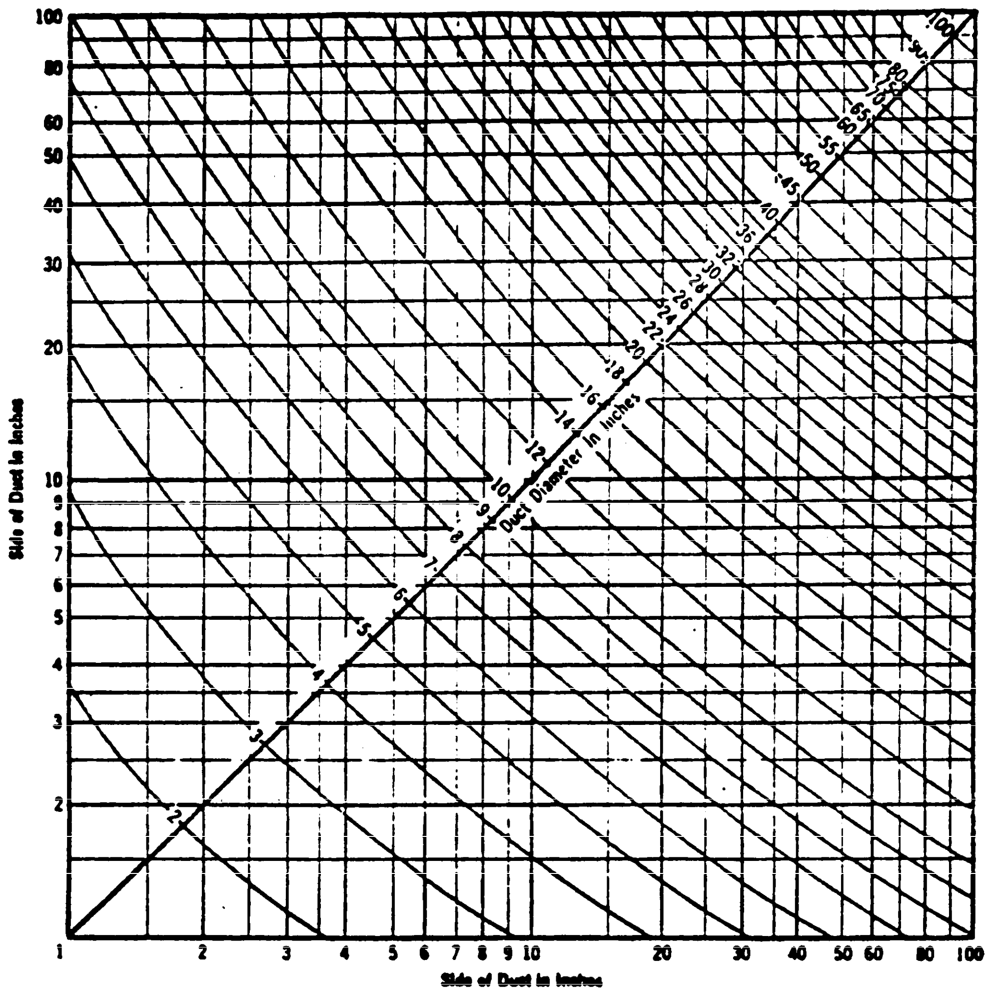


FIGURE 64. Conversion Chart for Converting Square or Rectangular Ducts to Round Ducts

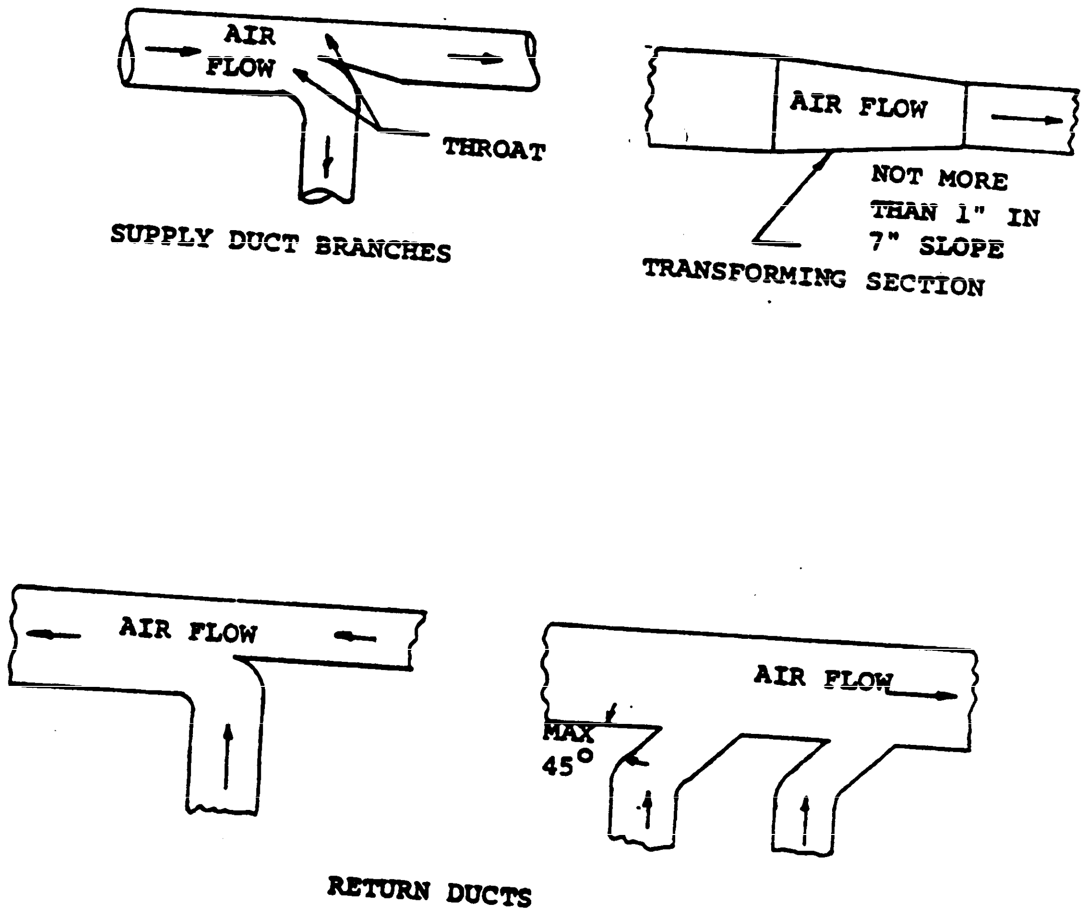


FIGURE 65. Some Fittings for Ducts

f. Air filters are usually desirable. See section 9.10.

g. The electronics should be on the downstream side of the blower. It is better to blow than suck!

9.6.2 Enclosed cabinets. Cabinets with closed loop forced air cooling systems almost always incorporate a heat exchanger for exchanging the heat transferred to the circulating air system to another fluid. On shipboard, air to fresh water cooled heat exchangers are commonly used. Such systems are discussed in detail in chapter 10. In aircraft, air to air exchangers are often used. The external air is often contaminated with excessive moisture and cannot be used for direct forced air cooling of the electronic parts.

The air flow in closed loop internal forced air to liquid coolant systems should be ducted throughout its entire flow circuit. The air cannot be scattered and be permitted to drift around inside the cabinet.

Closed loop internal forced air systems are excellent and offer the advantage of isolation from the external environment. Also, on shipboard, this prevents the overheating of the space in which the equipment is installed and eases the load on the air conditioning systems which are usually designed to handle only the personnel comfort load. On the other hand, one disadvantage is that the equipment cannot be operated without the external coolant. On shipboard provision can be made for operation during emergency conditoins by providing air inlet and exhaust doors that can be opened to permit the circulation of cabin air.

Closed loop internal forced air cooling systems can be designed for cooling with ram air or "ambient air." The fan or blower for the external air systems using "ambient air" must have constant mass flow rates regardless of air density. A number of successful systems have been built. Usually, these systems cannot be used in high speed high altitude aircraft. Section 9.9 discusses this in detail.

9.7 Ventilation and air conditioning

9.7.1 General. Adequate design of ventilation and heat transfer systems for spaces and shelters housing electronic equipment must be based on a large number of factors. The rate of heat transfer through the compartment surface should be estimated by methods described in heating and air conditioning manuals. This depends chiefly on the type of construction, the outside and inside temperatures, wind direction and velocity, and exposure to sunshine. This natural heat transfer rate is to be added algebraically to the heat liberated by the equipment and personnel inside the shelter, the result being a measure of the problem.

Paragraph 4-8, Solar, Celestial, and Terrestrial Radiation, of Reference 8 gives a basic, clear exposition of the radiation environment for shelters. Figures 67 and 68, taken from Reference 8, are included here to show the type of concise information available. Not only direct sunlight, but radiation from the sky and from the earth are discussed. Unless the shelter is shaded from these sources of radiation, they must be taken into account by the designer.

If the shelter houses personnel, the design should ensure a reasonably comfortable climate on the average and at least an endurable climate at the extremes. Certain temperature-humidity ranges are desirable and a minimum ventilation of 10 cubic feet of fresh outside air per minute per man is necessary. The structure and the outside wind determine the rate of air leakage, which should be estimated and compared with the requirements. Most structures are far from hermetically sealed and leak enough air for safety, but louvers or roof ventilators are helpful in controlling the air flow.

The moisture content of the air is a very important factor. For human confort temperature and humidity are so related that the average of Fahrenheit temperature and per cent relative humidity is a useful index. As temperature rises, humidity should drop. The specific heat of water vapor is higher than that of dry air, so moist air is a better coolant for equipment.

Ambient conditions for an electronic equipment shelter may range from -54°C (arctic) to 57°C, with 90% humidity (tropical) or 10% humidity (desert). It is futile to formulate any general rules or specifications; each situation must be studied in detail to determine the optimum air conditioning system. As an example, consider a shelter 8 x 10 x 7 feet, of ordinary wood siding with plaster on 4 inch studs, with two windows each 2 x 3 feet, and one door 6 x 3 feet. A rough calculation gives 78 watts of heat conduction per °C through walls, roof, and floor. Air leakage into the building is roughly estimated as 900 cubic feet per hour with a 15 mile wind, 2200 cfh with a 30 mile wind. This air leakage causes an additional heat transfer of 9 to 22 watts per °C. The minimum air leakage is safe for two occupants.

In arctic conditions, -40°C and a 30 mile wind, this shelter would dissipate roughly 5500 watts with an inside air temperature of 15°C. In ropical conditions, 38°C outside and 27°C inside, it would receive 860 watts by conduction plus 400 watts from sunshine on the roof, a total of 1460 watts, including air leakage. Since a man dissipates about 100 watts, the heat produced by the occupants is usually negligible in electronic equipment shelters. Such a shelter might well contain equipment dissipating 5000 watts of power. In arctic conditions it would maintain an inside temperature of about 15°C with no auxiliary heat transfer apparatus, and the electronic equipment could be adequately cooled by recirculating the compartment air. In tropical conditions it would require nearly two tons of refrigerating capacity to maintain an inside temperature of 27°C. (One ton of refrigeration = 3520.)

Removing the 6460 watts of heat be ventilating with dry air at 15°C would require 925 cubic feet per minute to maintain an inside temperature of 27°C.

These figures are rough estimates, but they indicate the wide range of conditions met with in the thermal design of compartments and shelters.

This section will not discuss the selection of air conditioning equipment. However, it is recommended that the size of air conditioning units be limited to 7.5 ton capacity, which is capable of removing 25 KW of power dissipation. Multiples of 7.5 ton air conditioners should be used when the heat to be removed exceeds 25 KW, because each such unit consumes the

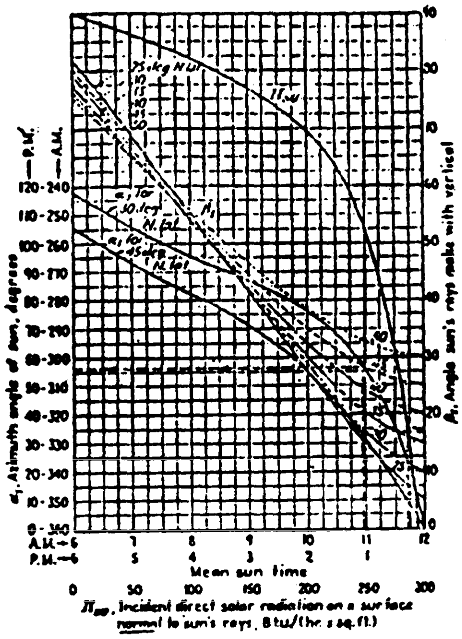


FIGURE 66. Direct Solar Radiation Received by a Surface Normal to the Sun and Solar Angles for the Period from May to August in Northern Latitudes

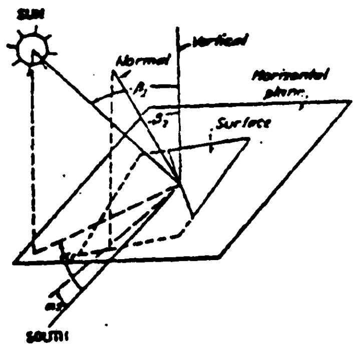


FIGURE 67. Definitions of Solar and Surface Angles

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entire output of a standard 30 KW engine generator. The heat dissipation to be used for ventilation system design should be the total electrical input power to the electronic equipment plus the energy consumed by the blower motors and other auxiliaries in the ventilation system. If the output of the electronic equipment exceeds 5 KW, the heat dissipation can be considered to be the difference between the electrical input and output.

If a shelter is designed specifically to house the equipment, then the ventilating and air conditioning duct work for both the shelter and equipment may be constructed as an integral system. If the equipment is to be placed in a shelter already equipped with ducts for ventilating the rooms, then certain modifications such as branch takeoffs to equipment, increased air delivery, and essential dampering to insure correct air distribution, will permit integration of the two systems. Substantial savings on installation and materials will be realized in both cases.

Under severe climatic conditions, special consideration should be given to the methods of ventilating and air conditioning the equipment and the shelter. In extremely cold climates, the shelter may be heated with a portion of the warm exhaust air from the equipment, the rest can be returned and mixed with the colder fresh air. Automatic control dampers may be used to regulate air temperature while mixing and eliminate the need for preheating the air.

9.7.2 Central ventilating system.

9.7.2.1 General. The ventilating system configuration depends on the size, complexity, temperature sensitivity, and heat concentration of the equipment, and on the thermal environment of the shelter. Electronic equipment ranges in size from complete weapons system equipment to simple ground based radio transceivers. In every case, the ventilating system must provide for the removal of waste heat from the electronic equipment without permitting excessive temperature at any point in the equipment. Ventilating systems therefore, vary widely in configuration and complexity. The system may include one or more of each of the following elements:

Fan and drive motor.
Means for air conditioning and dehumidifying.
Means for preheating, reheating, and tempering air.
Air cleaning equipment - filters.
Actuating and responding controls.

9.7.2.2 Year around system. These elements are separately selected to match specific requirements. Figure 68 shows a cross section view of a possible centralized ventilation and air conditioning system which can clean, heat or cool, and dehumidify the air for year around operation. Outside air may enter from the left at (A), desirably from an intake on the side of the shelter least exposed to solar heat, and not close to the ground or to a sunheated or dust-gathering roof. The damper (B) for proportioning the volume of outside air is interlocked with the return air damper (C) in such a manner that the mixing air temperature may be controlled. The return air duct (D), carries the warm air exhausted from the equipment and dumps all the excess outdoors. All the air it will be observed, must pass through the filters (E) and there should be ample room on both the inlet and outlet sides of the filters for servicing. Except in very warm climates, a heating or tempering coil (F) is required to warm the air unless it is possible to recirculate enough of the air so that the air drawn from outside, after mixing with the warm exhaust air will reach the proper temperature level.

The second group of heat transfer devices in a year around system includes an air cooling component (G) for use in warm weather. Its surface may be chilled by direct expansion of an approved refrigerant within its tubes, or the surface may be cooled by a pump-circulated liquid such as water or brine. The coil must be sufficiently cold to cool the air mixture to a temperature slightly below the existing dew point, in order to maintain constant relative humidity. Under extreme operating conditions the coil will be almost dry due to high sensible heat factors. A water tight drainage tank must be installed under the cooling coil and should extend some distance toward the fan. The second group of heat transfer devices also includes an air heating component similar to the tempering coil capable of warming the air leaving the cooling surface.

9.7.2.3 Heating and cooling exchangers. For service under severe arctic conditions, heating equipment, such as preheaters, reheaters, tempering coils, or booster heaters, may be used. Exchangers used for air cooling must have accompanying dehumidification. The cooling media normally used in surface coils is Freon 12.

The air to tube surface offers the largest resistance to heat transfer, generally reduced by the use of fins on the coils. Tube material for heating and cooling coils should generally be copper or copper alloys. The counterflow arrangement of tubes to air is almost universally used to take advantage of the highest possible mean temperature difference for given entering air and liquid temperatures. The proper selection of coils requires an understanding of each system (see paragraph 9.4.9).

9.7.3 Air duct design.

9.7.3.1 General. The principles and procedures for the design of air ducts have been presented in section 9.5, with particular reference to small ducts inside of equipment. These same principles and procedures are applicable to large ventilating ducts. There are, however, a few details which require additional attention.

9.7.3.2 Static regain method of design. In section 9.5, the equal friction method of duct design was recommended. For large ducts, the static regain method is recommended as being more precise and more theoretically correct. In this method, the velocity is reduced at each branch or takeoff so that the recovery in static pressure due to this reduction will exactly offset the friction in the succeeding section.

This method is theoretically sound, since it meets the requirements of maintaining uniform static pressure at all branches and outlets. The method provides a convenient means of designing a long run of duct having several takeoffs in that essentially the same static pressure exists at the entrance of each branch. For this type of application, little or no dampering is required. However, each outlet should be equipped with means of regulating air volume.

The electronic equipment to be cooled should be arranged in rows of units such that the overhead ducting delivering the air will be arranged fairly symmetrically. The branches and outlets to each unit should be generally spaced at regular intervals. The equal friction method will lend itself well to the design of such a system.

9.7.3.2.1 Duct velocity. The main supply duct velocity should be limited to a maximum of 2000 FPM. This practical limit is determined by the required fan power and by the noise produced. Lower velocity will require a larger duct, and will be determined in part by the discharge aperture of the fan selected.

9.7.3.2.2 Pressure losses. Pressure losses are calculated by the procedure described in section 9.3. However, the ventilation of enclosures introduces some details which require special attention.

Figure 69 extends the usefulness of the standard air friction chart and simplifies the solution of problems involving various densities and viscosities. The data used are the same as for the other charts but the friction loss is expressed in the number of pipe diameters of duct length which would have unity velocity-head loss.

Above Reynolds numbers of 3000 to 4000, flow is definitely turbulent, but not necessarily completely so. In this transition zone, in which lie the majority of cases of air flow encountered in ventilation, the friction factor (f) depends on both the relative roughness of the pipe wall and on the Reynolds number (Re) according to the Colebrook function (equation 9-92). (Reference 55)

In the transition zone in which the fluid is turbulent but not completely turbulent, the Colebrook function expresses the relationship between the friction factor, pipe roughness, and Reynolds number as:

$$\sqrt{\frac{1}{f}} = -2 \log \left[\frac{\frac{E}{D}}{3.7} + \frac{2.51}{Re \sqrt{f}} \right] \tag{9-90}$$

$$= -2 \log \left[\frac{\frac{E}{D}}{3.7} + \frac{2.51\nu}{VD \sqrt{f}} \right]$$

where:

f = friction factor, dimensionless
E = pipe roughness, feet
D = pipe diameter, feet

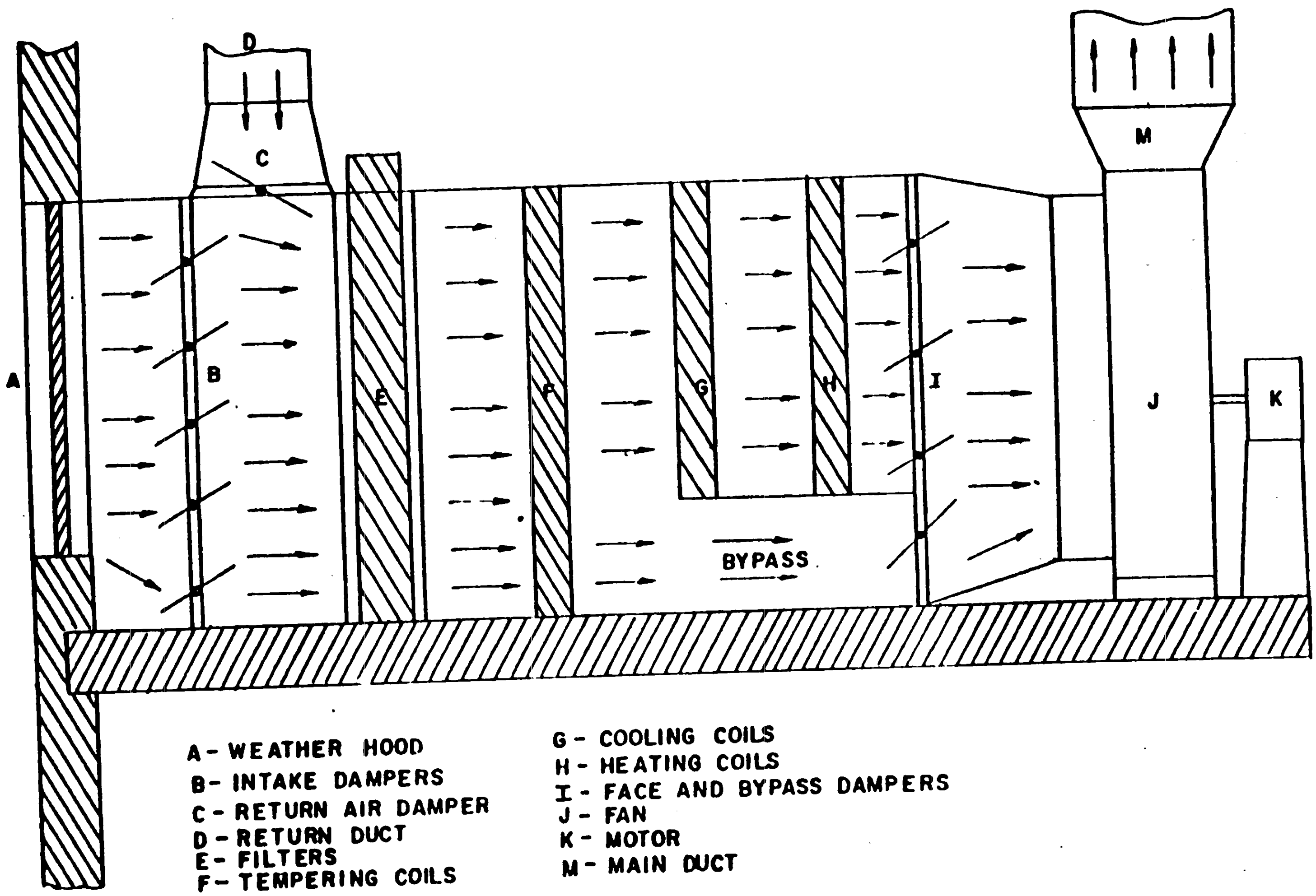


FIGURE 68. Central Arrangement of Equipment for Year Around Ventilating System

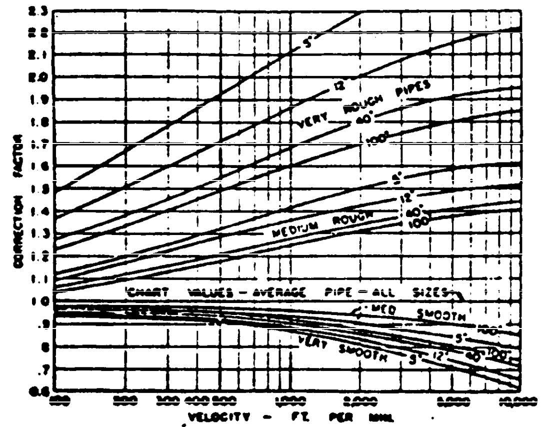


FIGURE 69. Air Friction Chart Extension

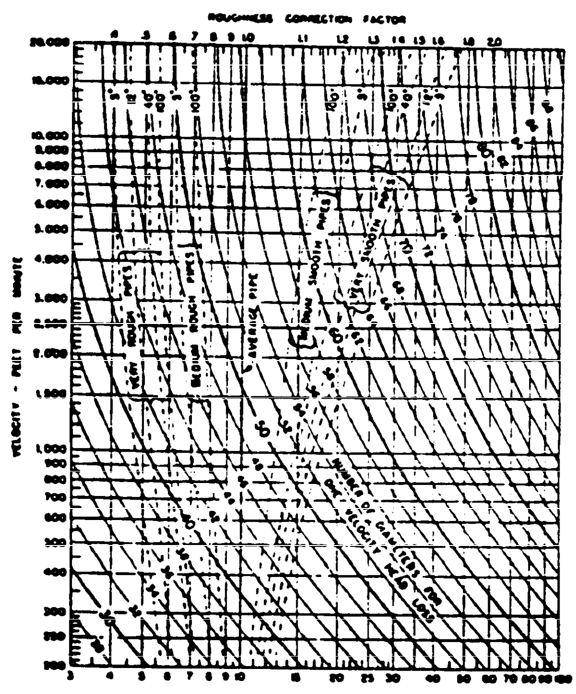


FIGURE 70. Correction Factor for Roughness

ν = kinematic viscosity, ft.^{2}/sec.
V = fluid velocity, ft./sec.
Re = $\frac{VD}{\nu}$ = Reynolds number, deminsionless

The air friction chart Figure 39 (Moody Diagram) includes as variables the pipe diameter, fluid velocity, friction loss, and capacity. In addition to these, two other variables should be included as indicated by the Colebrook function, namely, pipe roughness and fluid viscosity, the latter being included in the Reynolds number. Therefore, it is desirable to apply suitable correction factors to the friction charts based on standard air. No correction factors need be applied for temperatures between 15 and 27°C.

Correction for roughness can be made by use of Figure 70 which is also superimposed on Figure 69. Friction loss values obtained from Figure 39 should be multiplied by the correction factor obtained from Figure 70 for the given velocity, roughness, and diameter.

Equation 9-90 shows that a change in the fluid viscosity can be accounted for, merely by including an equivalent but reciprocal correction to the fluid velocity. It is preferable to apply this equivalent correction to the velocity rather than the diameter, as by doing so, only one term of the Colebrook function is involved and the procedure simplified. The use of the charts along with the standard air friction charts will be demonstrated in an illustrative problem.

Turning vanes may be advantageously employed in elbows, both to reduce the pressure loss and to provide a more uniform velocity distribution downstream from the bend.

9.7.4 Fan selection. The construction and characteristics of fans are discussed in section 9.2 of this handbook. Selection of the proper fan is a very important step in the design of the system. The following information is required to select the proper type and size of fan:

a. Capacity usually expressed in CFM
b. Total system resistance
c. Air density
d. Type of application or service
e. Arrangement of layout of the system
f. Type of motive power available

The fan selected for the duct system must not only produce the static pressure required to overcome the resistance of the duct work and the pressure drop through the electronic equipment, but also the additional pressure required by accessory equipment, such as heating and cooling coils and filters. Pressure losses of these components should be obtained from the manufacturers. Some of the possible resistances that a fan must overcome in delivering air are tabulateed as follows. These items will vary for each system.

Item	Possible Range of Loss ($in.H_2O$) Calc. & Mfgs. Data
Fan entry	0.005 to 0.1
Air heaters or coolers (one to several)	0.1 to 0.35
Air filters	0.2 to 0.4
Duct system	0.04 to 0.4
Screens, grilles, etc.	0.1 to 0.2
Less any regain	0.01 up
Total static pressure loss	1.0 to 1.6

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressure:

a. Volume of air (CFM) at operating conditions
b. Outlet velocity
c. Fan rotational speed
d. Horsepower
e. Tip speed
f. Static pressure

One especially important factor in selecting fans for ventilating systems is the efficiency, which affects the cost and noise of operation. A fan that is operated at a point considerably beyond maximum efficiency is usually noisy.

Fans of all types follow definite laws of performance. The fundamental law states that for a given airway system with constant air density:

a. The fan capacity varies directly as the speed.
b. The pressure varies as the square of the speed.
c. The power varies as the cube of the speed.

9.7.5 Example

9.7.5.1 Statement of Problem. Figure 71 presents a typical unit to be cooled. The diagram has a cut-away view showing a simple baffle system representing the internal resistances causing pressure losses. Assuming that the forced air cooling requirement of the equipment are known, the problem is to design the proper duct system to deliver and exhaust the air.

The conditions for the problem are as follows:

4 KW dissipation per unit
51.7°C (125°F) inlet air - (maximum condition)
65.5°C (150°F) exhaust air
14°C (25°F) temperature rise
2.0 in. H_2O pressure drop through equipment cabinet.

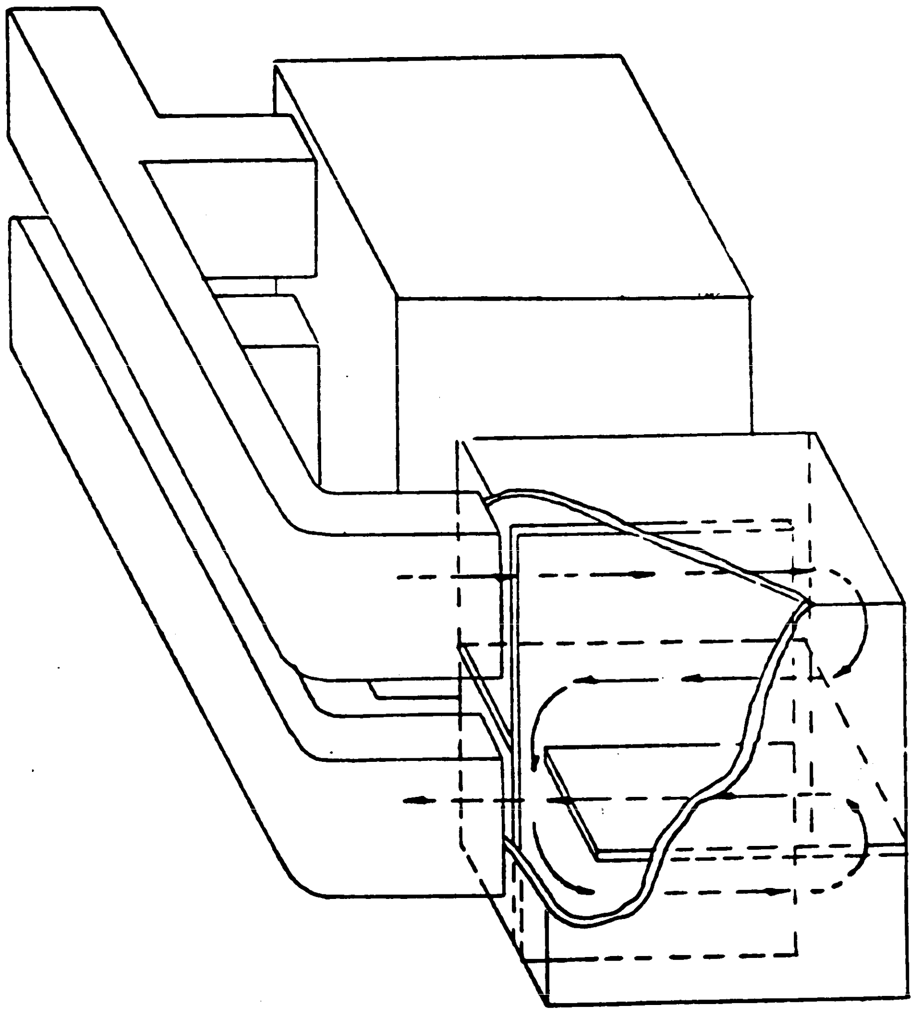


FIGURE 71. Electronic Units

The solution to the problem will involve finding the amount of air required per unit for the given air temperature rise, the size of ducts to deliver and exhaust the air, the inlet and outlet port sizes for the cabinets, and the selection of air conditioning equipment.

Figure 72 shows an arrangement of five units which may be only one run of many. The units can be arranged symetrically and connected to a main duct coming from a central location.

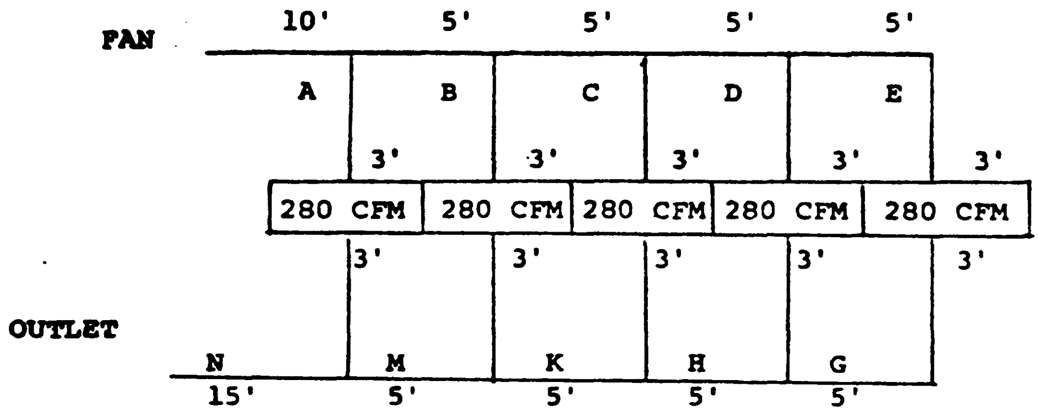


FIGURE 72. Arrangement of Five Heat Dissipating Units

9.7.5.2 Design Method. The equal friction method of duct design is most suitable for solving this problem. This method requires little balancing for symmetrical layouts in which all runs have about the same resistance. However, because the takeoffs are so short, a damper should be installed at each branch to regulate air volume.

9.7.5.3 Air Volume. First calculate the required air per unit. The density of air at (51.7°C) 125°F and at approximately atmospheric pressure is given by the following relation:

$$\rho = 1.5 \frac{P}{t + 273} \qquad (9\text{-}91)$$

where:

ρ = density, lbs./ft.^{3}

P = absolute pressure, lbs./in.^{2}

t = temperature, °C

$$\rho = \frac{1.5(14.7)}{51.7+273} = .068 \text{ lbs./ft.}^3$$

thus, one pound of air will occupy $\frac{1}{0.068}$ = 14.7 ft.^{3}

The specific heat for air at 125°F (constant pressure)

$$Cp = 7.62 \frac{\text{watt-min.}}{°C - \text{lbs.}}$$

The air leaving the units will be at 65.5°C (150°F) giving a 13.8°C temperature change. One pound of air will remove:

$(7.62)(13.8) = 105 \frac{\text{watt-min.}}{\text{lbs.}}$. The heat dissipated by each unit is 4 KW, so the mass of air required per unit will be:

$$\frac{4000}{105} = 38.1 \text{ lb./min.}$$

Since the air occupies 14.7 ft.^{3}/lb., the flow rate per unit will be:

$$(38.1)(14.7) = 560 \text{ CFM}$$

This is the volume delivered at 51.7°C, but the air is exhausted at 65.5°C and the density decreases to:

$$\rho = \frac{1.5\ (14.7)}{65.5 + 273} = 0.065 \text{ lb./ft.}^3$$

giving a flow rate of

$$\frac{0.065}{0.068} \times (560) - 585 \text{ CFM exhausted from each unit.}$$

9.7.5.4 Duct sizing.

(1) Delivery

Next, the required duct sizes and the inlet port to deliver the required air volume must be calculated. The usual procedure for the equal friction method is to select the main duct velocity to be consistant with good practice. The maximum allowable duct velocity has been limited to 2000 FPM at fan discharge. The total air to be delivered by the fan, neglecting duct leakage, is calculated as:

$$5(560) = 2800 \text{ CFM}$$

The required duct cross section will be:

$$A = \frac{Q}{V} = \frac{2800}{2000} = 1.4 \text{ ft.}^2 = 201 \text{ in.}^2$$

giving an equivalent duct diameter equal to 16.0 in.

A correction for viscosity change at 52°C (125°F) must be calculated before using the air friction charts. The kinematic viscosity of air at 51.7°C is 2.0×10^{-4} ft.^{2}/sec. and air at 21°C is 1.63×10^{-4} ft.^{2}/sec. giving a ratio of $\frac{2.0}{1.63} = 1.22$ (see American Society of Heating and Ventilating Engineers Handbook) (Reference 55). The equivalent velocity for use with Figure 69 is therefore, $2000 \times \frac{1}{1.22} = 1630$ FPM. The number of diameters for one velocity head is then obtained from Figure 69 at the intersection of the 1630 FPM velocity line and the 16.0 in. diameter line and is equal to 55.5 diams.

The velocity pressure varies as the square of the velocity and directly as the density. Since this density varies inversely as the absolute temperature, the velocity pressure will also vary inversely with the absolute temperature. The velocity to use in this calculation is the actual velocity, not the equivalent as figured above. In this case the velocity head will be:

$$\left(\frac{2000}{4005}\right)^2 \times \frac{460 + 70}{460 + 125} = 0.23 \text{ in. } H_2O$$

There is 0.23 in. H_2O pressure loss in 55.5 diameters or 0.23 in. H_2O loss in 74 ft. of 16 in. diameter duct. $\frac{100}{74} \times (0.23) = 0.31$ in. H_2O in 100 ft. of 16 in. diameter duct.

Now that the correction for temperature has been made, the standard air friction chart may be used (see Figure 39).

Using the 0.31 in. H_2O pressure loss line as a constant and finding the air volume through each succeeding section, the duct diameter, and velocities may be read from Figure 70. The ducts are converted from round to rectangular shape by use of Figure 64.

DUCT CONVERSION FOR SAMPLE PROBLEM

Section	Air Volume	Diameter	Velocity	Rect. Duct
A	2800 CFM	16.2 in.	1950 FPM	10 x 22 in.
B	2240 CFM	15.0 in.	1875 FPM	10 x 19 in.
C	1680 CFM	13.5 in.	1700 FPM	10 x 14 in.
D	1120 CFM	11.6 in.	1550 FPM	10 x 11 in.
E	560 CFM	8.8 in.	1300 FPM	10 x 8 in.

(2) Exhaust Duct Design

The exhausted air will be at a flow rate of 585 CFM from each unit because of the 14°C (25°F) temperature increase. This gives a total of 5(585) = 2925 CFM to be exhausted. The maximum velocity of 2000 FPM should be reached before mixing with the incoming fresh air. The duct area will have to be,

$A = \frac{Q}{V} = \frac{2925}{2000} = 1.46$ ft.^{2} or 211 in.^{2} giving an equivalent duct with a diameter of 16.4 in. A new correction factor for air at 65.5°C must be calculated before using the friction charts. The ratio of the kinematic viscosity of air at 65.5°C to air at 21°C is 1.2. The equivalent velocity for use with Figure is:

$$2000 \times \frac{1}{1.3} = 1540 \text{ FPM.}$$

The number of diameters for one velocity head is found on Figure equal to: 55.5 diameters.

The velocity head will be:

$$\left(\frac{2000}{4005}\right)^2 \times \frac{460 + 70}{460 + 150} = 0.22 \text{ in. } H_2O \text{ in 55.5 diameters or 0.22 in}$$

H_2O in 76 ft. of 16.4 in. diameter duct, equal to $\frac{100}{76}$ (0.22) = 0.29 in. H_2O in. 100 ft. Using the standard air friction chart, keeping the 0.29 in. H_2O loss line as a constant and using flow rates, the following are found:

Section	Air Flow Rate	Diameter	Velocity	Rect. Duct Size
N	2925 CFM	16.5 in.	1950 FPM	10 x 24 in.
M	2340 CFM	15.3 in.	1875 FPM	10 x 20 in.
K	1755 CFM	13.7 in.	1700 FPM	10 x 16 in.
H	1170 CFM	11.7 in.	1550 FPM	10 x 11.5 in.
G	585 CFM	9.0 in.	1300 FPM	10 x 8 in.

It is recommended that only one dimension of the duct size should be changed to minimize dynamic losses and to simplify construction. Also, at each takeoff from the main duct, the branch should slice into the main duct to help distribute the flow.

9.7.5.5 Pressure loss.

(1) Duct

Next calculate the pressure loss in the longest run of the system to find the total friction pressure the fan must work against to deliver the required air. Tracing the longest run on the schematic diamgram, A, B, C, D, E, G, H, K, M, N plus the loss in two elbows and the pressure drop through the cabinet constitute the total resistance. If the contractions and enlargements of the duct are gradual, no pressure loss will occur. We will assume the elbows in the duct will have radius ratios of R/W = 1.5. The duct size at the elbows is 8 in. x 10 in. The aspect ratio H/W is:

$$H/W = \frac{10}{8} = 1.25.$$

Therefore, from Table XIX the additional equivalent of duct in terms of the width L/W = 4.5,

so: $L = \frac{4.5(8)}{12} = 3$ ft. per elbow. The total length of duct having 0.31 in. H_2O per 100 ft. is made up of section A, B, C, D, E, plus one elbow and equal to: 10 + 4 (5) + 3 + 3 = 36 ft. Add 10% for a minimum of safety to get 39.6 ft. This compensates for any air leakage. The resistance is equal to $\frac{39.6}{100} \times 0.31 = 0.123$ in. H_2O. The exhaust section has a pressure loss of 0.29 in. H_2O per 100 ft. and is made up of sections G, H, K, M, N, and one elbow and equal to: 15 + 4 (5) + 3 + 3 = 41 ft. Adding 10% for air leakage total equivalent length = 451. ft. The resistance will be

$$\frac{45.1}{100} \times (0.29) = 0.13 \text{ in. } H_2O$$

(2) Equipment

In addition to this net static pressure loss in the duct system, the electronic unit adds another 2.0 in. H_2O loss plus the losses in accessory equipment. Computed or represented values for these losses will be tabulated. The first item is the static pressure loss in moving the air through the grill (intake) and louvers. The velocity in the trunk duct at the fan outlet is 1950 FPM. This loss expressed in decimal parts of one velocity head is 1.5 for entrance.

$$\left(\frac{1950}{4005}\right)^2 \times 1.5 = (0.238)(1.5) = 0.36 \text{ in. } H_2O$$

tempering heater (Mfg's data)	= 0.10 in. H_2O
filters (Mfg's data)	= 0.25 in. H_2O
dehumidifier (Mfg's data)	= 0.22 in. H_2O
reheater (Mfg's data)	= 0.10 in. H_2O
outlet grill (Mfg's data)	= 0.05 in. H_2O
net static pressure loss in duct, (computed before)	= 0.25 in. H_2O
pressure loss in each unit	= 2.0 in. H_2O
Total static pressure	= 2.97 in. H_2O

The fan selected must have a static pressure not less than 2.85 in. H_2O, when delivering 2800 CFM at 51.7°C. The total pressure of the fan, which takes into consideration the fact that air is inducted and accelerated to the fan outlet velocity, is obviously greater than 2.97 in. H_2O.

9.8 Forced air cooling and air conditioning systems in aircraft. This section comprises an introductory discussion of the cooling of electronic equipment on aircraft and related air conditioning systems. Much of the material presented has been taken directly from Reference 21, to which reference is made for further details.

It is assumed that the primary medium used for heat transport between the electronic package and the cooling equipment will be air. The air will either pass through a finned heat exchanger or cold plate designed into the electronic package or will, in some manner, be brought into thermal contact with the electronic parts.

9.8.1 Aircraft and missions. Many modern military aircraft are capable of flight at very high altitude and speed. This results in three cooling problems. First, the ambient air is of low density and therefore, has poor cooling effectiveness. Second, although the ambient air temperature at high altitude is low, its temperature is greatly increased when taken on board the aircraft, particularly for flight at supersonic speeds. While air of higher pressure and density may be obtained by bleeding air from the jet engine compressors, this air will be at a temperature much higher than that of ram air, due to the temperature rise in the compressor, which can not be less than that due to an ideal compressor. As will be discussed later, devices may be incorporated in the cooling system to reduce these air temperatures, although such devices impose a weight penalty on the aircraft. A third cooling problem arises from the fact that the aircraft surface temperature will be high at high Mach Numbers so that the electronic equipment will see high wall temperatures, unless mounted in cooled areas.

9.8.1.1 Ambient air conditions. The standard variation of air temperature and pressure with altitude is shown in Table XX, which is taken from Reference 51. A table using smaller altitude intervals is given in reference 21. The standard temperature profile is shown as the center line Figure 73. The temperature extremes applicable to military aircraft are defined by MIL-STD-210 as a Hot Atmosphere and a Cold Atmosphere. The hot and cold atmosphere temperatures are also shown in Figure 73.

Humidity is an important consideration in the design of cooling systems, particularly if the production of essentially dry air is required. For example, 0.022 lbs. of water per lb. of dry air, representing a dew point temperature of 26°C at sea level, is taken as the design vapor content up to the altitude (approximately 8000 ft.) where this becomes the moisture content of saturated air, the dry bulb temperature of which is that for the Hot Atmosphere of MIL-STD-210. Now the method usually employed for removal of water vapor is to condense the vapor by cooling and then remove the liquid. But, the condensation of 0.022 lbs. of water vapor requires a cooling capacity of approximately 0.022 x 1050 = 23 BTU per pound of dry air. The removal of this quantity of heat would cool one pound of dry air 53°C. Furthermore, a value of 0.043 lbs. of water per lb. of dry air, representing a dew point of 38°C at sea level, is a design maximum to be considered when determining the probable condensed moisture conditions in the cooling system.

At higher altitudes, up to 40,000 ft., the design vapor content is the saturation value corresponding to the altitude and the Hot Atmosphere temperature. Above 40,000 ft., where the Hot Atmosphere temperature starts to rise, the design vapor content is that of the specific humidity existing at the 40,000 ft. level, which is 0.0003 lbs. of water per lb. of dry air.

TABLE XX. U.S. Standard Atmosphere, [illegible]

Altitude ft.	Temperature °C	Pressure in Hg.	Pressure lb/in.^{2}
- 1,000	17.0	31.02	15.24
0	15.0	29.92	14.70
5,000	5.1	24.90	12.23
10,000	- 4.8	20.58	10.11
15,000	-14.7	16.89	8.30
20,000	-24.6	13.76	6.76
25,000	-34.5	11.12	5.46
30,000	-44.4	8.90	4.37
35,000	-54.2	7.06	3.47
37,000	-56.5	6.42	3.00
40,000	-56.5	5.56	2.73
45,000	-56.5	4.38	2.15
50,000	-56.5	3.44	1.69
55,000	-56.5	2.71	1.33
60,000	-56.5	2.14	1.05
65,000	-56.5	1.68	0.83
70,000	-55.2	1.33	0.65
75,000	-53.7	1.05	0.51
80,000	-52.2	0.83	0.41
85,000	-50.7	0.66	0.32
90,000	-49.2	0.52	0.26
95,000	-47.7	0.41	0.20
100,000	-46.2	0.33	0.16

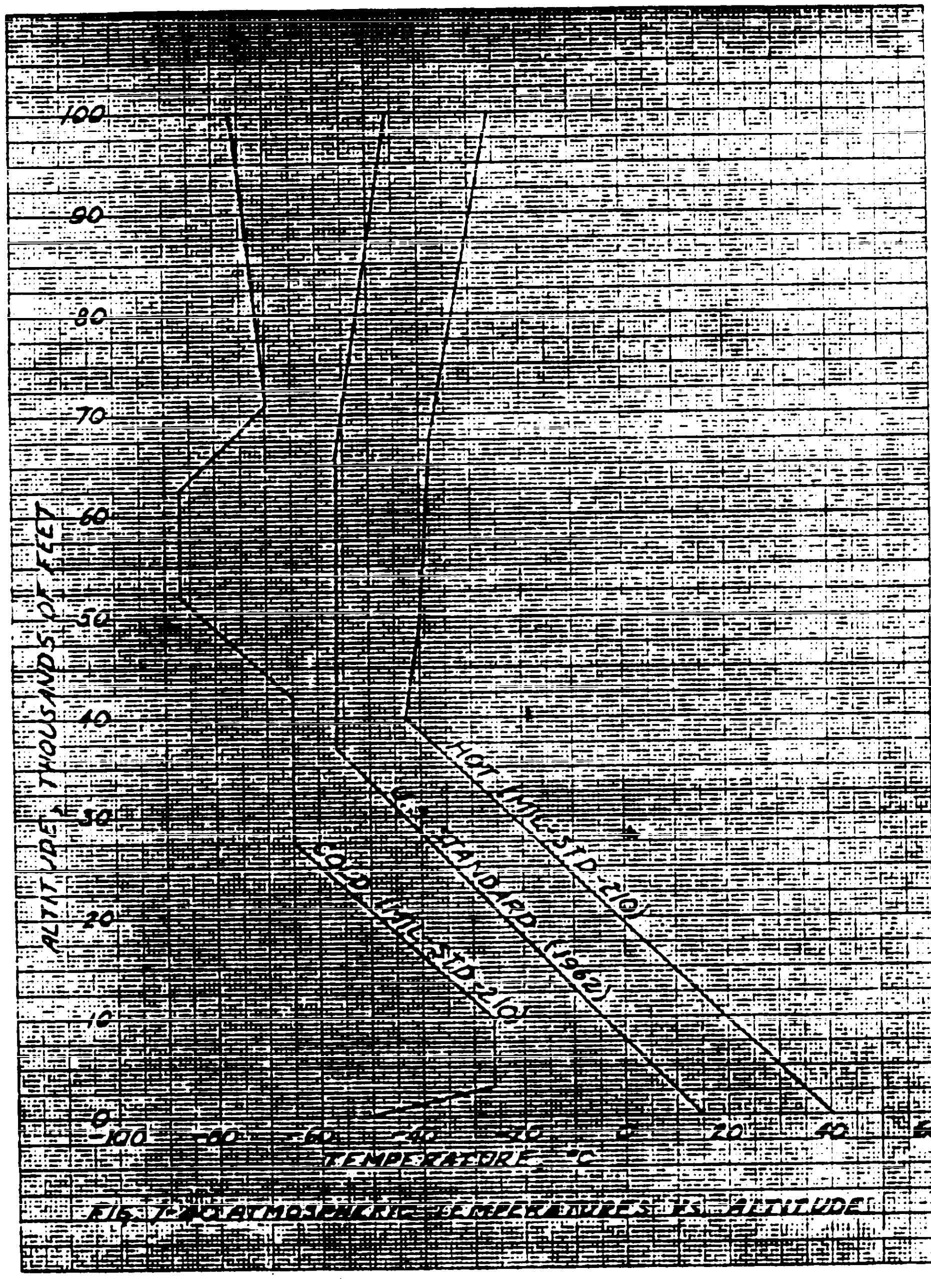


FIGURE 73. Atmospheric Temperatures vs. Altitude

9.8.1.2 Temperature of available cooling air. The ambient air has a high velocity relative to the aircraft, but when brought on board for cooling purposes this velocity is reduced to a relatively low value. When taken aboard by a forward facing ram intake, the temperature of the air will rise by an amount close to that given by equation 9-92, which is the theoretical equation for bringing the air to rest by adiabatic compression. (Reference 21)

$$\Delta t_r = 0.2M^2T_a \tag{9-92}$$

Where:

Δt_r = ram temperature rise, °C

M = Mach Number, the ratio of the airplane speed to the speed of sound at the ambient air temperature

T_a = absolute ambient air temperature, °K

An efficient ram intake will produce a considerable pressure rise as well as a temperature rise. Any other type of intake will produce significant temperature rises without advantage of maximum pressure rise.

From equation 9-92 we have

$$t_r = t_a + 0.2M^2T_a \tag{9-93}$$

Where:

t_r = ram temperature, °C

t_a = ambient temperature, °C

If, instead of taking air on board, a heat exchanger is mounted on the surface of the aircraft, the air in immediate contact with the surface will be brought to rest by skin friction. As a result, this air will also rise in temperature. The rise would be the same as that given by equation 9-92, except that there will be a transfer of heat across the boundary layer next to the aircraft to the colder ambient air. The actual rise will be Δt_r from equation 9-92 multiplied by a "recovery factor." But a representative value for the recovery factor is 0.85 (Reference 52), so that the temperature rise approximates that for ram air.

Figure 73 shows that the temperature of the MIL-STD-210 Hot Atmosphere is close to -40°C between the altitudes of 40,000 and 70,000 ft. Using that ambient temperature, the ram temperature has been computed from equation 9-93 and plotted on Figure 74.

9.8.1.3 Aircraft surface temperature. Assuming no other sources of heat input or output, the surface temperature of the aircraft will be equal to that of the air in immediate contact with that surface, so that

$$t_s = t_a + 0.2\, r\, M^2T_a \tag{9-94}$$

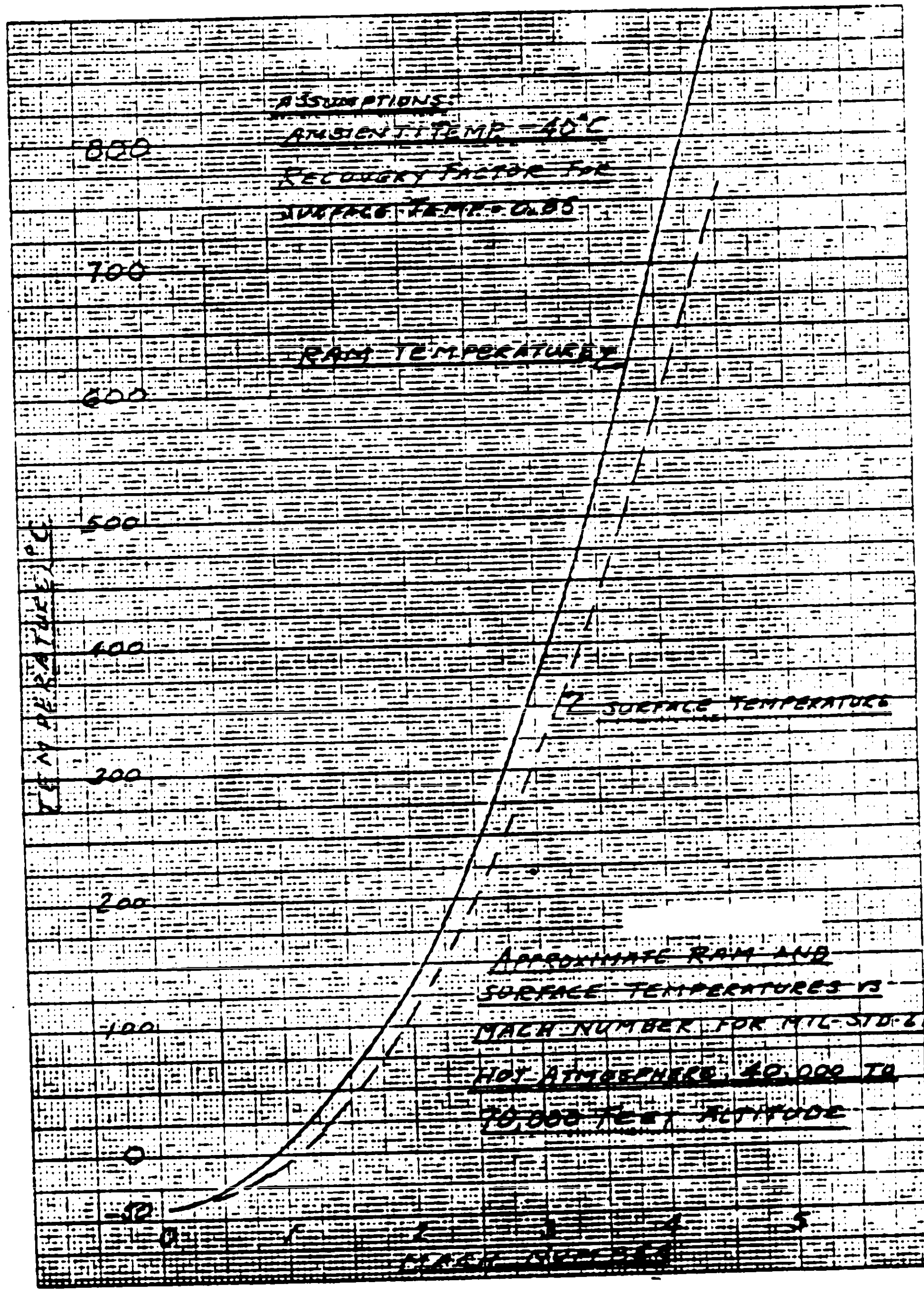


FIGURE 74. Approximate Ram and Surface Temperatures vs. Mach Number for MIL-STD-210 Hot Atmosphere, 40,000 to 70,000 Feet Altitude

Where:
t_s = surface temperature, °C
r = recovery factor (typically 0.85)

Actually, there will be heat inputs due to solar radiation and to heat dissipation inside the aircraft, and there will be heat output due to radiation away from the aircraft, a significant factor when the surface temperature becomes very high. There will, of course, also be heat transfer paths between aircraft surface elements due to conduction and internal radiative heat exchange. However, the approximate surface temperature may be computed from equation 9-94, taking r = 0.85. This has been done for an ambient temperature of -40°C, and the results have been plotted on Figure 74.

9.8.1.4 Aircraft operational envelope. The operating pattern or operational envelope of an airplane determines the overall cooling problem. Basic information needed includes:

a. Mission profiles showing altitude as a function of flight duration. Each profile will be similar to the upper (solid) curve of Figure 75.

b. Mission profiles showing flight Mach Numbers as a function of flight duration, as shown in the lower (dotted) curve of Figure 75.

c. Mission profiles showing altitude vs. Mach Number, as shown (for a different mission) in Figure 76. In most airplanes, the cooling system design point is at the airplane maximum Mach Number (maximum ram air and skin temperature) and maximum altitude (lowest density air).

In addition to meeting flight requirements provision must be made for cooling the electronic equipment during ground operation. MIL-E-5400 specifies ground operating requirements for equipment falling into various classes.

9.8.2 Ram air. Ram air is obtained by taking air on board an airplane in flight through a suitably designed forward facing intake or scoop. In practice, the ram temperature rise and the ram air temperature will be close to the theoretical values given by equations 9-92 and 9-93, respectively.

The pressure rise, for 100% efficient adiabatic compression, will be,

$$\Delta P_r = P_a(1 + 0.2M^2)^{3.5} - P_a \qquad (9\text{-}95)$$

Where:
ΔP_r = ram pressure rise, in. Hg
P_a = absolute pressure of ambient air, in. Hg
M = flight Mach Number

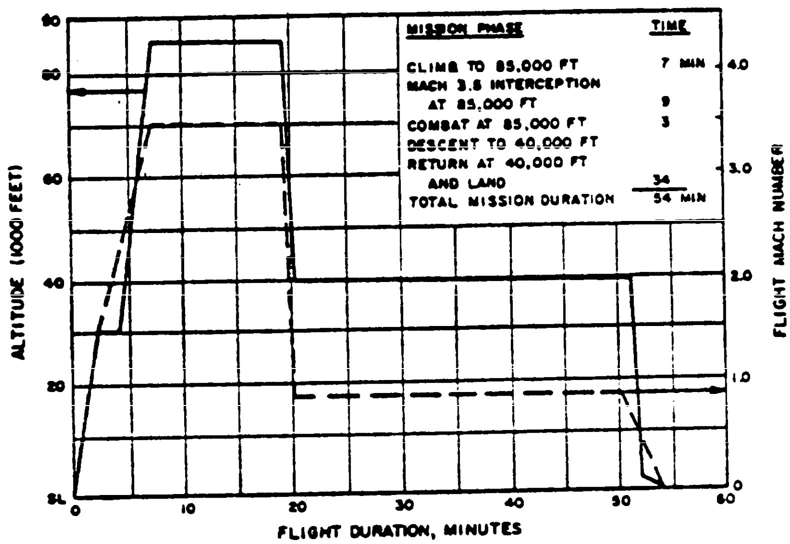


FIGURE 75. Hypothetical Mission Profile Mach 3.5 Fighter-Interceptor
(From Reference 21)

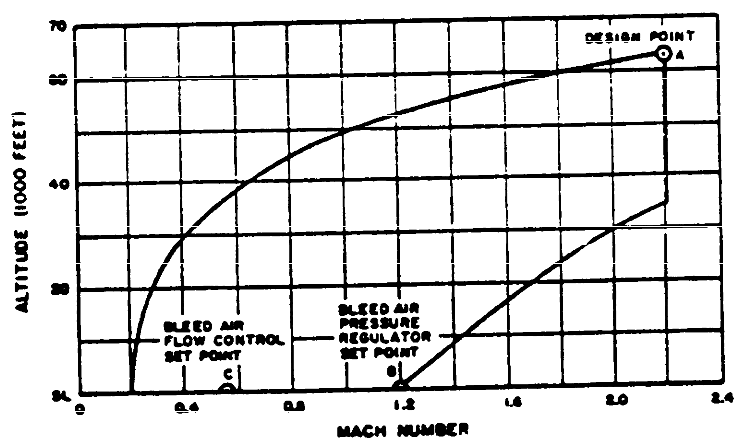


FIGURE 76. Flight Mission Profile, Altitude vs. Mach Number
(From Reference 21)

In practice, the recoverable ram pressure rise will be 50% to 75% of that computed by equation 9-95. The ram pressure rise may be converted to inches of water by multiplying by 13.6, the specific gravity of mercury.

Table XXI lists ram air temperature and ram air pressure rise for some representative values of altitude and Mach Number, for the MIL-STD-210 Hot Atmospheres. The ram air temperature is the theoretical value based on equation 9-93, which is close to that obtained in practice. The ram air pressure rise is taken as 70% of that given by equation 9-95 for adiabatic compression. It will be seen that temperature and ram pressure rise vary widely for different altitudes and Mach Numbers.

For actual airplane application, there occasionally must be a limit on the amount of ram air available under certain flight conditions. Also, the ram air discharge point may be located at a point on the airplane where the static pressure is less than at the ram inlet. Such a pressure differential may add to the pressure drop available for forcing air through the equipment.

An obvious limitation on the use of ram air for cooling is the fact that it is not available for ground cooling.

TABLE XXI. Ram Air Characteristics in MIL-STD-210 Hot Atmosphere

	Ram Air		
	Temperature °C	Pressure Rise* in. Hg	in. H_2O
Sea Level			
M = 0.6	63	5.7	77
M = 1.4	164	45.6	620
M = 2.2	344	203	2760
40,000 Ft. Altitude			
M = 0.6	-26	1.1	15
M = 1.4	47	8.5	116
M = 2.2	180	37.7	513
80,000 Ft. Altitude			
M = 0.6	-16	0.16	2
M = 1.4	61	1.3	17
M = 2.2	199	5.6	76

*Taken as 70% of the pressure rise for adiabatic compression.

9.8.3 Jet engine bleed air. Bleed air is available from the compression of jet engines. The three characteristics of interest are temperature, pressure, and available flow rate. The values all vary with airplane altitude and must be determined from the jet engine characteristics. If ground cooling is to depend on bleed air, then the ground level characteristics of the engine with respect to bleed air must be known. Both temperature and pressure are much higher for bleed air than for ram air. To be useful for cooling, the temperature of the bleed air must be greatly reduced. A large reduction can be accomplished by passing the bleed air through a heat exchanger cooled by ram air, but this will still leave the bleed air at too high a temperature. Devices for cooling to lower temperatures are discussed in the following paragraphs.

9.8.4 Water boilers. An effective method for reducing cooling air temperature is to pass the air through a water boiler, taking advantage of the high heat of vaporization of water. At high altitude, water boils at reduced temperature, making possible a lower air-out temperature from the boiler. Of course, sufficient water must be carried for the flight mission, which incurs a weight penalty, since the water vapor must be expended into the atmosphere.

9.8.5 Turbines, compressors, and fans. Bleed air is normally at a pressure which is much higher than that required to force air through the electronic equipment or to supply cooling air to the cabin. By expanding this air through a turbine, its temperature may be greatly reduced. However, in order to extract heat, mechanical work must be performed by the air, which means that the turbine must drive a load.

The usual method for loading the turbine is to use it to drive a fan or compressor, located so as to induce air flow in some part of the cooling system. There is no sharp line of demarcation between a fan and a compressor, but the air pressure ratio (outlet to inlet) is greater in the compressor. Generally, a compressor will have a pressure ratio of 1.05 or higher.

9.8.6 Water separators. As a result of air temperature drop, the turbine discharge temperature may be below the dew point, condensing water vapor in the air and producing entrained moisture in the air stream. Several types of water separators are available for removing this moisture, but the centrifugal type is most commonly used in aircraft systems. The entrained water droplets first coalesce to a size sufficient to respond to removal by centrifugal means. A swirling motion is then imparted to the air by rigid vanes, throwing the water droplets toward the outside. A collector, or eliminator, then diverts these droplets to a drain where they flow overboard. It is of great importance that water separators be protected from freezing temperatures, since ice buildup causes high pressure drop in the separator which tends to reduce air flow. References 71 and 72 are recommended for aircraft applications.

9.8.7 Air cycle systems. An air cycle system is basically an arrangement of components which provides cooling air at suitable temperature and pressure. In addition to cooling electronic equipment, such a system may provide cooling air for the cabin or for pressure suits. Many different arrangements are used, a few of which are discussed below.

9.8.7.1 Simple air cycle system. Figure 77 shows a schematic diagram of a simple air cycle cooling system used when high pressure bleed air is available from the engine compressor, or high pressure air from another pressure source.

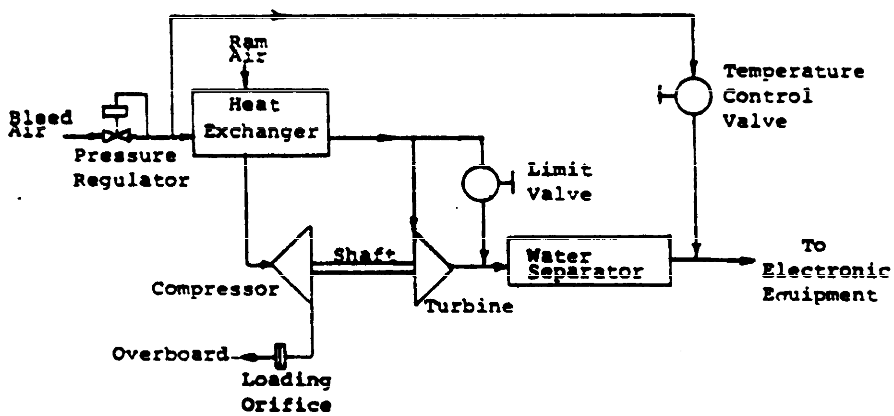


FIGURE 77. Simple Air Cycle Cooling System
(From Reference 21)

In this system jet engine bleed air is first cooled by passing it through a heat exchanger where it gives up heat to the cooler ram air. Further, cooling is accomplished by expansion through a turbine coupled to a compressor. The compressor loads the turbine and aids in exhausting the ram air, through a loading orifice which serves to balance the system. The air discharged by the turbine contains entrained moisture due to condensation resulting from the large temperature drop which has occurred. The water separator serves to remove this moisture before the air is directed to the electronic equipment. Various controls are incorporated to insure optimum operation at various altitudes and flight speeds. It will be noted that the temperature control valve shown in the diagram can permit some of the bleed air to bypass the cooling system, in order to prevent excessively low temperatures in the air fed to the electronic equipment.

9.8.7.2 Air cycle system with water boiler. The simple air cycle system shown in Figure 77 often provides sufficient cooling if the aircraft Mach Number is less than 2.0. However, if lower temperature cooling air is desired, or if the flight Mach Number is greater than 2.0, it becomes desirable to insert a boiler into the circuit between the heat exchanger and the turbine. Because of its high latent heat of vaporization (1037 BTU/lb. at 38°C) water is the best fluid to use in the boiler from the standpoint of minimum weight of expendable fluid required. But the possibility of damage from freezing, as when the aircraft is parked in cold weather, may lead to the use of water-ethyl alcohol, water-methyl alcohol, or water-ethylene glycol solutions. The latent heats of vaporization for these solutions are 584, 590, and 386 BTU/lb., respectively, at 38°C, for -54°C freezing temperature.

At flight Mach Numbers of the order of 3.0 or higher, the use of fuel as a supplemental heat sink is indicated.

9.8.7.3 Bootstrap air cycle system. The "bootstrap" type of air cycle cooling system, diagrammed in Figure 78, may give improved performance when the available bleed air pressure is relatively low. After passing through the ram air cooled primary heat exchanger, the bleed air pressure is boosted by the compressor. The bleed air then passes through a secondary heat exchanger which is also ram air cooled, then through a water boiler and through the cooling turbine which drives the compressor. In effect, part of the heat energy in the bleed air is converted into pressure energy by the turbine-compressor combination. As before, the exhaust air from the turbine passes through a water separator before going to the electronic equipment. Sufficient controls, not shown in Figure 78, ar incorporated to obtain optimum performance at various altitudes and flight speeds.

9.8.7.4 Regenerative air cycle system. The regenerative type air cycle cooling system diagrammed in Figure 79, eliminates the need for ram air cooling and expendable coolant (as from a water boiler) at high Mach Numbers. Also, a closed loop cooling system can be used, which aids greatly in reducing the moisture problem for electronic equipment and eliminates the need for a water separator. The key element of the system is the regenerative heat exchanger, which serves to cool the hot compressor bleed air before it enters the expansion turbine, where further cooling takes place. (References 21 and 52)

9.8.8 Other cooling systems. Various sytems employing refrigerants such as ammonia or freon are effective in producing low temperature cooling air. Two of these systems are described below. A third system of a different type, operating on the cooling tower principle, is also described briefly.

9.8.8.1 Expendable ammonia system. An expendable ammonia cooling system is diagrammed in Figure 80. A primary advantage of ammonia for some applications, is its low boiling piont, -33°C at standard sea level pressure. The latent heat of vaporization, 478 BTU/lb. at 38°C, is relatively high for fluids other than water. Storage at rather high pressure is necessary to prevent evaporation. Ammonia is highly toxic, involves rather severe handling problems, and has undesirable chemical properties. Its use as an expendable fluid should be considered only for cases where very low temperature is needed. (Reference 52)

The system shown in Figure 80 is very simple. Cooling air leaving the electronic equipment compartment is blown into the boiler. The latent heat of vaporization required to boil the ammonia is extracted from the cooling air, and also from the ammonia itself. As a result, the ammonia leaves the boiler as low temperature vapor, and the air is cooled before being recirculated to the electronic equipment. In a typical case, the cooling air might enter the electronic compartment at -20°C and leave at 60°C.

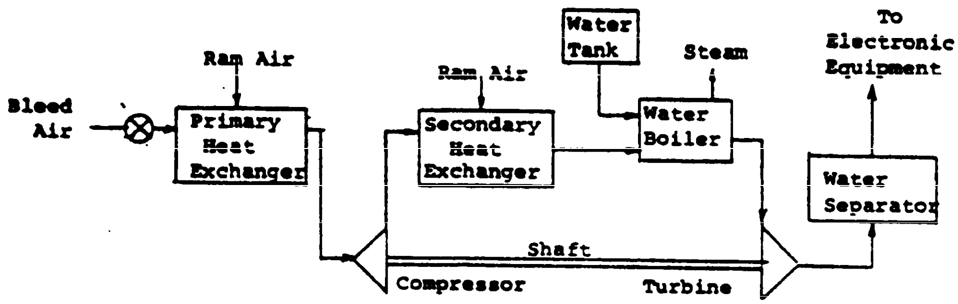


FIGURE 78. Bootstrap Air Cycle System
(From Reference 21)

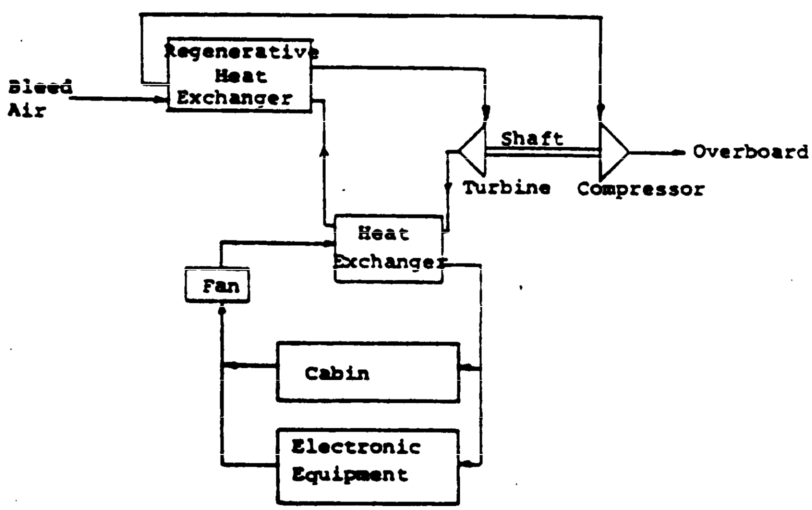


FIGURE 79. Regenerative Air Cycle System
(From Reference 21)

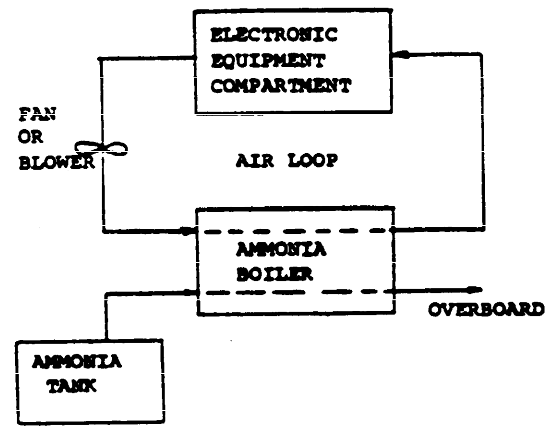


FIGURE 80. Expendable Ammonia Cooling System
(From Reference 21)

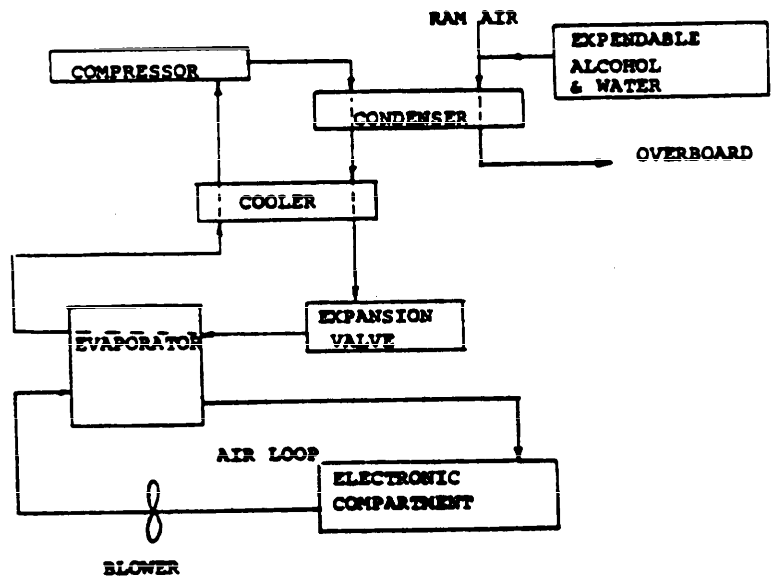


FIGURE 81. Freon Vapor Cycle System
(From Reference 21)

9.8.8.2 Freon vapor cycle system. One of the methods for cooling aircraft electronic equipment at high flight speeds is by means of a mechanical compression vapor cycle system. In this type of system, heat is transferred from the equipment by utilizing external energy to pump the heat to a sink at a higher temperature. The heat pumping fluid undergoes changes of state in going through the cycle. The cooling effect is obtained because of the heat required to change the fluid from the liquid to the vapor state. (Reference 52)

Figure 81 presents a diagram of such a system, using freon as the refrigerant. The essential parts of the system are:

An evaporator in which freon is evaporated at a lower temperature.

A blower which forces air in a closed loop system through the electronic equipment and then through the evaporator where the air temperature is reduced; typically, from 60°C to -20°C, as in the expendable ammonia system described above.

A compressor which removes freon vapor from the evaporator and compresses the vapor to a high pressure, greatly increasing its temperature.

A condenser, cooled by ram air, in which the high pressure freon vapor is changed to a liquid.

An expansion valve, in which the pressure of the freon liquid is greatly reduced. A portion of the liquid evaporates as the pressure drops, extracting the required heat of vaporization from the remaining liquid and from the vapor itself. The resulting low temperature mixture of liquid and a relatively small amount of vapor enters the evaporator where, ideally, all remaining liquid evaporates at constant temperature.

Two additional features are incorporated in Figure 81:

A cooler in the form of a heat exchanger extracts heat from the liquid freon leaving the condenser before it enters the expansion valve. This results in a lower freon temperature. The cooling effect is derived from freon leaving the evaporator, which rises in temperature before entering the compressor. Also, if the freon leaving the evaporator still contains any liquid, this liquid is evaporated in the cooler, contributing to the cooling effect on the liquid leaving the condenser.

For flight conditions producing particularly high ram air temperature, as when the airplane Mach Number is increased by activating a jet engine afterburner, expendable alcohol and water from a tank is injected into the ram air, to lower the ram air temperature by evaporating the liquid.

9.8.8.3 Evaporative spray "cooling tower" system. A type of aircraft electronic cooling system has been investigated (Reference 53) which is similar in operation to a cooling tower. The technique permits the use of low density high temperature air with a small amount of water as the cooling medium. Water in a fine spray from nozzles cools high temperature air as it enters a cooling chamber, the outside surface of which serves as a cold plate, for mounting electronic components. A rapidly moving film of liquid water flows over the inside of the chamber. A much smaller quantity of air is required than that which would be required for cooling by ram air alone.

9.8.9 Penalties. The addition of a cooling system to an aircraft influences aircraft performance by the imposition of additional weight, drag and power consupmtion.

Additional weight is added by the dead weight of the cooling equipment. Weight of expendable fluids also penalizes the aircraft, but to a less extent since, like the fuel, the fluids are consumed as the flight progresses.

Additional drag results from the momentum drag caused by taking ram air on board. That is, momentum relative to the airplane is taken out of the ram air in the process of bringing it on board, and this requires the continuous exertion of a forward force on the air, the reaction to which is an aft force, or drag, on the airplane. The momentum drag is directly proportional to the ram air flow rate, the Mach Number, and the square root of the absolute ram air temperature.

The extraction of shaft power from the engine to drive a fan, blower or compressor makes necessary an increase in fuel flow rate to maintain constant thrust. A larger, heavier engine will also be necessary to provide the same maximum thrust. Extracting bleed air has similar effects, since this process also decreases the power available from the engine.

The additional drag and power consumption imposed on an airplane by the cooling system can be expressed in terms of extra fuel load required to maintain the same mission characteristics as for a comparable airplane without the cooling system. The effect of the cooling system on aircraft performance can be based on range, endurance or rate of climb as the evaluation factor. To preserve the same flight characteristics, any increase in equipment weight or fuel weight must be matched by a corresponding increase in the size of the power plant to provide greater thrust. For the same structural integrity, the airframe must be of heavier construction to carry the additional load, and may also have to be larger in dimensions to maintain a suitable wing loading.

9.9 Design considerations for forced air cooled equipment.

9.9.1 General. The following ground rules are pertinent:

a. Minimize air flow noise in shipboard and ground based equipment by using slow speed blowers or fans and relatively large air ducts.

b. Protect fragile fins.

c. Design so that forced covection is aided by natural convection.

d. Make sure that all heat sources actually have the required air flow. See chapter 15.

e. If possible, protect against equipment damage in case of blower failure.

f. Make heat exchangers and air filters adequate and accessible for easy cleaning and replacement.

g. Make sure that air intakes and exhausts are far apart.

h. Make sure air flow paths are free and unobstructed and of proper size.

i. Avoid reuse of cooling air. If secondhand or series flow air must be used, the sequence of air passage over cooled parts must be carefully designed so that temperature sensitive parts are cooled first and that the air has sufficient thermal capacity to maintain required temperature for all parts.

j. With shipboard fresh water cooled closed loop forced air systems, it has been found that with typical equipment at least 25% of the dissipated heat is rejected by natural means to the environment and a maximum of 75% of heat is transferred into the water. The exception is with concentrated high power heat sources such as water cooled klystrons. In this case, most of the heat dissipated by the high power device will be transferred into the water.

k. Avionic equipment must be as lightweight and small as practicable. Consequently, the same constraints apply to the cooling system. Small high speed blowers or fans are mandated with ducts and air passages of minimum cross sectional area. This results in air pressures somewhat higher than in ground based equipment and the possibility of air leakage and air flow bypassing the desired path.

Care must be exercized in minimizing these undesirable effects.

9.9.2 Spot cooling. In some situations spot cooling of individual heat sources is required. Jet impingement cooling offers certain advantages for such applications. See section 9.4.8.4.

9.9.3 Moisture considerations, psychrometrics. It has been pointed out (in paragraph 5.1.4) that the condensation of moisture within electronic equipment can cause serious damage. On shipboard the operation of equipment at temperatures below the dewpoint of high humidity air must be avoided, and this has led to specifying that the temperature of fresh water used for cooling be not less than 40°C. In recently developed airborne equipment, the atmospheric air, which can be moisture laden, is frequently isolated from the electronics by the use of indirect forced air cooling techniques with internal air or cold plate type heat exchangers.

Psychrometric charts, found in various handbooks, show the moisture content of air, usually in grains per pound of dry air vs. dry bulb temperature for various values of the relative humidity. (one grain = 1/7000 lb. and dry bulb temperature is practically that obtained by inserting a thermometer in the air-vapor mixture) For a given temperature and relative humidity, the moisture content can be found. The dew point is then the temperature at which this moisture content produces 100% humidity.

Psychrometric charts are usually drawn for sea level pressure only, but charts for higher and lower pressures are given in Reference 21.

9.10 Air filters and cleaners. In the closed forced-air cooling systems used in electronic enclosures, external air cleanliness is not of the greatest importance, but internal air cleanliness is. In theory, if outside air leaks into the equipment, heavy dust particles affected by gravity might settle somewhere in the equipment and the finer particles would remain suspended in circulating air and be ejected from the surfaces of the electronic parts when the velocity exceeds the critical velocity of the dust particles. Unfortunately, such is not the case and "flug" forms causing a build up of a sludge like deposit throughout equipment (see page 21). The formation of "flug" on the surfaces of fan blades and impellers and heat sources can significantly reduce the effectiveness of a forced air cooling system. Consequently, cleanliness is of vital importance over the life of the equipment. Air filters are mandated, especially on shipboard, with either open or closed forced air systems.

The various filter types are:

9.10.1 Dry filters strain the air through orifices smaller than the dust particles. The screens are made of cellulose, cloth felt, or similar material. They depend upon the fineness of mesh to screen out the impurities. The effectiveness of dry filters is a function of the denseness and thickness of the filtering medium. Dry filtering has a high resistance to air flow.

9.10.2 Wet filters function through impact against viscous-coated barriers having interstices larger than the dust particles. Viscous filters usually use oil as a means of dust separation and can be very effective. However, once the dust is in contact with the oil it is difficult to release. This feature, while advantageous in the collection of dust, is somewhat of an inconvenience in its ultimate disposal.

9.10.3 There has been developed a dry filter which operates on the principle of electrostatic precipitation without the associated high voltage power supply, charged wire electrodes, and electronic equipment. A plastic material is given an electrical charge which re-orients the molecules to maintain a permanent electrostatic charge (electret). The charged plastic is then shredded and packed into a conventional filter unit. Dust particles are attracted to the charged plastic filter by the electric field. When dirty, the filters can be removed and washed with a fine spray of cool water which neutralizes the static charge and releases the dirt. When dry, the filters resume their static charge and may be used again. The shredded plastic material must be firmly held in place in the filter by crossties to prevent settling and dislodgement of the filter material during washing or transportation. Any significant displacement of the filter material will render the filter ineffective. The use of hot water for cleaning will destroy the electrical charge and render the filter ineffective.

9.10.4 It is necessary to keep all filters clean and free from excessive oil in order that dust and oil be kept from the parts to be cooled. As explained previously, the resulting "flug" can hinder the transfer of heat from the parts.

When dirty, filters may present a fire hazard in as much as the dirt and "flug" they accumulate may be flammable. The duct and connections to filters should be designed to allow uniform flow over the entire filter area.

9.10.5 All air filters should be mounted on or close to the front of the cabinets for accessibility. If they cannot be readily removed for inspection, cleaning, or replacement they will not be cleaned! Note that this applies to both open and closed forced air cooling systems, using internal or external air.

9.10.6 The air filter net opening for air flow will usually be reduced from 25 to 50% before the filter is changed. The overall air system including the blower should be designed for normal cooling with dirty filters. Thus, the actual pressure developed by blowers must be greater than that which a design with clean filters would require. Filter manufacturers provide pressure drop and flow data for filters in various conditions of cleaniness.

9.11 Maintenance considerations. The following maintenance considerations are important in the design of forced air cooled equipment:

9.11.1 The design should be such that when a cabinet drawer is withdrawn for test or maintenance, the forced air cooling system will continue to function. The point is that the parts can be severly overstressed and even the electrical performance changed if the cooling is lost when a drawer is withdrawn when the equipment is operating. Some deterioration in cooling air system performance may be tolerable, but a major change in air flow should not be permitted.

9.11.2 Design for maintenance accessibility! The removal of heat exchangers for cleaning for example, should be readily accomplished in a short period of time. Too often the removal of the heat exchanger at the rear of large cabinets has been a very expensive and time consuming process. As a result not only were equipment MTTRs excessive, but the exchangers were not cleaned as frequently as necessary and equipments overheated with subsequent poor reliability.

9.11.3 Design forced air cooling systems so that the cleaning of fan blades, impellers, and heat sources (especially fins) can be readily accomplished. Periodic cleaning is absolutely necessary to remove the buildup of flug and other foreign material. Air filters must be very accessible and readily changed.

9.11.4 Forced air cooling systems that are cooled with externally supplied air, such as airborne equipments, must be provided with specially designed cool air supplies for maintenance and testing on the ground or flight deck. These cool air supplies should as a minimum provide air flow identical to that supplied in the aircraft. Special test benches or cooling carts are usually supplied for this purpose. The point is that too often the special cooling supply has marginal or inadequate performance. Also, these special cooling air systems can degrade in performance due to the buildup of flug, especially on an aircraft carrier deck. The special cooling systems should also be designed for maintenance of their air flow systems.

10. THERMAL DESIGN OF LIQUID COOLED ELECTRONIC EQUIPMENT

10.1 Theory

10.1.1 General. Liquid cooling systems may be broadly classified as (1) direct liquid cooling, wherein the heat dissipating components are directly immersed in the cooling liquid, or (2) indirect liquid cooling, wherein the parts are not in the liquid and heat is removed from the parts by other methods of heat transfer, such as conduction, for subsequent transfer to the cooling liquid. In either case, for those portions of the thermal circuit involving liquid cooling, the effects of radiation are usually negligible, and heat transfer to and in the liquid occurs primarily by convection, and by conduction.

Liquid convection may be natural (convection currents are induced by variations in liquid density which in turn are generated by temperature differences) or forced (an external source, such as a pump, generates a pressure differential to cause circulation of the fluid).

In theory, the basic equations developed in chapter 8 for free convection and in chapter 9 for forced convection apply to any fluid, be it gaseous (air) or liquid. In reality, since air is so common a coolant, more general knowledge and empirical data have been accumulated for air cooled systems, and many of the relationships have been simplified based on the extensive knowledge of the properties of air. Because of the lack of information on the detailed properties of many of the potential liquid coolants, such simplified expressions generally do not exist for liquid cooling systems. Many formulae for convective heat transfer (either free or forced), found in the literature, should be used with caution. Very often, these formulae are based on empirical relationships derived from experimentation in a limited range of specific conditions. Unless these identical conditions exist, the formulae are not necessarily valid.

The following simple example, using thermal circuit terminology, illustrates the benefits of liquid cooling. (See Figures 82 & 83)

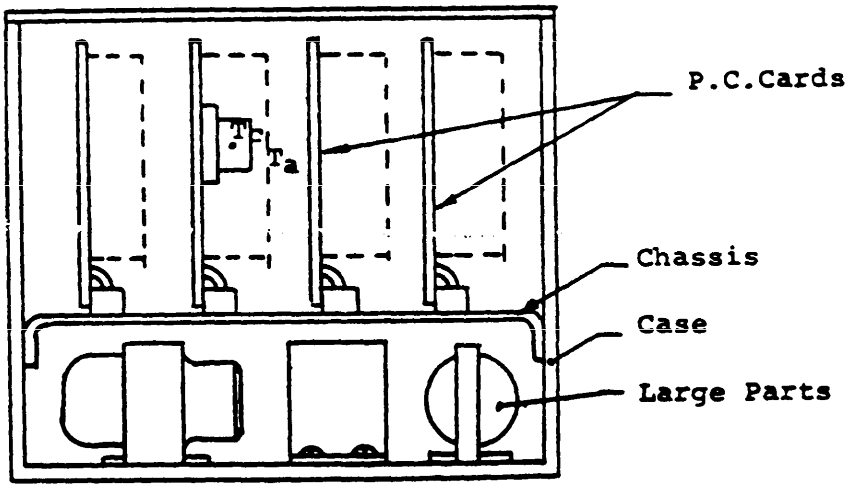


FIGURE 82. Crossection of Equipment

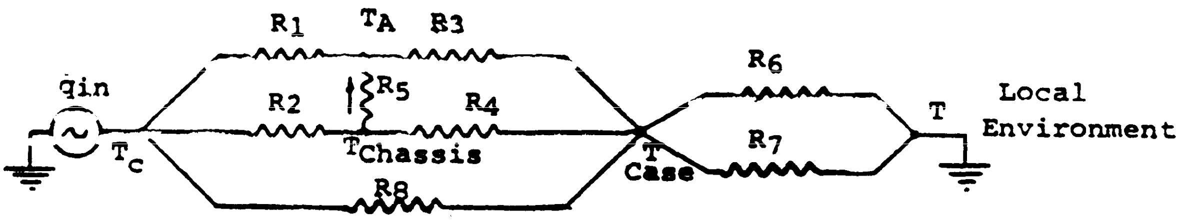


FIGURE 83. Equivalent Thermal Circuit

R_1 = Convection Resistance, Component-to-Internal Air
R_2 = Conduction Resistance, Component-to-Chassis
R_3 = Convection Resistance, Internal Air-to-Case
R_4 = Conduction and Interface Resistance, Chassis-to-Case
R_5 = Convection Resistance, Chassis-to-Internal Air
R_6 = Convection Resistance, Case-to-Environment
R_7 = Radiation Resistance, Case-to-Environment
R_8 = Radiation Resistance, Component-to-Case

For the air filled case, R_1, R_3, and R_5 are high. Normal design procedure is to minimize R_2 and R_4. R_6 and R_7 may become limiting resistances.

As heat flux density increased, or due to the need for electrical insulation or other construction constraints, the temperature rise established across R_2 and R_4 may become undesirably large. Assuming R_2 and R_4 (the conduction resistances) cannot be reduced further with any practical design, two approaches are possible. One is to reduce the external convection resistance, T_6, such that the total thermal resistance from T_{case} to T is again brought to a reasonably value. Addition of external fins to increase the convection area is the simplest method, but it is limited in applicability (chapter 8). External forced air cooling may be used (chapter 9) to drastically reduce R_6, but at the expense of a prime mover (blower) for the air.

The other approach is to reduce the internal convection resistance R_1 and R_3 (and to a lesser extent R_5). Direct liquid cooling (immersion) is capable of reducing these resistances by approximately an order of magnitude.

With direct liquid cooling within the module to reduce R_1 and R_3, the external convection-radiation thermal resistance once again becomes foremost in the total resistance path from the component to the ultimate sink. As previously mentioned, external forced air cooling can be used to reduce the external convection resistance. For high heat loads, liquid cooled cold plates are often more practical. This is indirect liquid cooling. (Note that a combination of direct liquid cooling to reduce the internal resistance (R_1+R_3) and indirect liquid cooling to reduce the external resistance (R_6) may co-exist in a given system).

If the unit heat concentrations and dissipations are such that liquid cooling methods prove inadequate, the designer should consider vaporization cooling techniques (chapter 11) or special cooling methods (chapter 12).

The basic equation for convection is:

$$q = h_c A_s \Delta t_s \qquad (10\text{-}1)$$

where:

q = the rate of heat transfer, watts
h_c = the coefficient of convective heat transfer, watts/in^2-°C
A_s = the surface area, in.^{2}
Δt_s = the temperature difference between the heated surface and the main fluid stream. °C

The value of the heat transfer coefficient, h_c is influenced by many factors, including not only the properties of the fluid, such as viscosity, density, etc., but the flow conditions, surface characteristics, and geometry of the parts as well. The parameters used in the determination of the heat transfer coefficient for free convection differ from those used to determine the coefficient for forced convection. When compared with gases, the convective heat transfer coefficients of liquids are generally very high. This may be partially offset in some applications by the fact that most liquids transmit very little radiation.

10.1.2 Free convection in liquids.

10.1.2.1 Correlation. Heat transfer coefficients in free convection are determined by three dimensionless parameters: the Nusselt, Prandtl, and Grashof numbers. The general relationship of the three parameters appears as follows:

$$Nu = c\,(Gr)^m\,(Pr)^n \qquad (10\text{-}2)$$

where:

$Nu = \dfrac{hD}{k}$ is the Nusselt number

$Pr = \dfrac{c_p \mu}{k}$ is the Prandtl number

$Gr = \dfrac{D^3 \rho^2 g \beta \Delta t}{\mu^2}$ is the Grashof number

and

h = the heat transfer coefficient, $\dfrac{\text{watts}}{\text{ft.}^2\text{-°C}}$

D = the characteristic dimension, ft.

k = the thermal conductivity of the coolant, $\dfrac{\text{watts-ft.}}{\text{ft.}^2\text{-°C}}$

c_p = the specific heat of the coolant, $\dfrac{\text{watt-min.}}{\text{lb.-°C}}$

μ = the absolute viscosity of the coolant, $\dfrac{\text{lb.}}{\text{ft.-min.}}$

ρ = the density of the coolant, $\dfrac{\text{lbs.}}{\text{ft.}^3}$

g = the gravitational acceleration, $\dfrac{\text{ft.}}{\text{min.}^2}$

β = the volumetric coefficient of expansion, $\dfrac{1}{°C}$

t = the difference in temperature between the heated surface and bulk of coolant, °C.

Note: In numerical calculations, all of these quantities must be expressed in a consistent system of units.

c, n, and m are constants dependent on the coolant properties, the configuration, and the temperature.

Generally, the exponents, n and m, are very nearly equal and, therefore, equation 10-2 can be presented as:

$$Nu = c\,(Pr\ Gr)^n \qquad (10\text{-}3)$$

When the dimensionless parameters are expressed in terms of the variables comprising them, the following equation results:

$$h = c\,\frac{k}{D}\left[\frac{D^3 \rho^2 g \beta \Delta t c_p}{\mu k}\right]^n \qquad (10\text{-}4)$$

Often equation 10-4 is written as:

$$h = c\,\frac{k}{D}\left[a D^3 \Delta t\right]^n \qquad (10\text{-}5)$$

where:

$a = \dfrac{\rho^2 g \beta c_p}{\mu k}$, the convection modulus

The significant dimension, D, to be used in the above equations is given in Table XXII. For irregularly shaped surfaces, the characteristic dimension can be taken as that of the most similar surface listed in the Table XXII. Table XXIII lists values to be used for the constant C.

TABLE XXII. Significant Dimension "D"

Surface	Position	D
Plane	horizontal	$\frac{(\text{Length}) \times (\text{Width})}{\text{Length} + \text{Width}}$
Plane (rectangular)	vertical	vertical height but limited to 2 ft.
Plane (non-rectangular)	vertical	$\frac{\text{area}}{\text{horizontal width}}$
Plane (circular)	vertical	0.785 x diameter
Cylinder	horizontal	diameter
Cylinder	vertical	height of cylinder but limited to 2 ft.
Sphere		radius

Usually the properties of the liquid under consideration are evaluated at the film temperature to give the film heat transfer coefficnet, which approaches the average value. Film temperature is defined as the arithmetic mean temperature between the average surface temperature and the average bulk temperature. Expressed mathematically,

$$t_f = \frac{t_s + t_b}{2}$$

where:

t_f = the film temperature
t_s = the average surface temperature
t_b = the average bulk temperature of the fluid

The exponent "n" in equation 10-5 is dependent on the value of $(aD^3\Delta t)$. If the value of $(aD^3\Delta t)$ lies between 10^3 and 10^9, "n" may be taken as 0.25. If $(aD^3\Delta t)$ exceeds 10^9, "n" may be taken as 0.33. Most free convection calculations for electronic equipment yield a value of $(aD^3\Delta t)$ between 10^3 and 10^9. Generally, this corresponds to a significant length, D, of about 2 inches for free convection in liquids, at the temperature normally encountered in electronic equipment.

TABLE XXIII. Values of the Constant C

Shape and Position	C
Vertical plates	0.55
Horizontal cylinders (pipes and wires)	0.45
Long vertical cylinders	0.45 - 0.55
Horizontal plates facing upward	0.71
Horizontal plates facing downward	0.35
Spheres	0.63
Small parts (see text)	1.45

10.1.2.2 Electronic parts. Reference 27 cites the following equation for free convection cooling of tubes, resistors, relays, and transformers when immersed in various fluids and placed in confined spaces:

$$Nu = 1.45\ (GrPr)^{.23} \qquad (10\text{-}6)$$

The high value of C in equation 10-6 is believed due to enclosure effects and the irregularity and smallness of the electronic parts. Equation 10-6 can also be written as:

$$\frac{hD}{k_f} = 1.45\left[a_f\ D^3 \Delta t\right]^{.23} \qquad (10\text{-}7)$$

When the amount of heat dissipated from electronic sources by convection is known, equations 10-1 and 10-7 after simplifying and rearranging the terms, will yield the following relationship:

$$\Delta t = .74D^{.252}\left(\frac{\lambda}{a_f}\right)^{.187}\left(\frac{q}{A_s k_f}\right)^{.813} \qquad (10\text{-}8)(D.E.)$$

where:

Δt = the temperature difference between heated surfaces and bulk of the fluid, °C.

D = the characteristic dimension, ft. (see Table XXII)

a_f = the convection modulus, "a", evaluated at the film temperature, $\frac{1}{ft.^3\text{-}°C}$

A_s = the surface area of the heat source, sq. ft.

k_f = the thermal conductivity of the liquid film, $\frac{watts\text{-}ft.}{ft.^2\text{-}°C}$

q = the heat dissipated in the fluid, watts

Equation 10-8 is presented in nomograph form in Figure 84, (q/A_s is in watts/sq. in.). Note that only the heat actually dissipated in the liquid should be used in equation 10-8 and the corresponding nomograph. The symbol λ represents an intermediate function. The heat which might be transferred directly to the surroundings without passing through the coolant liquid, should be subtracted from the total heat dissipated to arrive at the correct value. However, such heat losses are usually negligible when the heat sources are freely suspended in the coolant liquid.

10.1.3 Forced convection in liquids.

10.1.3.1 Correlation. The forced convection heat transfer coefficient is determined by the following dimensionless parameters: Nusselt Number, Prandtl Number, and Reynolds Number. The general relationship is presented in the following manner:

$$Nu = cPr^m Re^n \qquad (10\text{-}9)$$

where:

$Nu = \frac{hD}{k}$ is the Nusselt Number

$Pr = \frac{C_p\mu}{k}$ is the Prandtl Number

$Re = \frac{GD}{\mu}$ is the Reynolds Number; G = mass flow density in $lb./ft.^2$-min.

A modification of equation 10-9 is often used in the data correlation for the turbulent flow of fluids in tubes. It is presented as a function of Stanton, Prandtl, and Reynolds Numbers in the following manner:

$$St = c(Pr)^{m-1}(Re)^{n-1} \qquad (10\text{-}10)$$

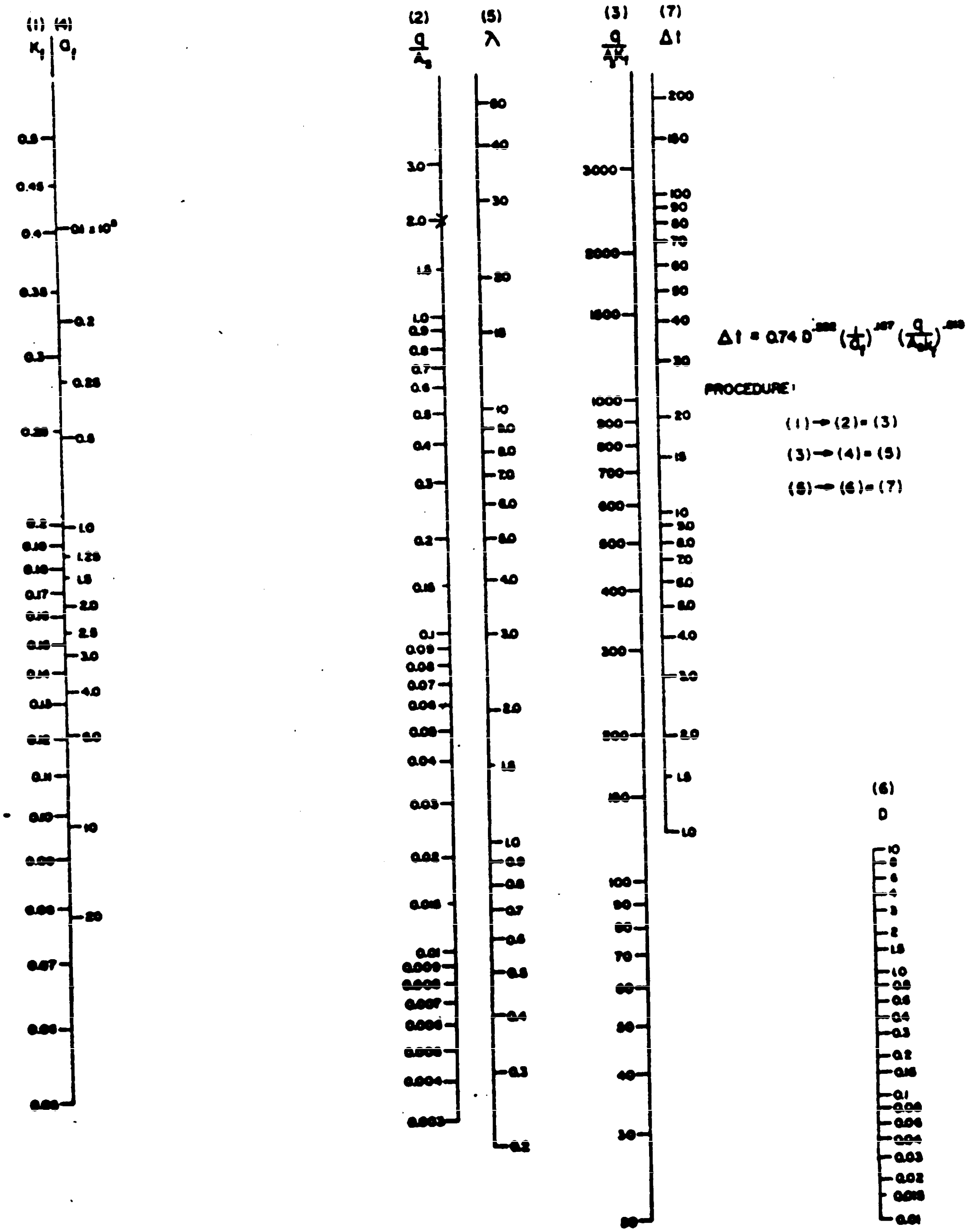


FIGURE 84. Design Nomograph for Equation 10-8

where:

$St = \frac{h}{c_p G}$ is the Stanton Number

Equation 10-10 will prove especially useful in the design of heat exchangers.

10.1.3.2 Flow within tubes.

(a) Laminar Flow

The relationships for the streamline (laminar) flow of fluids within tubes are quite complicated. However, the flow of liquids within tubes will generally be turbulent in the design of electronic cooling equipment. No discussion is therefore included on the topic of streamline flow inside tubes.

(b) Turbulent Flow

A well-known and widely used form of equation 10-9 in which all parameters are evaluated at bulk temperatures, is given by McAdams (Reference 26).

$$h = 0.023 \frac{k_b}{D} Re_b^{0.8} Pr_b^{0.4} \qquad (10\text{-}11)(D.E.)$$

The subscript b refers to the values at the bulk temperature.

This equation can also be written as:

$$h = 0.023 \, (k_b)^{0.6} \left(\frac{c_p}{\mu b}\right)^{0.4} \frac{G^{0.8}}{D^{0.2}} \qquad (10\text{-}12)(D.E.)$$

Equation 10-11 may be used over the following ranges of parameters.

(i) $0.07 < Pr < 120$. This range covers gases, water, and all commonly used liquid coolanets which are warm enough to flow freely. It excludes mercury and those liquid metals which are used as high temperature coolants.

(ii) $10{,}000 < Re < 120{,}000$. For fully developed turbulent flow Reynolds number should be above 10,000, and it should not be much higher than, say 20,000, in order to keep pumping power low.

(iii) L/D ratio should be greater than 60. This requirement is generally easy to satisfy. D = tube diameter for round tubes. L = tube length.

(iv) The temperature difference should be "moderate." All the authorities make this stipulation but give no indication of what "moderate" may mean. McAdams shows very good agreement with test data for a ratio of surface to bulk temperature $\leqq$ 1.6, and increasingly poor agreement as this ratio increases. The ratio t_s/t_b, rather than the actual temperature difference, is the important consideration.

Giedt (Reference 9) recommends the use of a form of Equation 10-10 in which the Stanton number is figured at bulk fluid temperature, Reynolds and Prandtl numbers at film temperature.

$$(St)_b = 0.023\ (Pr)_f^{-0.67} (Re)_f^{-0.2} \qquad (10\text{-}13)$$

This equation can also be written as:

$$h = 0.023 \frac{(c_p)_b}{f^{0.47}} \left(\frac{k}{c_p}\right)_f^{0.67} \frac{G^{0.8}}{D^{0.2}} \qquad (10\text{-}14)(D.E.)$$

Equation 10-14 is limited to the same ranges of parameters as is equation 10-11, but is said to be more precise at larger temperature differences.

For ducts of non-circular section use the relation

$$D = \frac{4 \text{ x cross section area}}{\text{perimeter of section}}$$

10.1.3.3 Crossflow

(a) Single Cylinders. Heat transfer coefficients for liquids flowing across single cylinders can be approximately evaluated by the following relationships, (Reference 1, page 267):

$$h = 0.91 \frac{k_f}{D} \left(\frac{DG}{\mu_f}\right)^{.385} \left(\frac{c_p \mu}{k}\right)^{.31} \qquad (10\text{-}15)$$

for Reynolds Number, Re_f, between 0.1 and 50.

and:

$$h = 0.6 \frac{k_f}{D} \left(\frac{DG}{\mu_f}\right)^{.5} \left(\frac{c_p \mu}{k}\right)_f^{.31} \qquad (10\text{-}16)$$

for Reynolds Number, Re_f, between 50 and 10,000

where:

D = the cylinder diameter, ft.

(b) Single Spheres. The following equations for heat transfer coefficient were obtained from Reference 26, Figure 10-13, page 267):

$$h = 1.73 \frac{k_f}{D} \left(\frac{GD}{\mu_f}\right)^{.288} \left(\frac{c_p\mu}{k}\right)^{.3} \qquad (10\text{-}17)$$

for Reynolds Number, Re_f, between 1 and 15.

and:

$$h = 4.0 \frac{k_f}{D} \left(\frac{GD}{\mu_f}\right)^{.127} \left(\frac{c_p\mu}{k}\right)_f^{.3} \qquad (10\text{-}18)$$

for Reynolds Number, Re_f, between 15 and 1000.

where:

D = diameter of the sphere, ft.

10.1.3.4 Relationship between heat transfer in forced air and in forced liquid cooling. Extensive data on forced air cooling are available in the literature in the following form:

$$Nu = cRe^n \qquad (10\text{-}19)$$

It can easily be seen that the above relationship is a special form of equation 10-9 since the value of the Prandtl Number for air can be considered constant in the temperature range generally encountered in electronic cooling. Usually, in the correlation of experimental data for various fluids, the Prandtl Number is raised to the 0.3 power. Therefore, when a relationship of the form of equation 10-19 is known for the forced air cooling of a given system, the approximate relationship for the liquid cooling of the same system can be derived by the following procedure. The right side of the equation for forced air cooling is multiplied by the factor

$$\left(\frac{(Pr)_L}{(Pr)_A}\right)^{.3}$$

where:

$(Pr)_L$ = the Prandtl Number of the liquid under consideration
$(Pr)_A$ = the Prandtl Number of air

In the range of temperatures generally encountered in the cooling of electronic equipment, the Prandtl Number for air is approximately 0.7. This reduces the above factor to:

$$1.11 \left((Pr)_L\right)^{0.3}$$

10.1.4 Liquid conduction. The mechanism of heat transfer in liquid conduction does not differ from conduction through any other media. Conduction is discussed in detail in chapter 8. Liquid conduction is comparable to the gaseous conduction that is usually noticeable for closely spaced heat sources. Heat transfer by conduction through a liquid separating two solid surfaces is given by the following equation:

$$q = \frac{kA_s}{x} \quad (t_h - t_k) \qquad (10\text{-}20)$$

where:

q = the heat transferred by conduction, watts
k = the thermal conductivity of the liquid evaluated at the average liquid temperature, watts/ft.-°C
A_s = the surface area, ft.^{2}
x = the distance between the two surfaces, ft.
t_h = the temperature of the hot surface, °C
t_k = the temperature of the cold surface, °C

The Appendix lists values of thermal conductivity for various liquids.

As x increases, a value is reached at which convection becomes much more significant than conduction. The properties of each liquid coolant determine this value of x.

10.2 Direct liquid cooling.

10.2.1 General; considerations on immersing electronic parts. In a direct liquid cooling system, the heat dissipating or temperature sensitive electronic parts are immersed in the cooling liquid. All effects of the intimate contact of the coolant and the electronics must be carefully considered, including thermal, electrical, chemical, and mechanical aspects.

From a thermal viewpoint, direct liquid cooling represents a highly effective method of heat removal. Equation 10-7 indicates that the heat transfer coefficient in natural convection varies directly as the fluid thermal conductivity, k, and is proportional to the 0.23 power of the convection modulus. Thus, as an example, comparing air and water at 40°C indicates a heat transfer rate for water approximately 10 times that of air, all other factors being equal.

This same intimacy of coolant and electronics which so enhances the heat transfer rate also necessitates careful selection of the fluid relative to the electrical performance. Higher frequency circuits must be designed to tolerate the dissipation factor and dielectric constant of the coolant used; high voltage circuit design must consider the dielectric strength of the coolant.

Chemically, the coolant must be compatible with all materials used in the electronic circuitry and in the structure of the module, over the entire anticipated temperature range. Toxic and highly flammable materials must be avoided to ensure personnel safety in the event of loss of seal.

The mechanical structure of the module must be adequate to withstand any internally developed pressure. Particular care should be given to case seals and case penetrations (for example, electrical connectors) to insure that the integrity of the sealed case is maintained.

10.2.2 Free convection. Free convection is the simplest form of direct liquid cooling. It is particularly applicable when used in a modular system; direct liquid cooling is employed in small, sealed, replaceable modules to transfer the heat from the heat producing parts to the liquid, and by liquid convection and liquid conduction to the outer module walls for subsequent transfer to the sink. The circulating convection currents are maintained by density variations, which in turn are generated by temperature differences. That is, the fluid in direct contact with the hot surfaces of the electronic parts is heated, expands, and becomes less dense than the surrounding fluid. Because of this bouyancy effect, the warmed fluid rises and is replaced by cooler surrounding fluid, in a continuous process. The warmer fluid circulates until it contacts the colder outer surface of the container, where the heat is transferred. As the fluid cools, its density increases, and the fluid falls. (Note that since the circulation depends on the gravitational force on fluids of varying density, the method is ineffective in a zero-g environment, as in a spacecraft or satellite).

One limitation of free convection direct liquid cooling is that adequate space must be provided around heat generating parts to allow development of convection currents circulating around the components. Because of the complexity of the circulating convection current flow paths, an exact determination of localized flow parameters is not possible, but a qualitative analysis should be made. Dissipative components should be placed with their major axes vertical, in the direction of the natural convection currents. Horizontal panels which might inhibit the free circulation of the fluid should be avoided, or perforated if their presence is necessary.

10.2.3 Direct forced liquid cooling. In the direct cooled systems previously described, the coolant is used to transport the heat energy from the dissipating parts to the outer container walls. The outer walls then reject the heat to the surroundings or sink. In general, this type of system may be designed to dissipate on the order of one-half watt per square inch of effective outer surface area of the container in free air, and as much as two or three watts per square inch if more effective external cooling is used (such as forced air).

When the heat flux density exceeds these values, or when extremely concentrated heat sources exist internally, forced circulation of the liquid coolant may be employed.

Just as increased cooling effectiveness is obtained with forced air compared to free, or natural air convection, forced circulation of the coolant greatly increases the cooling rate when liquid cooling is used. The coolant may be pumped into an external heat exchanger for transfer of the heat into a "heat sink." The pump, of course, requires power for its operation and, unfortunately, most of the energy is expended in the coolant in the form of heat. Thus, the total heat rejected is increased by the amount of power consumed by the pump. However, the advantages of the small temperature gradients and increased cooling rates achieved with forced liquid cooling may outweight the disadvantages of increased system complexity.

A liquid system of this type is presented in Figure 85. The electronic components are completely immersed in a compatible liquid coolant such as silicone fluid or transformer oil. A low-pressure pump circulates the coolant through the system.

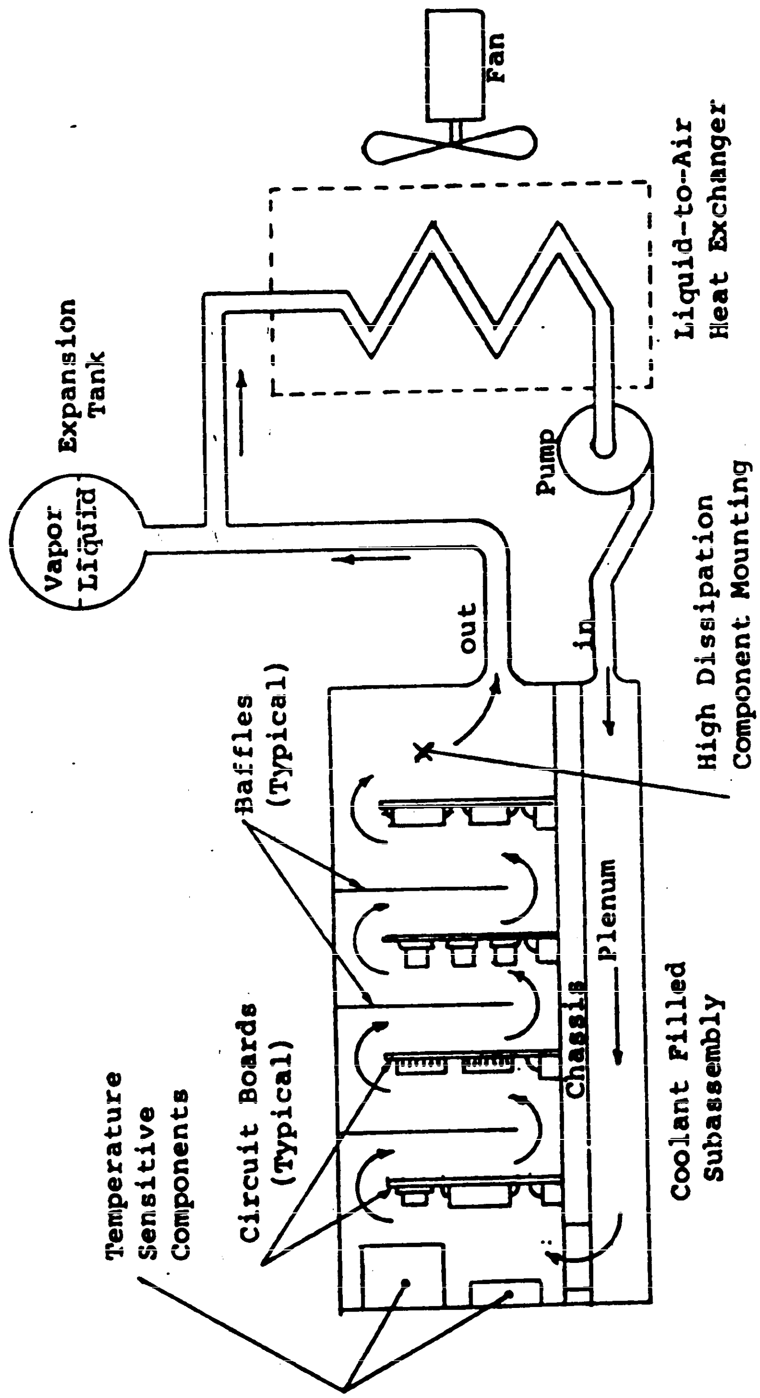


FIGURE 85. Direct Forced Liquid Cooled System Schematic

The accumulator (air cushion tank) allows for expansion of the fluid and serves to minimize vapor lock in the system. The heat exchanger removes the heat from the liquid before it is re-circulated through the electronic equipment. Care must be exercised in the orientation of the parts in order to obtain maximum cooling effectiveness. Further, if relatively high pumping pressures are used, the high pressure stream should not be directed upon fragile electronic parts. Card guides transfer negligible heat from the cards to the coolant because of the low thermal resistance from the parts to the liquid coolant. Thus, this heat flow path can be ignored in the design.

10.2.4 Expansion and pressure effects. In any direct liquid cooled system, be it free convection, free convection with agitation, or forced direct cooled, it is usually necessary to design a sealed system. Sealing is required to prevent loss of coolant by leakage or evaporation and to prevent contamination of the fluid by external materials. Since most all direct cooling fluids exhibit a volumetric coefficient of expansion greater than that of normal container structural materials, provision must be made to relieve or withstand the pressures generated as the system temperature increases. Several methods are available.

The simplest method involves partial filling of the container with the cooling fluid, with the remaining volume filled with air, or with an inert gas, if contamination by air is a potential problem. The compressibility of the gas provides a means for allowing for fluid expansion with increased temperature. Of course, as the gas is compressed, the internal pressure of the module will increase, and the structure and any seals involved must be adequately designed to withstand the internal pressure without rupture or permanent deformation. A variation of this method is that shown in Figure 85, where an accumulator, or expansion tank is provided.

Another technique in use involves filling and sealing the container while the entire assembly (including coolant) is heated to a temperature greater than the highest anticipated operating temperature (this is the operating temperature of the module, not the maximum environmental temperature). On cooling, the pressure within the container will drop, resulting in a low pressure vapor space made up of the volatile constituents of the fluid. This volume will allow for expansion of the fluid, provided that the module operating temperature never exceeds the filling temperature. The container in this case must be adequately designed to withstand an external pressure; that is, normal atmospheric pressure outside the container will be greater than the partial pressure of the cooled liquid coolant within the container.

Any system employing a compressible gas to allow for expansion as previously described must also consider altitude and attitude effects, particularly in airborne installations. Altitude effects refer to the fact that any sealed module operated at altitude will be subjected to a pressure differential due to low ambient pressure in addition to that developed due to thermal expansion effects. (Even shipboard or ground based modules may be subjected to altitude pressure differential effects during air transport, although the modules would not be operational under transport conditions).

Attitude effects refer to the fact that if a sealed module utilizing a free gas or vapor pocket to allow for thermal expansion is operated in an inclined or even inverted position (as during aircraft maneuvering), the pocket may locate itself in a region where heat dissipating components are located. Even though the effect may be short term or momentary, the component for this time period is deprived of liquid cooling effects. Attitude effects may be compensated for by separating the gas and liquid with an elastomeric diaphragm or metal bellows, or by including a hollow elastomeric shape within the filled enclosure.

In those applications wherein the fluid is not subject to deterioration if contaminated, and wherein equipment is operated in a fixed location and always remains in an upright position, it may be possible to operate the electronic assembly in a container of fluid which is vented to the atmosphere. Further, if desired, non-spillable vents such as are used on aircraft batteries may be provided. Some of the difficulties encountered with fluids that have large coefficients of expansions can thus be avoided.

10.2.5 Safety and controls. Any military electronics system must include not only compliance with specification requirements as far as operating environment is concerned, but also recognize any safety hazards to the operating personnel in case of catastrophic failure. For example, even a non-repairable sealed module could be ruptured due to a direct hit. Consequently, toxic or inflammable cooling fluids must not be used, since they could present a safety hazard. Consideration should be given to rupture diaphragms on sealed modules operating at a positive internal gage pressure, to insure a predictable failure mode in case of severe overload, instead of an explosion.

Direct liquid cooling systems (except direct forced liquid) are normally passive thermal designs. They should be designed to accommodate the maximum anticipated heat load at the maximum service operating temperature, and once designed, the heat flow paths are fixed. Thus, temperature control with varying heat dissipation or sink temperature is not possible. Since a direct forced liquid cooling system utilizes an external pumping source, regulation of flow, and generally advisable for reasons of economy. Control of a direct forced liquid cooling system is similar to that of indirect liquid cooled systems, and is described in more detail in section 10.6.2.

10.3 Indirect liquid cooling.

10.3.1 General. In an indirect system, the liquid coolant does not come into direct contact with the electronic parts. Heat is removed from the parts by natural convection, conduction, or radiation to a liquid cooled panel or cold plate and "carried away" to the heat sink by the cooling liquid. At the heat sink, the heat is removed from the coolant by a heat exchanger. The electronic equipment may be internally designed to emphasize one, or any combination of the various methods of heat removal. Where metallic conduction is involved, the contact interface thermal resistances should be minimized. (See Reference 25). Figures 86 and 87 show two typical indirect liquid-cooled chassis.

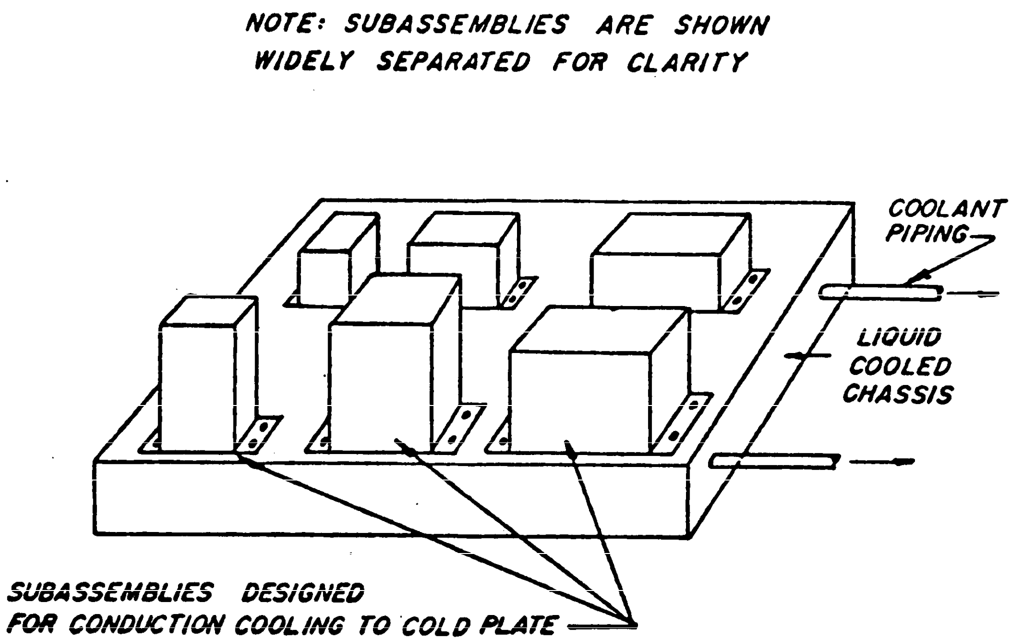


FIGURE 86. Liquid Cooled Cold Plate

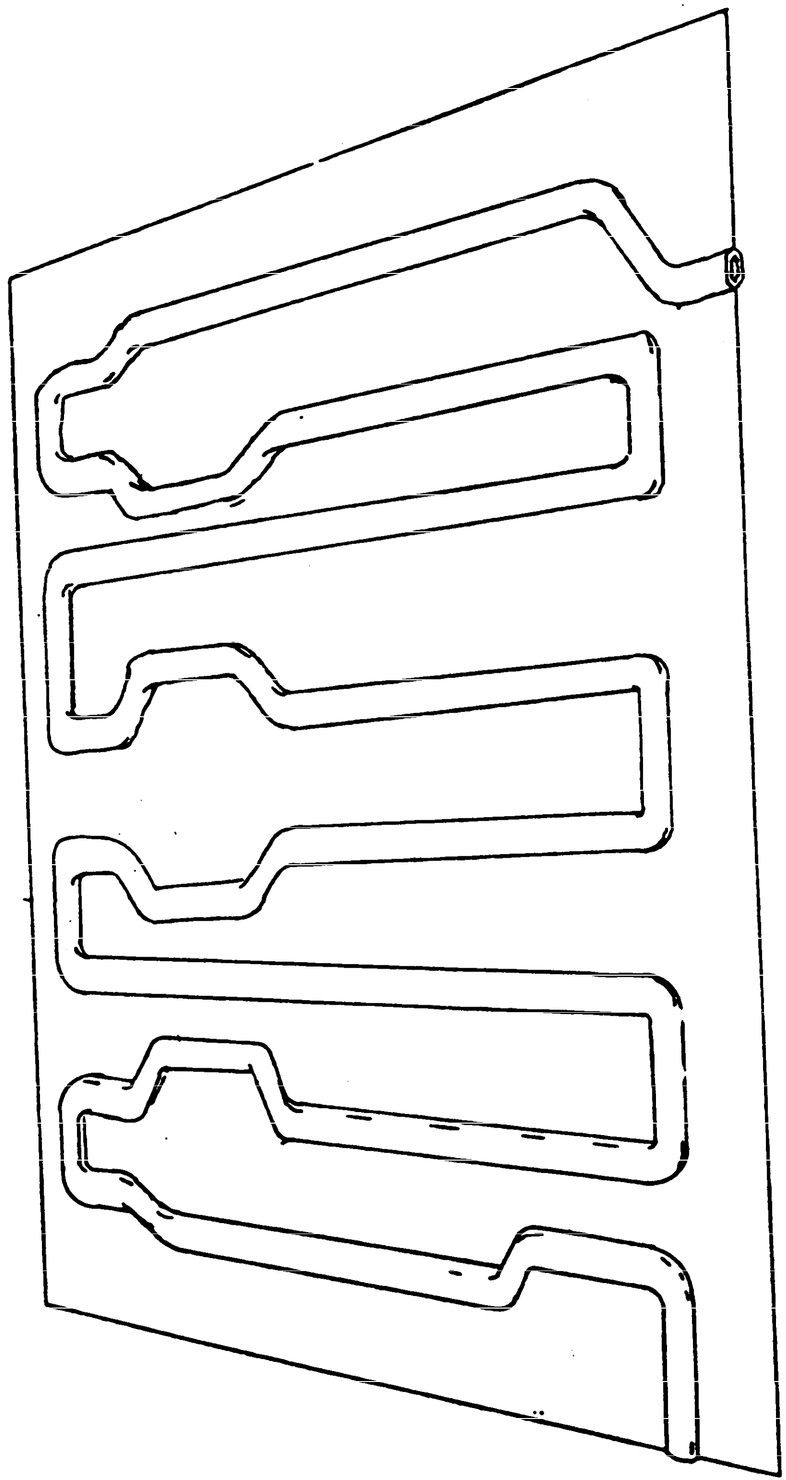


FIGURE 37. Expanded Metal Thermo Panel Chassis

Fresh water is recommended as the fluid to be used for most liquid cooled shipboard applications. MIL-W-21965, "Water Cooling of Shipboard Electronic Equipment", must be followed.

10.3.2 Cold plates.

10.3.2.1 General. A "cold plate" is a chassis or panel incorporating a heat exchanger on which electronic assemblies are mounted. Liquid-cooled cold plates are more efficient than air-cooled cold plates.

The plate may be cooled by the liquid flowing through tubes which are in intimate contact with one side of the plate, or the plate itself may consist of a "sandwich" type heat exchanger through which the liquid is forced. (See Figure 87) In the case of expanded metal chassis, passages through which the cooling liquid flows are an integral part of the surface on which the electronic parts are mounted.

The use of fresh water for cooling shipboard electronics is much more effective than air in a cold plate because the film coefficient is greater (as much as 100 times). Another advantage in using water is that its specific heat is more than four times that of air, so that, for the same temperature rise and weight flow rate, water will absorb four times as much heat. Further, water is rather easy to transport, and high flow velocities may be obtained without excessive noise. Since water is over 800 times as heavy as air, the piping will be much smaller than the equivalent air ducting. Thus, it follows that the water cooling of electronic equipment has considerable merit, provided that water supply and return piping is made available in each compartment (and properly maintained).

10.3.2.2 Applications. Cold plate cooling systems are widely used because they provide a highly efficient heat transfer method. The use of cold plates on shipboard, using ship-supplied fresh water coolant, has already been alluded to. Aircraft and spacecraft frequently use cold plate systems because of the high heat transfer rate/weight ratio. Aircraft cold plate systems eventually dissipate the heat absorbed by the coolant into the air, by ram or bleed air cooling or by a "wet skin" heat exchanger (Section 10.3.4). Spacecraft dissipate the coolant heat by skin radiator heat exchangers.

In some systems the pumping of the coolant may be self-contained by vaporizing the coolant at the heat source, condensing the vapor in a coolant-to-ultimate sink heat exchanger, and returning condensate to the source. These lead to vaporization cooling systems (chapter 11) or heat pipe designs (section 12.2).

10.3.3 Heat transfer from parts to liquid cooled cold plates. A properly designed liquid cooled cold plate will efficiently transport dissipated heat from the source region to an ultimate sink, usually by way of an external heat exchanger. The equipment designer, however, must assume responsibility for heat flow paths within the equipment from the heat dissipating parts. Generally, the natural cooling methods of conduction, free convection, and radiation are available for this thermal link. Of these, conduction is preferred.

Resistors having significant dissipation should be provided with a low resistance thermal path to the cold plate chassis by use of bonding materials, or, preferably, by a metal clamp. Low power transistors, diodes, and similar devices should likewise be provided a direct thermal link to the cold plate, unless it can conclusively be established that free convection and radiation is adequate for cooling. Power transistors, transformers, and devices of high heat dissipation must be mounted with low thermal resistances to the cold plate chassis. Receiver type tubes should be mounted in conduction cooled shields. High power tubes usually require special thermal treatment, with forced air or conduction cooling or direct immersion in a liquid.

In evaluating the thermal path between any given component and the cold plate chassis, there will be at least one, and possibly several interfaces in the path. These interfaces must be carefully designed, particularly where the thermal flux density is very great (usually at the component mounting interface). Interface resistance in conduction paths has been discussed in detail in chapter 8, but the effect is significant enough in indirect liquid cooled systems to warrant a brief summation. Interface thermal resistance is reduced by (1) increased contact pressure at the interface, (2) low values of surface roughness and flatness, (3) and cleanliness of the mating surfaces. If interface resistance cannot be sufficiently reduced by these means (for example, it is very difficult to obtain appreciable mating pressures between some integrated circuit devices and a cold plate or substrate), then the use of a filler material with a high thermal conductivity is warranted.

10.3.4 "Wet skin cooling" - Airborne. The "wet skin" cooling system is peculiar to airborne applications. The system is essentially a liquid-to-forced air heat exchanger built into the aircraft outer skin with the air side of the exchanger exposed to the external air stream. Fins may be provided to increase the forced air convection area on the air side. A typical wet skin heat exchanger configuration is shown in Figure 88.

Design of this type of heat exchanger can be complex due to the large number of variables involved. The first requirement is a definition of the aircraft operating envelope, that is, the mission profile of altitude vs. Mach number, and both altitude and Mach number as a function of flight duration. Without developing a detailed design procedure, it may be noted that the heat exchanger designer is primarily interested in obtaining the overall heat transfer coefficient, U. For the air side, U is a function of the air density, air temperature, and flow rate. In turn, the air flow rate is a function of the Mach number and air density; the air density and temperature are functions of altitude; and at higher Mach number, an aerodynamic heating load is superimposed. Because of this complex interaction of variables, designs of this type are best left to specialists in the field. Computers are nearly always utilized to examine the effect of changing parameters.

It should be noted that for this type of cooling system no cooling (other than natural convection) is provided at zero air flow rate. Consequently, supplementary air flow or coolant heat exchangers must be provided if the electronic systems are operated at full power during maintenance and checkout on the ground.

10.4 Equipment design considerations for liquid cooling.

10.4.1 General. The use of liquid cooling, whether direct or indirect, requires that certain construction procedures and layout principles be followed. Assemblies should be so constructed that the heat-producing parts are separated from the passive parts and particularly from those that are temperature-sensitive. In all cases, full advantage should be taken of conduction. Conductive metal heat flow paths should be designed into the equipment for the most direct transmission of heat from sources to sink.

10.4.2 Direct liquid cooling. In direct liquid cooling systems, it is necessary to expose a maximum of the surface of the heat-producing parts to the coolant and to direct the free convection currents of the fluid around these parts. Thus, in liquids, as in air, parts should be mounted to promote convective cooling. Metallic conduction paths of low thermal resistance from the heat-producing parts to the surface of the case are not as important in direct liquid-cooled assemblies as in most other types of assembly since adequate cooling is usually obtained by convection. The parts may be supported by solid insulating materials so long as the fluid is permitted to freely circulate around the parts. Where applicable, the part should be mounted such that the longer part dimension is vertical. The construction of this type of equipment must be given special consideration. With viscous coolants, a slight mechanical advantage is gained by the immersion of the electronic parts in the fluid, since the fluid tends to lend support to parts. Also, it can provide a damping action which assists in resisting vibration and shock, dependent upon the viscosity of the fluid. Generally, the lowest thermal resistances from the parts to the coolant are obtained with direct liquid cooling.

An accurate quantitative analysis of the temperature distribution within a sealed direct liquid cooled system is usually very difficult to obtain, due to the indeterminate overall nature of the circulating convective currents. Nevertheless, at least a qualitative inspection should be made with regard to parts placement. Thus, temperature-sensitive parts should not be placed where they would be in the convection current generated by a high dissipation part. Usually a temperature gradient will exist in a direct cooled module, with the lowest temperature at the bottom. Therefore, the more sensitive parts should be placed in the lowermost locations. Holes can be provided in subchassis to direct the flow of the free convection currents around the heat producing parts. The use of direct liquid cooling should not be considered a panacea for thermal problems. The effectiveness of the contained coolant depends primarily on the generation of auto-convection currents. Parts placement should recognize this requirement and not interfere with circulation. Large plates such as chassis or printed circuit boards, should be mounted vertically, since a horizontal orientation would restrict the convection currents. If horizontal plates are required, numerous and large coolant passage areas in the form of slots or holes must be provided.

While direct liquid cooling generally allows an increase in parts density in a module because of improved thermal paths, the increase in packaging density is limited by the space requirement for allowing adequate circulating convection current to develop. Sufficient gaps must be provided between adjacent vertical plates to insure liquid flow (as, for example, between a

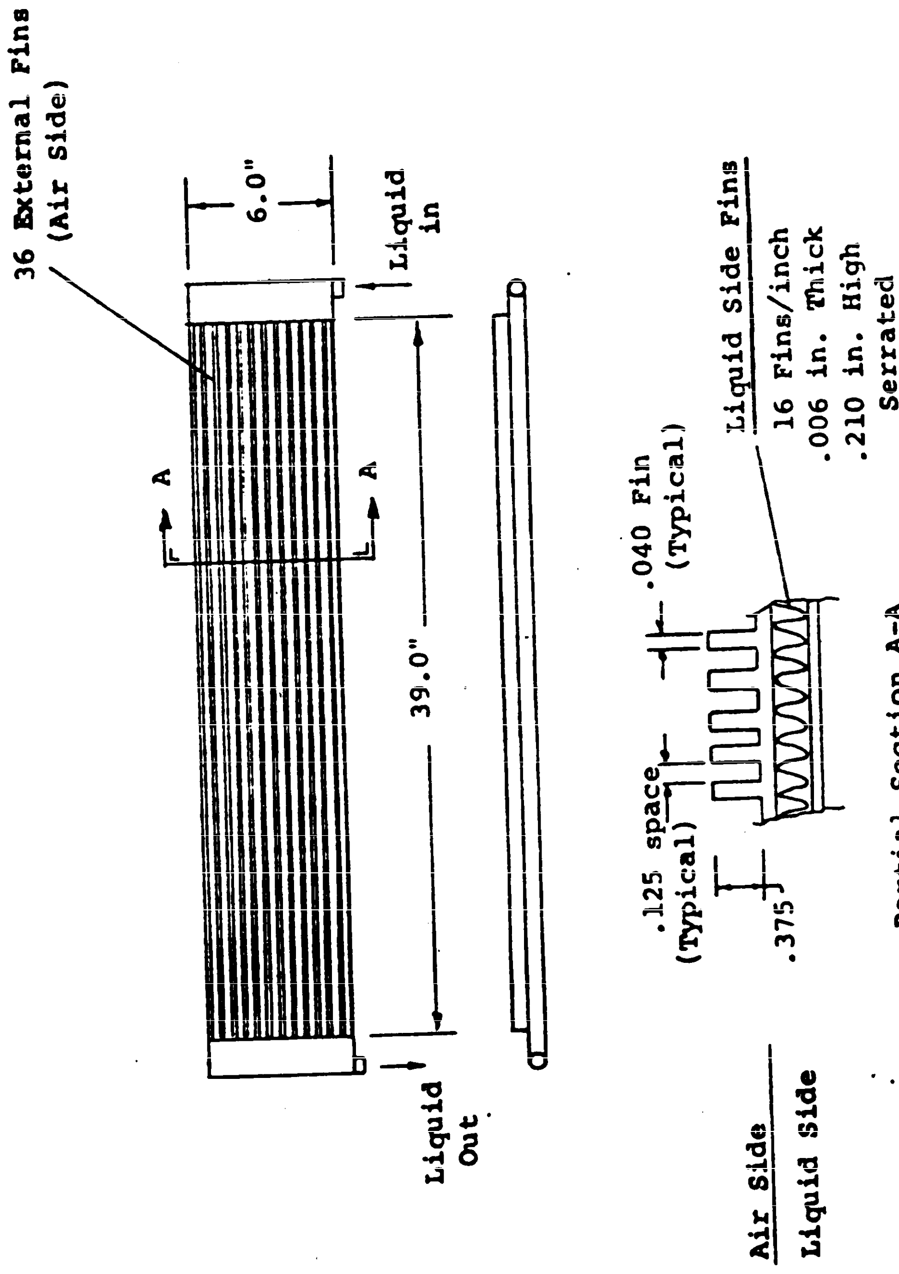


FIGURE 88. "Wet Skin" Heat Exchanger

circuit board and a module wall). When packaging density requirements and circulation area requirements conflict, consideration should be given to agitation of the coolant to promote circulation.

10.4.2.1 Considerations for direct immersion of electronic parts. In considering direct liquid cooling as an alternative cooling technique, much more is involved than merely replacing the internal air with a liquid coolant. The full effects of the coolant intimate contact-thermal, electrical, chemical, and mechanical must be evaluated, both from a circuit and from a component viewpoint.

The primary effect from a circuit viewpoint is, of course, electrical, and involves the breakdown voltage (dielectric strength) of the coolant in high power applications, and the dielectric constant of the coolant in higher frequency applications. In addition, the fluid selected must not be adversely affected by the electrical fields or voltages normally produced in the equipment.

Components must be carefully screened to insure compatability with the coolant. The electrical effect on normal air-dielectric components such as trimmer capacitors is obvious. Chemical compatability with component cases, circuit board material, rosins, and potting compounds, and any other internal module material must be verified. Plastic encased semi-conductors are susceptible to leakage in some fluids when the parts are immersed directly in the liquid.

Heat transfer inside a direct liquid-cooled electronic assembly may be difficult to predict due to the complicated shape of electronic parts and the variation of film coefficients. Satisfactory preliminary designs for small modules can usually be achieved by considering the assembly as a single heat source. By constructing a breadboard model and conducting electrical and thermal tests, the system may be modified as required in order to obtain final design data. The tests on the breadboard model should provide the temperature limits through which the liquid must be maintained for variations in the thermal environment of the heat exchanger. In order to operate within these temperature limits it may be necessary to add control equipment.

In direct forced liquid cooling, part layout and location are most important. The cooled liquid or that coming from the heat exchanger should be directed to cool the temperature-sensitive parts first and later be directed adjacent to the higher power heat producing parts.

10.4.2.2 Sealing considerations. In a direct liquid cooled system, sealing is usually necessary, since most coolants suffer deterioration when contaminated with air or water. The sealing methods employed depend to a large degree on the size of the system being cooled.

Direct liquid cooling is particularly applicable to small non-repairable modules, which can be cooled externally by natural convection and radiation or by direct conduction to a chassis, which may in turn be "cold plated" with a liquid cooling system. Modules of this type should be hermetically sealed, with appropriate provisions made for fluid expansion with temperature. Induction soldering with high temperature solder is a particularly appropriate sealing method. Special consideration must be given to container penetration, such as for electrical connectors. These penetrations generally represent a stress point in the structural design of the sealed unit when it is subject to a pressure differential. Connectors, particularly for coaxial lines, must be leakproof.

Larger direct cooled systems and direct forced cooled systems are not generally hermetically sealed since accessability must be provided for maintenance, repairs, and replacements. These units generally have removeable covers or access plates, sealed with an elastomeric gasket. In these designs, the cover must be stiff enough, and the cover screws must be spaced closely enough to ensure that there will be not loss of seal and consequent coolant leakage under any pressure differential resulting from thermal expansion and/or altitude effects.

Components of the more elaborate closed systems used in direct forced cooled or indirect cooled systems-piping, joints, valves, pumps, expansion tanks, etc., are discussed in section 10-6.

10.4.3 Indirect liquid cooling considerations. Indirect liquid cooling has certain advantages over immersion or direct liquid cooling. Among these are easier accessibility for maintenance, less possibility of fouling equipment with coolant, less handling of coolants, since the coolants could be contained in a completely closed system, the ability to use coolants that have excellent thermal properties but poor electrical properties (e.g., water) and the capability of temperature regulation under varying heat load and environmental conditions. Indirect liquid cooling can be utilized in one of two ways: (1) using forced-air cooling to actually remove the heat, and then exchanging the heat to a coolant in an air-to liquid heat exchanger which is an integral part of the equipment, or (2) using conduction cooling to a liquid-cooled panel or chassis on which the heat producing parts and assemblies are mounted directly.

Refer to chapter 9 and to sample problem No. 1, section 10.5.3.5 for the particulars of utilizing the forced-air-to-liquid heat-exchanger approach.

In the design of an indirect liquid-cooled equipment, consisting of liquid cooled panels or chassis, metallic conduction to the cooled panels will be the predominant heat transfer mode, with the coolant being used to remove the heat from chassis or panels rather than from the hot parts directly. Each component assembly of an equipment utilizing this cooling method should be designed to permit the conduction of the heat produced to a heat conducting surface where the assembly is attached to the cold plate. Smaller parts, not part of an assembly, should be mounted directly on the cooled panel. Chapter 8, section 8.3.3 outlines procedures for conducting the heat from the heat sources to cold chassis or panels. The cold plate liquid is generally cooled by a liquid-to-liquid or liquid-to-air heat exchanger (section 10.5).

10.4.3.1 Mounting of parts for indirect liquid cooling. In indirect liquid cooling, good metallic conduction from the electronic heat sources to the cooled panel, plate or chassis, is required in order to arrive at an efficient configuration. Initial consideration of the parts to be used in an equipment design should include choice of mounting for conduction cooling. (See section 10.3.3)

The following are recommended mounting procedures for small parts:

(1) Resistors should be thermally bonded to the panel or chassis by a clamp of low thermal resistance.
(2) Receiving-type electron tubes should be mounted in tube shields designed for conduction cooling. High power tubes usually require forced air cooling.
(3) With transformers care should be taken to insure that the core is in intimate contact with the chassis. Metal shims may be added if required.
(4) High power semiconductors should be mounted directly to the cold plate chassis, with a heat conducting compound at the joint.

10.4.4 Dewpoint. Many military equipments are expected to operate under conditions of moisture condensation, as specified in the applicable environmental specification. Some units, however, are designed for operation in controlled atmospheres, where ambient temperature and humidity are kept within specified bounds. Many computers or portions of computer systems fall into this category. If liquid cooled systems are used in these applications, the possibility of condensation exists due to portions of the liquid cooled system being operated at a temperature below the highest possible ambient dew point. While most heat transfer components, such as the coils of a liquid cooled heat exchanger, would not in themselves be damaged by condensate, the effect on adjacent electronics could be seriously detrimental. Cold plate chassis are particularly susceptible to condensation under adverse conditions. Because electronics are directly mounted to cold plate surfaces, the equipment designer must either ascertain that condensation will not occur or must recognize the possibility in the design of the equipment.

Shipboard fresh water cooling systems normally supply coolant at a maximum temperature of 40°C (105°F) (See MIL-STD-1399). If the characteristics of the ambient air are known, the possibility of condensation may be checked with a psychrometric chart.

In many military systems, condensation is unavoidable because of the operating conditions to which the system is being designed. In these instances, wide trace spacing and component or module encapsulation or conformal coating must be used. Drain holes should be provided where necessary to prevent accumulation of condensate in natural pockets.

10.5 Heat exchangers.

10.5.1 Types, limitations, and characteristics. Heat exchangers are normally classified according to their internal construction. The common types are:

10.5.1.1 Conventional shell-and-tube type heat exchangers. In this type of heat exchanger one fluid flows inside the tubes and the other fluid flows across or along the outside of the tubes depending upon the construction. Extensive design data are available on this type of exchanger.

The classification of shell-and-tube type heat exchangers is further subdivided into parallel flow heat exchangers, counterflow heat exchangers, reversed flow heat exchangers, and crossflow heat exchanger, based upon the direction of flow of the shell-side fluid relative to the tube-side fluid (see Figure 89).

Counterflow heat exchangers are more efficient thermally than parallel flow heat exchangers. In counterflow heat exchangers the outlet temperature of the hot fluid can be lower than the outlet temperature of the cold fluid. Reversed flow is a combination of parallel and counterflow.

10.5.1.2 Extended surface heat exchangers. For greater compactness, additional heat transfer surface can be obtained by the use of fins in good thermal contact with the primary heat transfer surface. There are many types of extended surfaces for heat exchangers. Figure 90 shows commonly used extended surfaces applicable to heat exchangers for electronic equipment cooling. Finned-tube heat exchangers are particularly effective when the tube fluid is a liquid and the second fluid is a gas at ordinary pressure. They are, therefore, suited for the design of air-to-water heat exchangers employed in forced air-cooled electronic equipment.

A relatively new modification of the extended surface heat exchangers is the "inner fin" surface. The longitudinal arrangement keeps the pressure drop at a low value. The "inner fin" provides a greater surface area, resulting in better transfer of heat (see Figure 90e).

10.5.1.3 "Tube-in-strip" heat exchangers. Cored heat exchangers usually are quite thick. The space which is normally available in electronic equipment for heat exchangers is sometimes limited in one dimension and does not always permit the use of wide compact heat exchangers. Flat heat exchanger panels are best suited for the cooling of equipment with a low heat concentration. Such exchangers have a high effectiveness, are low in weight, and because of their thinness, fit well along the walls of electronic equipment enclosures.

There are two basic types of panel heat exchangers. One type consists of two plates with suitable embossings welded together to form the necessary flow channels. The other type consists of a single sheet of metal (copper, brass, or aluminum) rolled from a casting or graphite-coated sheet and then inflated to produce flow channels. Due to the method of manufacturing, this type panel can be patterned quite intricately to suit the needs of any particular cooling system. (See Figure 87)

10.5.1.4 Comparative features. Shell-and-tube devices are well adapted to high pressures, and can easily be designed so that the inside of the tubes can be cleaned by brushes or reamers. The outside of the tubes is difficult to clean; the shell should thus contain clean or non-scale-forming fluid. Expansion is easily allowed for by making one tube header floating and by placing an expansion joint in the shell. Shell-and-tube designs are heavier and bulkier than the other types.

Figures 91 and 92 illustrate the temperature gradients in tubular heat exchangers. With counterflow, the temperature difference is nearly constant, and the exit temperature of the cold fluid can be higher than the exit temperature of the hot fluid. All portions of the tube surface have about the same

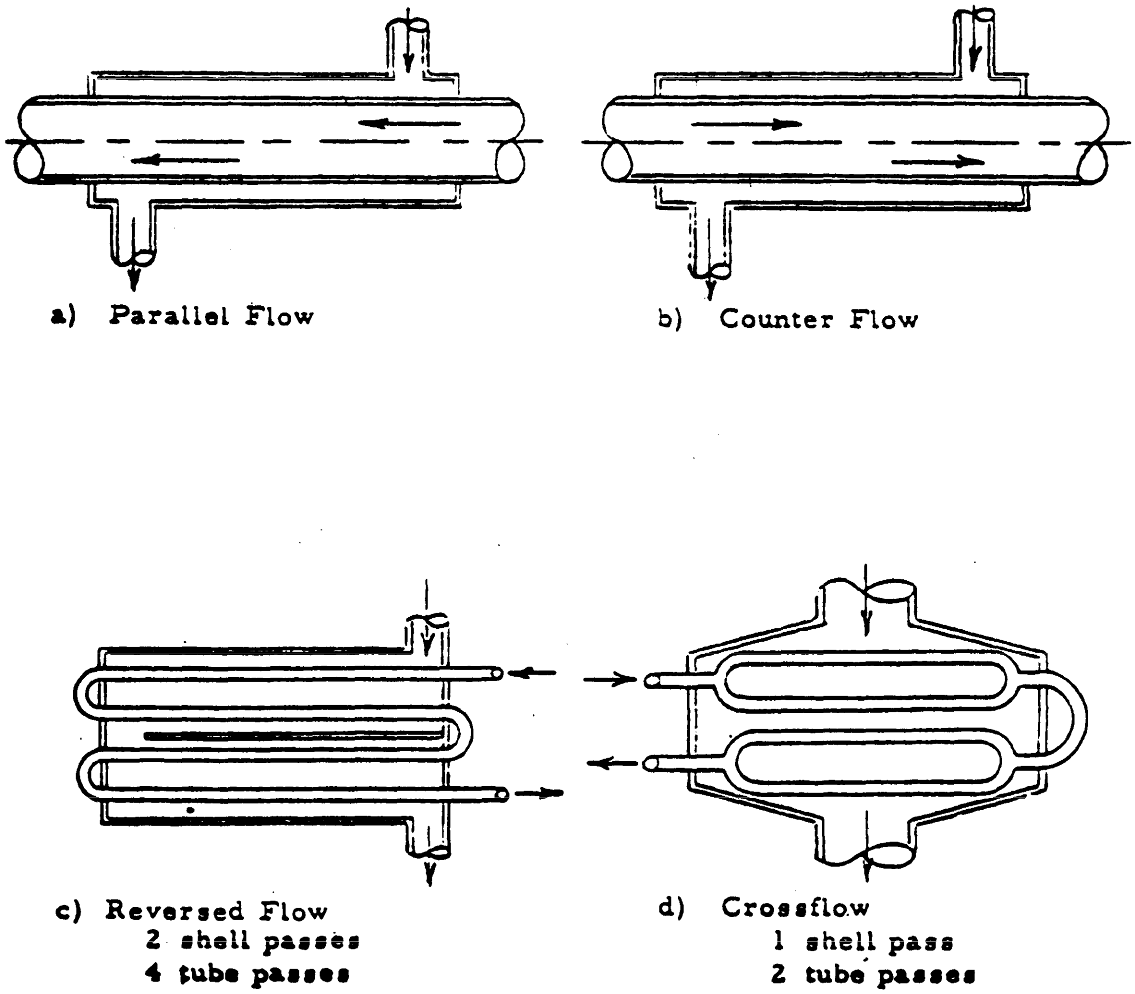
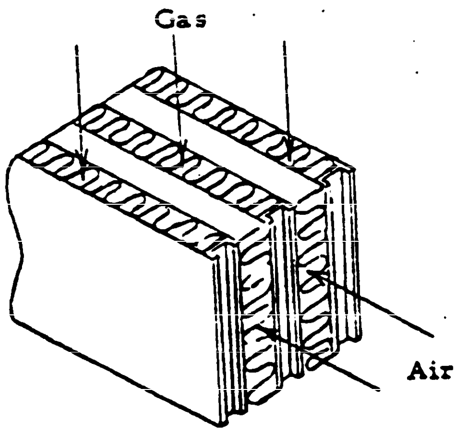
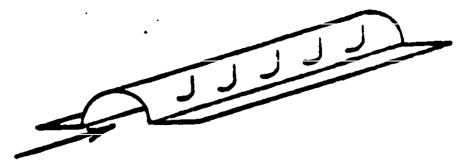


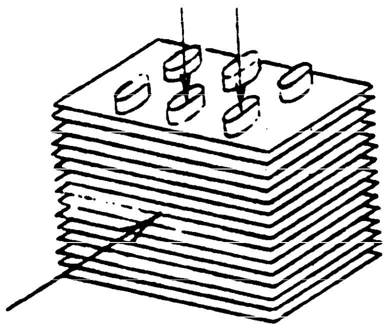
FIGURE 89. Shell-and-Tube Type Heat Exchangers



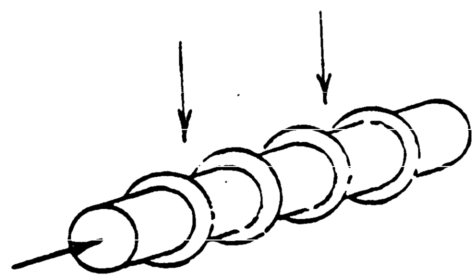
a) Plate Fin Surface



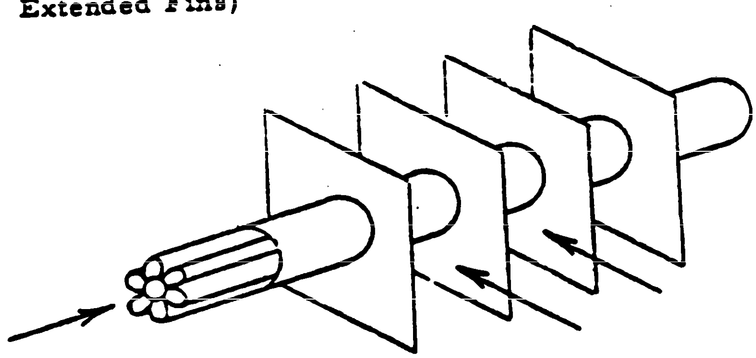
b) Louvered Fin Surfaces



c) Finned Tube Surface (Flattened Tubes, Extended Fins)



d) Finned Tube Surface (Round Tubes)



e) "Inner Fin" Extended Surface

FIGURE 90 Details of Typical Extended Surface Heat Exchanger heat transfer effectiveness, and extreme temperature differences are not present. With parallel flow, the heat transfer rate is high over the first part of the tube length, and then decreases. Initial cooling (or heating) is therefore, rapid. Parallel flow exchangers tend to be shorter than counterflow exchangers.

In crossflow exchangers the flow pattern is complex and very difficult to calculate, so that empirical formulas must be used. Turbulent flow in the shell side is easily developed. Crossflow exchangers tend to be "boxy," roughly square in shape. Counterflow exchangers are long and of small diameter. Parallel flow exchangers are shorter and of larger diameter.

In crossflow, as in parallel flow designs, the highest temperature of the cold fluid is lower than the lowest temperature of the hot fluid, and the temperature difference varies widely from point to point in the core.

Extended surface types can be made much lighter in weight (for low pressures) than shell-and-tube types, and the construction methods are frequently cheaper. Mechanical removal of scale is difficult or impossible, but dust and dirt accumulated on a gas side is easily removed.

10.5.2 Design.

10.5.2.1 General. In a heat exchanger, heat is transferred from one fluid to another, separated from the first fluid by a solid wall. The wall iself is commonly made of material of high thermal conductivity. Each fluid forms a laminar boundary layer on the wall surface. The wall and the layers of scale are treated by conduction theory, and the hot and cold fluids are treated by convection theory. The mechanism of heat transfer in the exchanger is thus, rather complicated.

10.5.2.2 The local heat transfer coefficient. Figure 93 is a schematic drawing of a heat exchanger surface, showing the temperature distributions for parallel flow.

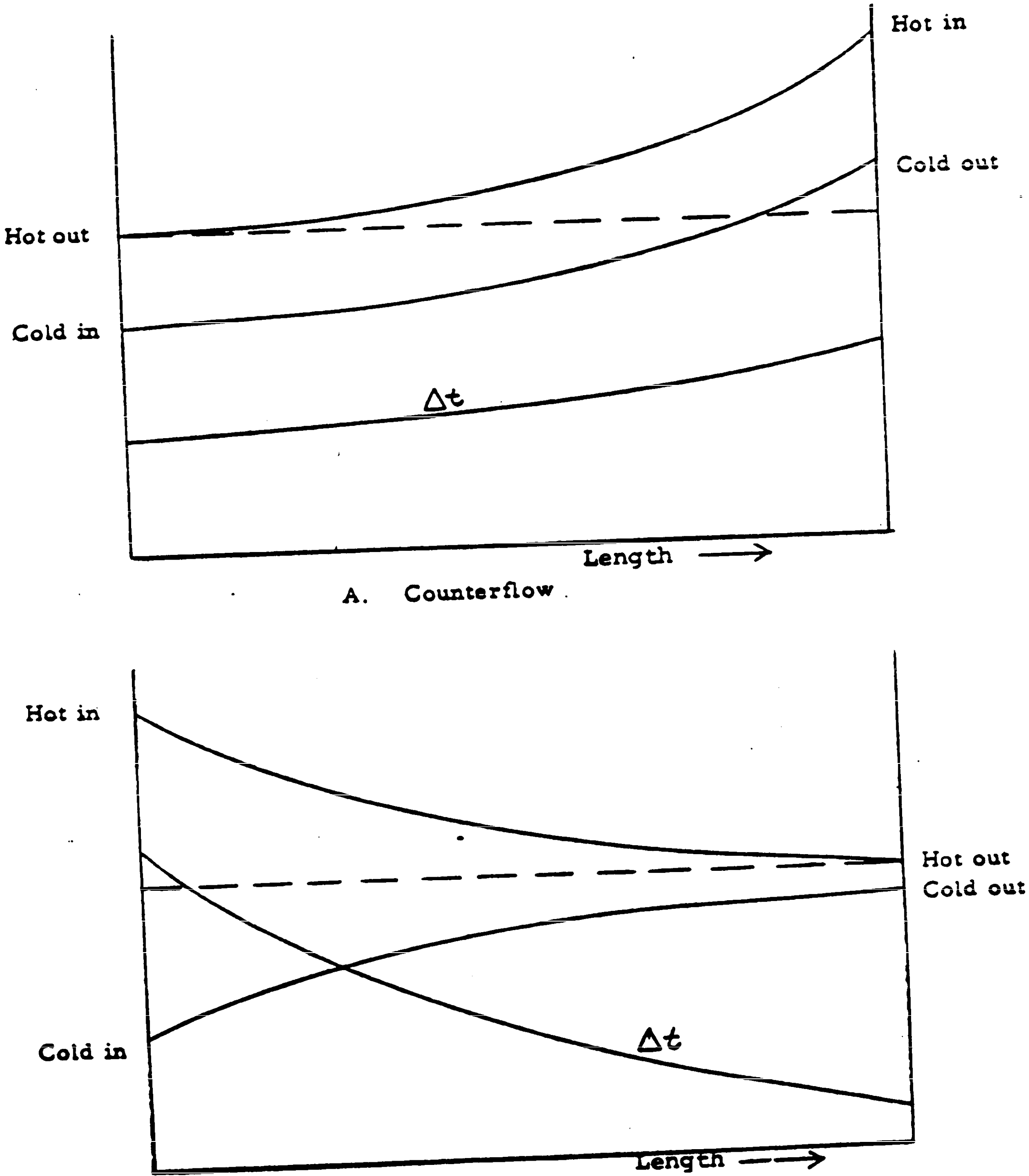
t_h = bulk temperature of hot fluid
t_k = bulk temperature of cold fluid
t_{wh} = wall temperature, hot side
t_{wk} = wall temperature, cold side
Subscript x refers to conditions at a particular point, longitudinally.
Subscript 1 refers to the entrance conditions.
Subscript 2 refers to the exit conditions.

At point x, the heat transfer from wall to fluid can be expressed in terms of local coefficients of convection, wall conductivity, and areas, as follows:

$$\Delta q_{h_x} = h_{h_x} \, \Delta A_h \, (t_{h_x} - t_{wh_x})$$

$$\Delta q_{w_x} = \Delta A_w \, \frac{k_{w_x} (t_{wh_x} - t_{wk_x})}{y}$$

$$\Delta q_{k_x} = h_{k_x} \, \Delta A_k \, (t_{wk_x} - t_{k_x})$$



A. Counterflow

B. Parallel Flow

FIGURE 91. Bulk Temperature Variation in Heat Exchangers

Note: The vertical scale represents temperature; the horizontal scale, and distance.

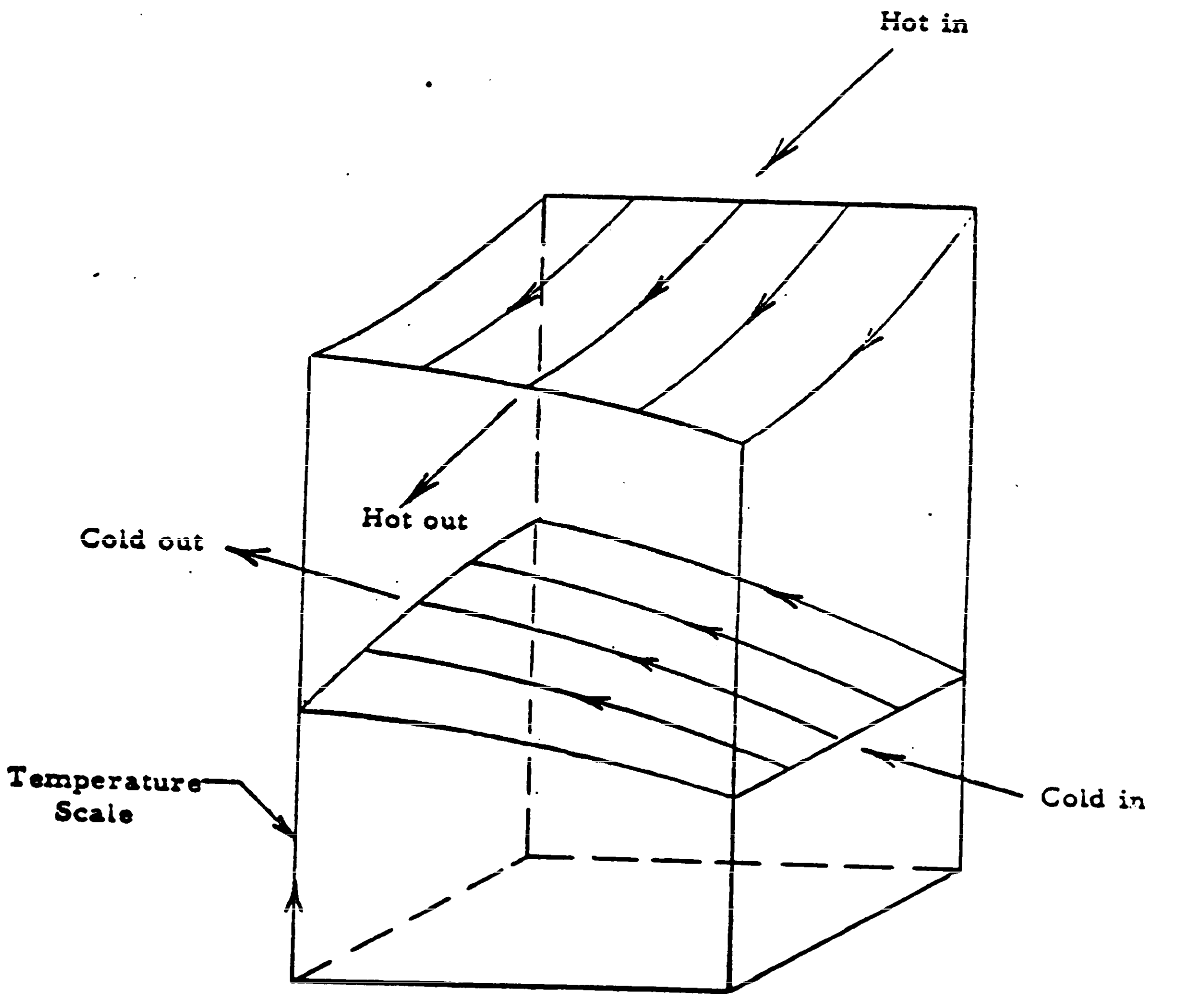


FIGURE 92. Bulk Temperature Variation in Crossflow Heat Exchangers

Note: The vertical scale represents temperature; the horizontal scale, and distance.

The incremental areas may, in general, be different, but the same amount of heat passes through each one, so

$$\Delta q_{h_x} = \Delta q_{w_x} = \Delta q_{k_x} = \Delta q$$

Eliminating the wall temperatures from these three equations, and allowing the incremental quantities to become differentials by the usual process of calculus, the following differential equation is obtained

$$t_{h_x} - t_{k_x} = dq\left(\frac{1}{h_{h_x}\, dA_h} + \frac{y}{k_{w_x}\, dA_w} + \frac{1}{h_{k_x}\, dA_k}\right) \qquad (10\text{-}21)$$

A local overall heat transfer coefficient, U_x, is now defined by the equation:

$$dq = U_x dA\,(t_{h_x} - t_{k_x}) \qquad (10\text{-}22)$$

The element of area dA is not necessarily the same as any of the other areas, and may be chosen arbitrarily by the designer.

Combining equations 10-21 and 10-22 results in the equation:

$$\frac{1}{U_x} = \frac{dA}{h_{h_x}\, dA_h} + \frac{dA}{h_{k_x}\, dA_k} + \frac{y\,dA}{k_{w_x}\, dA_w} \qquad (10\text{-}23)$$

If the wall is of composite structure or covered with a layer of scale, additional terms similar to the last term in equation 10-23 will be added, the general form being:

$$\frac{(\text{thickness of layer}) \times dA}{(\text{conductivity of layer}) \times dA_w}$$

Equation 10-23 shows that the local heat transfer coefficient is analogous to the electrical conductivity of several elements in series. If the surfaces are smooth and parallel, all the elements of area will be equal, and, where subscript s refers to scale deposit:

$$\frac{1}{U_x} = \frac{1}{h_{h_x}} + \frac{1}{h_{k_x}} + \frac{y_w}{k_{w_x}} + \frac{y_s}{k_{s_x}} \qquad (10\text{-}24)(\text{D.E.})$$

10.5.2.3 The logarithmic-mean temperature difference. The local heat transfer coefficient, U_x, will vary from point to point, since it depends on the properties of the fluids, the local temperature, velocity, and other conditions. It is customary to calculate or estimate an average overall value of coefficient U, based on the area of tube surface. This is discussed in section 10.5.2.4 below.

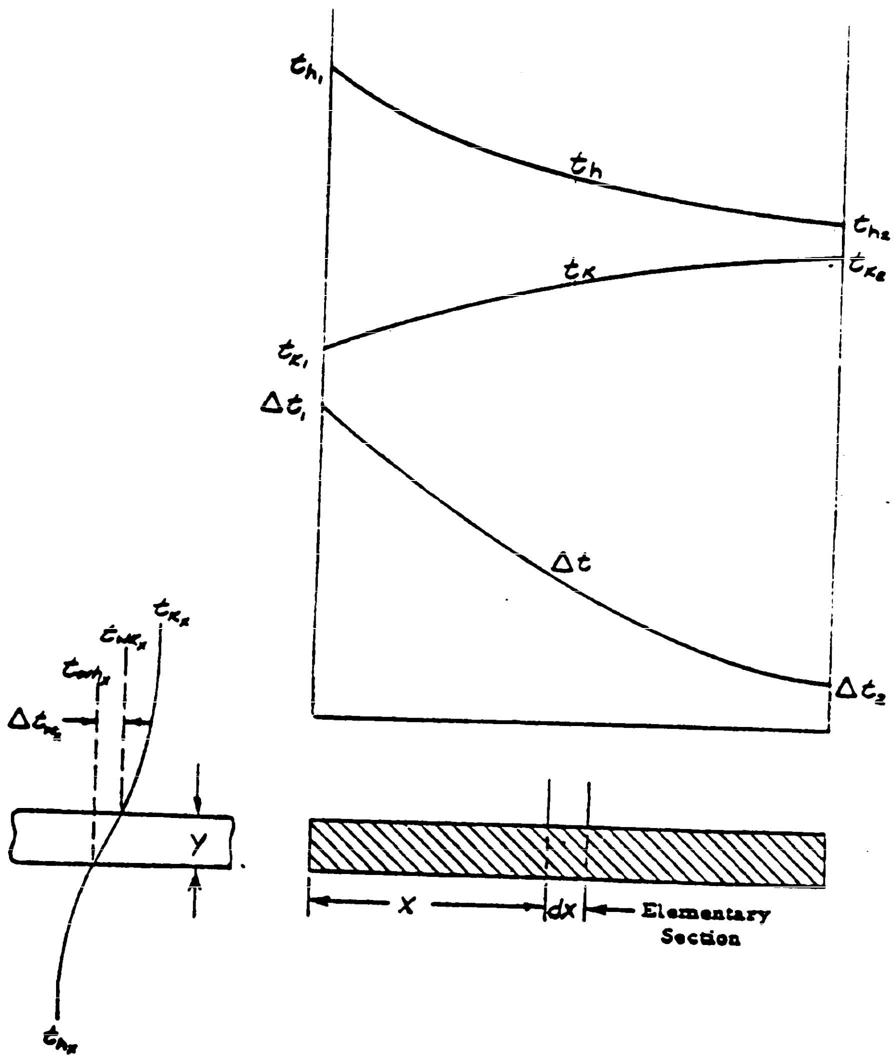


FIGURE 93. Elementary Parallel Flow Heat Exchanger

Note: $\Delta t = t_h - t_k$

In order to calculate the total heat transfer it is necessary to have some mean value of temperature difference. The bulk fluid temperatures and the local temperature difference vary nonlinearly as shown in Figures 91 and 93 so that an arithmetic mean value is not correct. The logarithmic mean temperature difference, given by equation 10-25, is used.

$$\Delta t_{lm} = \frac{\Delta t_1 - \Delta t_2}{\ln(\Delta t_1 / \Delta t_2)} \qquad (10\text{-}25)(D.E.)$$

10.5.2.4 Overall heat transfer coefficient, U.

a. General

In heat exchanger design it is necessary to determine a reasonably correct value of the average or overall transfer coefficient, U. This is equivalent to integrating U_x with respect to the length x, and dividing this value by x, as shown by equation 10-26.

$$U = \frac{1}{x} \int_0^x U_x \, dx \qquad (10\text{-}26)$$

Obviously, this mathematical operation cannot be carried out analytically, since the variables which determine the value of the local coefficient, U_x, are not known functions of x. The integration must be done by an estimating or an approximating process. Table XXIV gives some typical values of overall heat transfer coefficient. These values are representative only, and should not be used except for very rough estimates.

TABLE XXIV. Typical Overall Heat Exchanger Coefficients

Type and Process	Btu./hr.-ft.^{2}-°F	Watts/ft^2-°C
Single pass crossflow exchanger (Air cooled by water) (Use outside tube area)	27.5	14.5
Large water-to-water exchanger Cooling 2500 gal./hr.	300	158
Oil heated by hot well water in 1/2-in. copper pipe (Use outside tube area)	75	40
Shell-and-tube air-to-water (Use face area of air side)	100 to 200	53 to 106

b. Smooth Surfaces

When the heat exchange surface or wall is smooth with parallel surfaces, and has a high thermal conductivity, the thermal resistance of the wall can be neglected, and the ratio Y_s/k_s can be treated as a film coefficient, h_s. All the elements of area are equal. Equation 10-23 becomes:

$$\frac{1}{U_x} = \frac{1}{h_{h_x}} + \frac{1}{h_{k_x}} + \frac{1}{h_{s_x}} \qquad (10\text{-}27)$$

Assuming scale on one side only, Table XXV gives some experimental values of heat transfer coefficient for scale deposits formed in certain kinds of water.

TABLE XXV. h_s for Various Kinds of Water (BTU/hr. - ft.^{2}-°F)
(From Reference 101)

Temperature of cooling water	52°C or less	Above 52°C
Temperature of heated fluids	Up to 115°C	115°C-204°C
Distilled water	2000	2000
Sea water	2000	1000
City, well, Great Lakes	1000	500
Hard (over 15 gm/gal.)	330	200

The local film coefficients are determined by the methods outlines in section 10.1. This requires an estimate of temperature gradients through the heat exchanger. Equation 10-26 can then be evaluated numerically, by averaging a set of values of U_x at selected values of x.

c. Finned Surfaces

It sometimes happens, as in air-to-water heat exchangers, that the film coefficient is much smaller on one side than on the other. In such cases it is desirable to use a larger area on the side with the lower coefficient to make the terms on the right side of equation 10-23 approximately equal. This is done by the use of fins. Since heat flows by conduction through the fin and thus establishes a temperature gradient, the heat transfer effectiveness per unit area becomes less toward the outer edge of the fin. This must be allowed for in calculating h. Since the actual area of the finned side is larger than that of the basic unfinned surface, there is an actual gain in heat transfer due to the fins. Ordinarily, only one side is finned. This may be either the hot or cold side. In gas-to-liquid exchangers the gas side usually requires fins. The following equations assume fins on the hot side, the arbitrary total area equal to the area of the unfinned side, the scale on the unfinned side only, as in air-to-water service. The following theoretical treatment is given in Reference 54.

$$\frac{1}{UA} = \frac{1}{\eta_o h_h A_f} + \frac{1}{h_k A} + \frac{1}{h_s A} \qquad (10\text{-}28)(D.E.)$$

where:

η_o = the overall surface effectiveness
A_f = area of finned surface

The overall surface effectiveness is found by equation 10-29.

$$\eta_o = 1 - \frac{A_f}{A}(1 - \eta_f) \qquad (10\text{-}29)(D.E.)$$

The fin effectiveness factor η_f is a function of fin geometry and is given in Figure 94.

The quantity m in Figure 94 is given by:

$$m = \sqrt{\frac{2h}{k\delta}} \quad \text{for thin sheet fins}$$

$$m = \sqrt{\frac{4h}{kd}} \quad \text{for circular pin-fins}$$

where:

h = average film coefficient, watt/ft.^{2}-°C
k = thermal conductivity of the fin material watt-ft./ft.^{2}-°C
δ = fin thickness, ft.
d = diameter of pins in pin-fin surface, ft.

d. "Tube-in-Strip" Panels (See Figure 87)

TABLE XXVI. Required Surface (both sides) of the Panels in Square Feet for Various Tube Sizes

Tube Inside Dia., Inches	Panel Surface Area in Square Feet per Foot Length of Tube
1/4	.0655
3/8	.0982
1/2	.1309
5/8	.1636
3/4	.1964
7/8	.2291
1	.2618
1 - 1/8	.2945
1 - 1/4	.3273
1 - 1/2	.3927
2	.5236

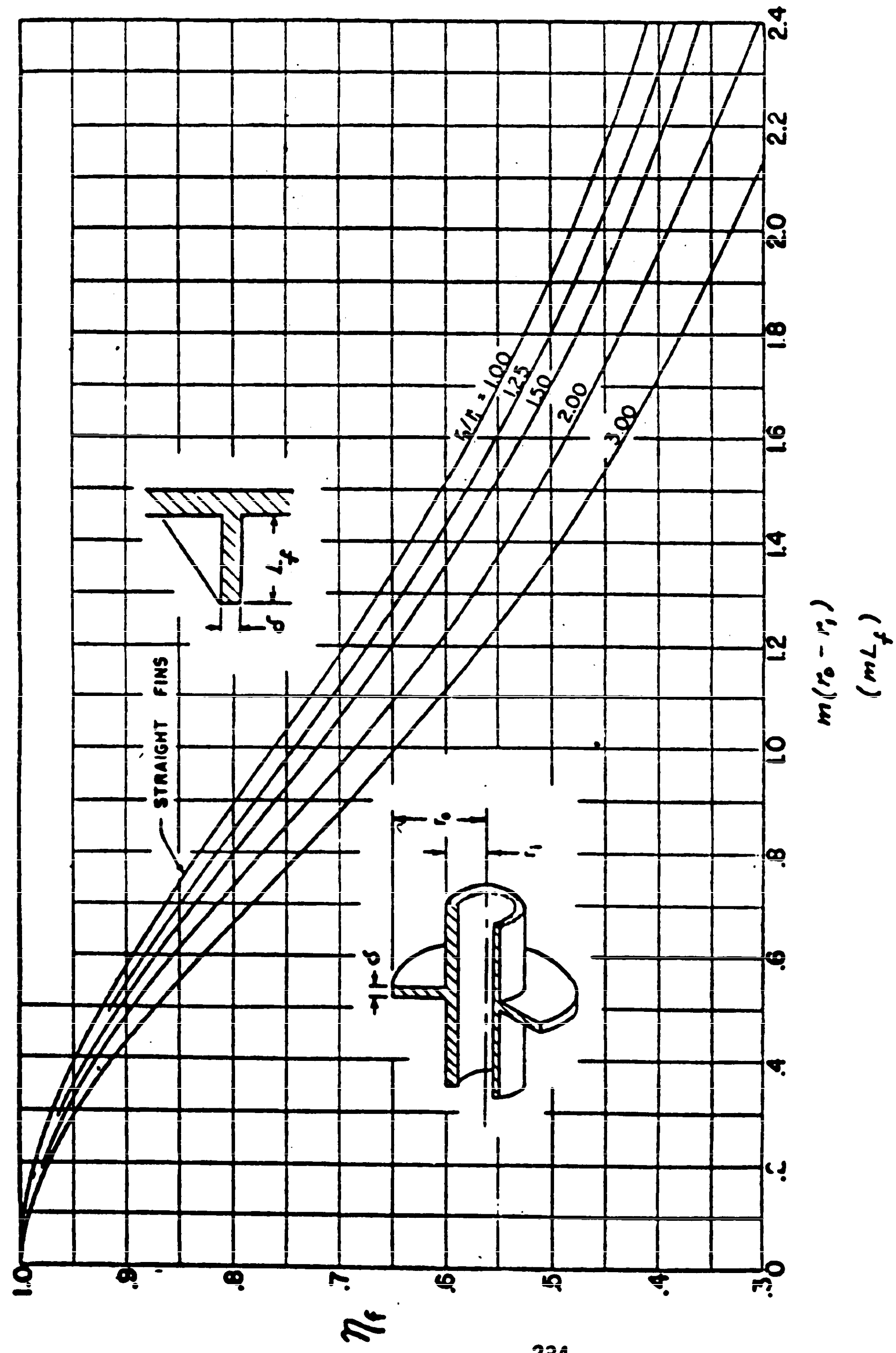


FIGURE 94. Fin Effectiveness, Circular and Straight Fins

Heat exchanger panels with internal tubes are designed in the same way as shell-and-tube exchangers of similar configuration. Table XXVI gives panel surface area for different sizes of tubes. These panels usually have liquid inside the tubes and gas outside. The outside area is larger than the inside tube area, to take care of the differences in film coefficients. Since the liquid film coefficient is large, the shape of cross section of the liquid passages is not very important and the hydraulic diamter can be used in Table XXVI .

Note: Hydraulic diameter is defined as four fimes the cross section area divided by the perimeter.

10.5.3 Heat exchanger selection.

10.5.3.1 General. The usual design problem involves the transfer of heat at a specified rate from a specified hot fluid to a specified cold fluid, with a minimum pumping power and in a minimum space or a space of specified size and shape. Entrance and exit temperatures of the hot fluid and entrance of the cold fluid are usually stipulated.

The fluid selection depends on several factors, including ambient conditions, type of service, availability, and heat transfer effectiveness. Fluid selection is discussed in section 10.7.

Other types of problems may arise. For example, it may be required to determine the performance of a given heat exchanger with different fluids, or with different velocities of one or both fluid. Problems of this type are best solved by actual test, but estimates may be made by use of the equations given.

Due to the large number of variables, heat exchanger design is a cut-and-try process, and a design is not ordinarily worked out directly from first principles. Rather, tabulated data resulting from experience are used, and the effects of changes in dimensions and operating conditions are calculated. This section presents equations by means of which these calculations can be made, and discusses the effects of design parameters.

This handbook is not intended to serve as a text on the design of heat exchangers. The intended purpose is to give electronic engineers sufficient knowledge of their design to enable them to prepare reasonable heat exchanger specifications, and to discuss the requirements with heat exchanger specialists.

10.5.3.2 The basic equations. For any heat exchanger, three basic equations can be written; one heat balance for the hot fluid, and one for the cold fluid, and one for the heat exchange equation. The hot fluid heat balance is:

$$q = m_h c_{p_h} (t_{h_1} - t_{h_2}) = c_h \Delta t_h \qquad (10\text{-}30)$$

Similarly, the heat balance for the cold fluid is:

$$q = m_k c_{pk} (t_{k_2} - t_{k_1}) = C_k \Delta t_k \qquad (10\text{-}31)$$

where:

q = heat transfer rate, watts
m = weight flow rate, lb./min.
c_p = specific heat, watt-min./lb.-°C
C = mc_p = capacity rate, watt/°C
Δt = overall temperature change of fluid, °C
subscript h refers to hot fluid
subscript k refers to cold fluid

The heat exchange equation is:

$$q = UA\,\Delta t_m \qquad (10\text{-}32)$$

where:

U = overall heat transfer coefficient, watts/ft.^{2}-°C
A = effective area of heat exchange surface, ft.^{2}
Δt_m = mean effective temperature difference, °C

10.5.3.3 The heat balance. The heat transfer rate, q, must be the same quantity in equations 10-30, 31, and 32 if heat losses from hot and cold sides to the external environment of the heat exchangers are neglected. Ordinarily, the heat transfer rate and the entrance and exit temperatures of the hot fluid are established as requirements. Also, the cold fluid and its entrance temperature will usually be known. The cold fluid exit temperature and the required flow rate can be estimated. The cold fluid exit temperature should be high, but it is controlled by the hot fluid exit temperature as shown in Figures 91 and 92.

Equation 10-30 aids in selecting a suitable hot fluid if a freedom of choice is allowed. If heat transfer rate and temperatures are specified, the capacity rate, C_h, is determined.

10.5.3.4 The heat exchange equation. In equation 10-32 only the heat transfer rate, q, will ordinarily be known with certainty. The effective temperature difference Δt_m is a complicated function of the fluid entrance and exit temperatures. The electronic designer cannot be expected to compute the heat transfer coefficient, U, from heat exchanger geometry. (Reference 101) Rather, he will make use of tabulated data such as those presented in Table XXIV . The effective area A is a measure of size of heat exchanger when the type or geometry has been decided.

10.5.3.5 Air-to-water heat exchangers.

a. Optimum Conditions

The film coefficient is much larger on the water side than on the air side. Consequently, the air side should be some form of extended surface. Parallel flow or counter flow arrangements usually use tubes with external longitudinal fins. Crossflow exchangers are of the fin-tube or plate-fin configuration.

Turbulence should be full developed to ensure effective heat transfer. The Reynolds Number should be about 10,000 on the air side and from 2300 to 4000 on the water side. Turbulators, in the form of helically twisted partitions, are sometimes used inside the water tubes. Staggered tube arrangements of suitable spacing increase the air stream turbulence.

High air velocity may cause objectionable noise, depending on the tube and fin arrangement. In-line tube banks whistle at a Reynolds Number greater than 9000. With flattened tubes whistling may be encountered at a Reynolds Number of 5000. If objectionable noise develops, its effect may be reduced by the use of sound absorbers. (See Reference 55, chapter 41) Increasing air velocity increases the heat transfer coefficient, but also increases the required blower power. It is recommended that heat exchangers be designed for a maximum air velocity of 800 ft./min., unless stringent space requirements outweigh the objections mentioned above.

Heat exchanger selection is not completed until the pressure drop and pumping power have been determined. Methods of calculating pressure drop are explained in section 9.3. The pumping power is given by:

$$p = 0.0226 \frac{m\Delta p}{\rho} \qquad (10\text{-}33)(D.E.)$$

where:

Δp = pressure drop, lb./ft.^{2}
ρ = density, lb./ft.^{3}
p = pumping power, watts
m = mass flow rate

b. Estimating Heat Exchanger Dimensions

Heat exchanger design is a converging process. The following example demonstrates a method of estimating the size of a compact air-to-water heat exchanger.

Sample Problem 1

An air flow of 2 lb./min. is to be cooled from 100°C to 40°C, using fresh water entering at 30°C.

c_{p_h} = 0.24 BTU/lb.-°F = 0.24 x 31.6 = 7.58 watt-min./lb.-°C

The hot air heat balance is, by equation 10-30, q = 2 x 7.58 x 60 = 910. watts

The water exit temperature in a crossflow exchanger is less than the air exit temperature. (See Figure 92) Assume a water exit temperature of 34°C.

c_{p_k} = 1 BTU/lb.-°F (for H_2O) = 31.6 watt-min./lb.°C

The water heat balance is, by equation 10-31, 910 watts = $m_k c_{p_k} \Delta t_k$ = m_k x 31.6 x 4

m_k = 7.2 lb./min.; Q = 0.87 gal./min.

Assuming a 1/2-in. diameter tube, the water velocity is given by:

$$m = \rho AV_2$$
$$A = \pi(0.5)^2/(4)(144) = 0.00136 \text{ ft}^2$$
$$\rho = 62 \text{ lb/ft.}^3$$
$$V = \frac{7.2}{62 \times 0.00136} = 85.4 \text{ ft./min.} = 5120 \text{ ft./hr.}$$

The Reynolds Number for the water side is now checked.

$$\mu = 2 \text{ lb./hr.-ft.}$$
$$Re = \frac{62 \times 5120 \times 0.5}{2 \times 12} = 6613$$

Since this is much higher than the recommended value, a 3/4 in. diameter tube will be used, resulting in a Reynolds Number of 4400. This is slightly above the recommended value.

Assuming an entering air velocity of 400 ft./min., the entrance cross section area is given by:

$$2(\text{lb./min.}) = 0.06 \ (\text{lb./ft.}^3) \times A \times 400$$
$$A = 0.0834 \text{ ft.}^2 = 12 \text{ in.}^2$$

An entrance of 1.5 x 8 in. would be satisfactory.

The hydraulic diameter is 4 x 12/19 = 2.52 in. = 0.21 ft.
The air side Reynolds Number is:

$$Re = \frac{0.66 \times 400 \times 60 \times 0.21}{0.05} = 6050$$

This is lower than the recommended value, but, since it is computed for the entrance section only, the local Reynolds Number at the tube and fin surfaces will probably be about 10,000.

The intended configuration is shown in Figure 95. It is a 3/4 in. tube, looped back and forth through plate fins. From Table XXIV for a typical heat exchanger of this type, a coefficient of 14.5 watts/°C-ft.^{2} based on outside tube area is a reasonable estimate. The logarithmic mean temperature difference is found by equation 10-25

$$\Delta t_{lm} = \frac{(100 - 30) - (40 - 34)}{\ln\left(\frac{100 - 30}{40 - 34}\right)} = \frac{64}{\ln 11.67}$$
$$= \frac{64}{2.46} = 26^\circ C$$

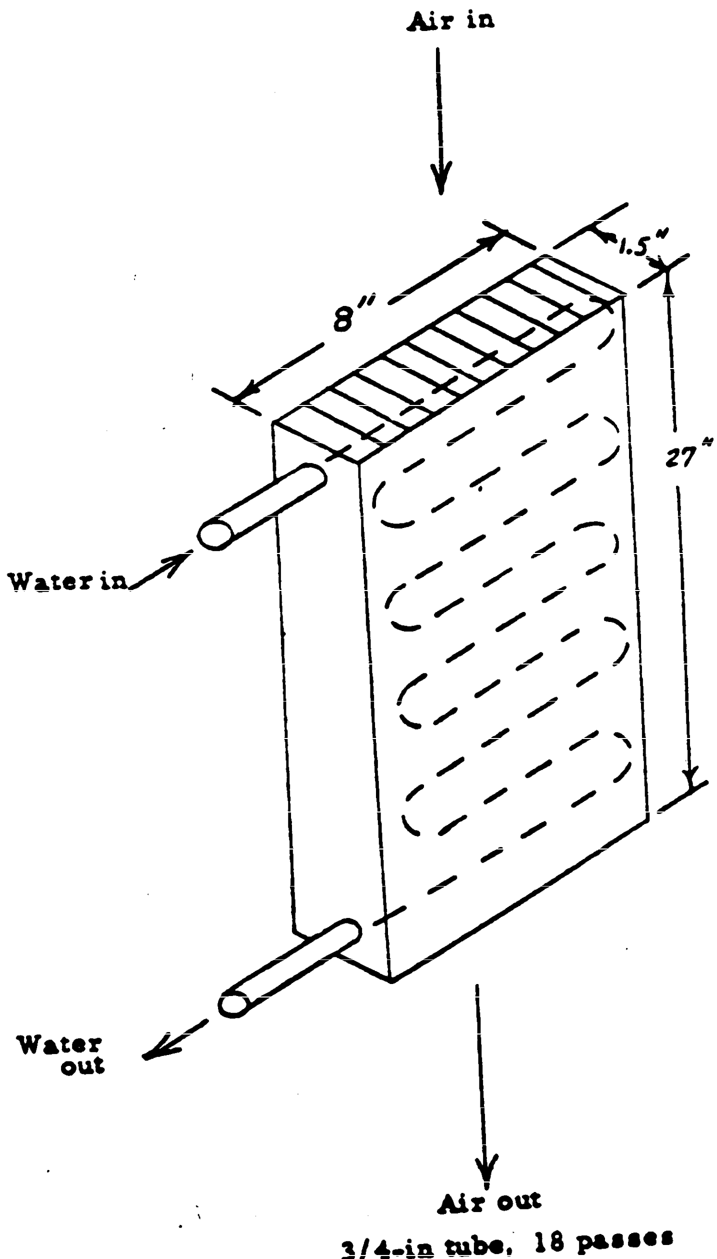


FIGURE 95. Tentative Core Design of Heat Exchanger

The outside tube area = $\frac{915}{14.5 \times 26}$ = 2.43 ft.^{2}. The area of one foot of 3/4 in. tube = 0.75 π/12 = 0.197 ft.^{2}. Length of tube required = $\frac{2.43}{0.197}$ = 12.3 ft.

In the 8-in. exchanger dimension, about 18 tube passes are required. Allowing 1.5 in. per tube pass, a depth of 27 in. is required. The exchanger will thus measure about 1.5 x 8 x 27 in., as shown in Figure 95

10.6 Cooling systems and fluid system components.

10.6.1 Definition of system-airborne, shipboard.

10.6.1.1 Shipboard cooling system interfaces. Figure 96 - (a) is a much simplified diagram of a ship's heat transfer loop for shipboard water cooling. A ship's system heat exchanger uses sea water to cool and regulate a fresh water cooling supply system. This fresh water coolant is supplied at 40°C (105°F) at the electronic equipment location, at a flow rate of 1.4 gallons per minute per kilowatt load based on a 2.8°C temperature rise in the water. NAVSHIPS specifications require that the water pressure drop through the equipment shall not exceed 10 lbs. per square inch. Thus, a system dissipating 50 kilowatts would require a circulation of 70 gallons per minute of fresh water, unless a larger temperature rise is permissable. MIL-STD-1399 - "Interface Standard for Shipboard Systems" is pertinent.

System A of Figure 96 (a) uses the fresh water coolant in the form of a cold plate indirect cooled liquid system with conduction paths from the dissipating elements to the liquid loop. System B of Figure 96 (a) uses forced air cooling with the air in turn cooled by an air-to-water heat exchanger. The ship electronic system interface may be at section 1-1 or section 2-2 . An interface at section 1-1 implies that the air-to-water heat exchanger is an integral part of the electronic system; that is, the electronic system interfaces directly with the ships fresh water cooling supply. This is the preferred interface, since it isolates the electronic system cooling system from other compartment heat sources. The interface at section 2-2 implies that the electronic cooling air is exhausted into an external air cooler which includes an air to fresh water heat exchanger. The use of compartment air for cooling is not always advisable. This method was frequently used in the past, but leads to unreliable systems and uncomfortable quarters, particularly as older systems were upgraded and more electronic systems added within a compartment. It should be used only for relatively low dissipation systems, and even then, the method of temperature control must be coordinated with the ship air conditioning system; the temperature control method selected must be compatible with the capability of the ship's system to accommodate changes in flow resistance and heat dissipation.

Once again, for emphasis, an electronic system cooling design utilizing the ships fresh water coolant supply is the preferred method.

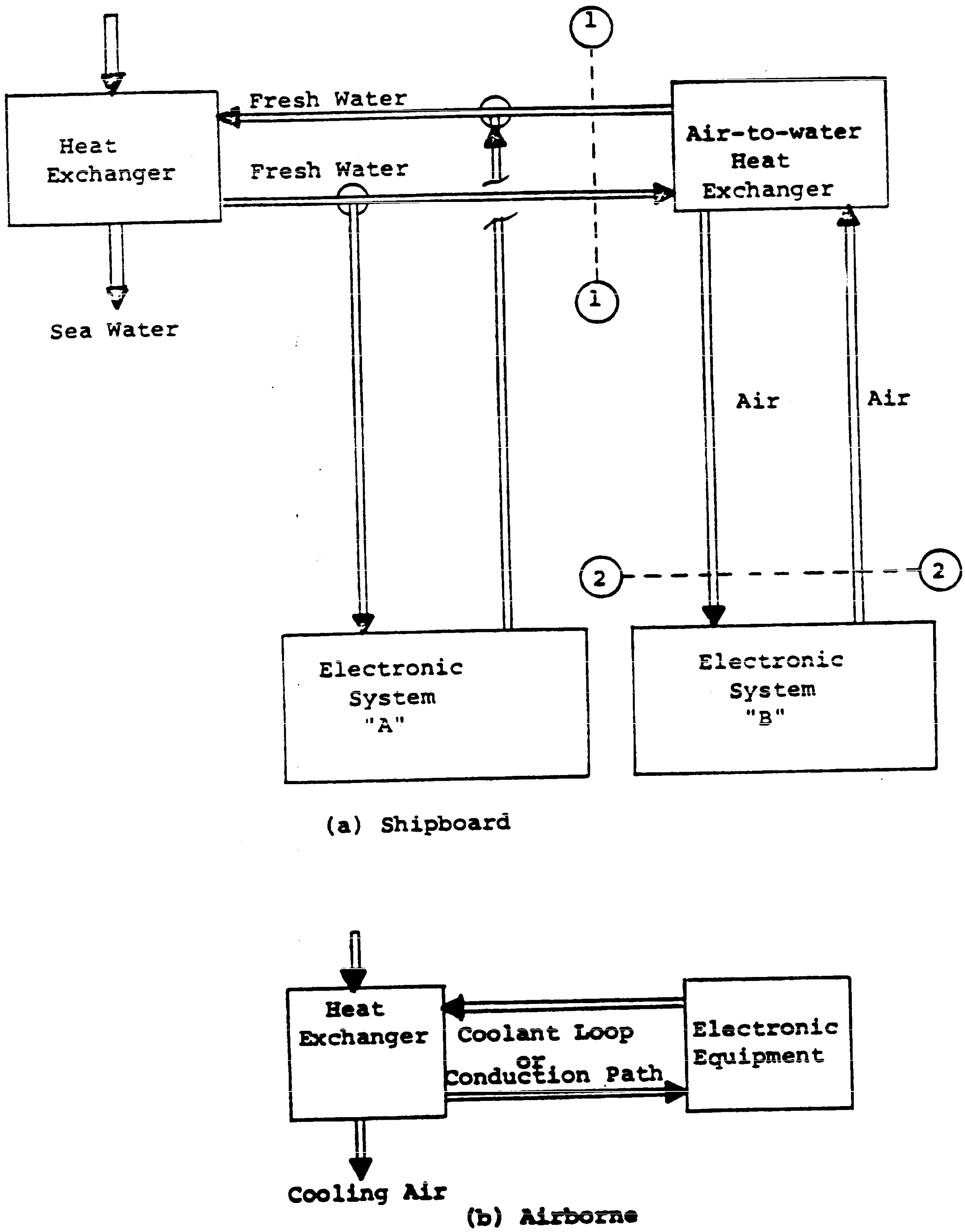


FIGURE 96. Heat Transfer Systems

The following advantages are offered by a ships fresh water cooling system:

1. The heat is readily transferred to an ideal ultimate sink, i.e., sea water. Water is a superior coolant compared to air.

2. The fresh water temperature is readily controlled within desired temperature limits.

3. The system can be more reliable, and less expensive than air conditioning or refrigeration systems.

4. All of the heat produced by the electronics is not rejected into the equipment spaces. Thus, the air conditioning systems need only supply personnel comfort loads plus equipment heat leakage.

5. The size and weight of a water cooling system is much less than that of an air cooled system. Small easily routed pipes are superior to large air ducts.

6. Parts temperatures are usually lower, since low thermal resistances can be achieved, especially with water cooled cold plates.

7. Thermal fluctuations and temperature cycling are minimized if water cooled cold plates are used.

The following disadvantages are offered by a ships fresh water cooling system:

1. The fresh water must be maintained at a high purity level to avoid buildup of scale and corrosion. Deinonizers, filters, and distilled makeup water must be used and properly maintained. Numerous difficulties have occurred on ships due to inadequate maintenance of the fresh water cooling system.

2. The sea water to fresh water heat exchangers must be clean and properly maintained. A single passage of the ship from sea water to fresh water (Panama Canal, for example) can result in fouling of the sea water side of the exchanger with mud, dead barnacles, and other debris. Again, poor maintenance has caused difficulty on existing ships.

3. Because the fresh water temperature is usually much lower than that of the air in an equipment, the water temperature must be controlled so that condensation of moisture will not occur. Usually the fresh water temperature is controlled at 105°F.

4. Because the fresh water temperature must be relatively high to avoid condensation of moisture, some of the heat dissipated by the electronics will escape into the surrounding space. With typical shipboard equipment only 75 to 80% of the total dissipated heat will be transferred into the water. If large high power water cooled heat sources (klystrons, for example) are used, then a larger percentage of the heat will go into the water.

10.6.1.2 Aircraft. The interface between an aircraft and avionic equipment generally offers fewer alternatives, and is more critical than the shipboard interface due to the more extreme environments encountered and the limited supply of primary (aircraft) coolant. Aircraft cooling air contains relatively great quantities of contaminants and impurities, so that is should not be used directly as a coolant for electronics. Consequently, a heat exchanger is required. The choice depends primarily on weight restrictions and allowable temperature rises. Liquid cooling systems for airborne equipment must be carefully evaluated in light of the stringent weight limitations imposed. Reference 56 indicates that with a system dissipation of 600 watts or less, air cooled systems are generally lighter than liquid cooled systems, but as total power dissipation increases above this figure, the weight of the air cooled system rises very rapidly with respect to liquid cooled systems due to the relatively massive air cooled heat exchangers required.

10.6.2 Control.

10.6.2.1 General. Liquid cooling systems are designed for the severest conditions anticipated for high unit heat concentrations and operating environments. When liquid cooling systems are used with varying heat loads or environments the capacity of the system can be reduced as a function of the cooling demand for increased efficiency. This will result in maximum system operating economy, but moreover, it can reduce temperature variations as the load or environment changes. As discussed in chapter 5, although the effects of thermal cycling on reliability have not been as well substantiated as have the effects of high temperature, nevertheless, thermal cycling has been shown to have an adverse effect on reliability, probably through the mechanism of bending fatigue failure due to repeated expansion and contraction. A regulated cooling flow system which minimizes thermal cycling will therefore, lead to improved reliability of components.

In immersion-type direct liquid cooling, the unit is relatively inflexible, and regulation of cooling system capacity is not possible. For forced direct liquid cooling or indirect liquid cooling, regulation of the cooling system capacity is possible by means of flow control, and is recommended for economy and reliability.

Regulation is accomplished by varying the flow rate (and thereby the temperature) with a valve or similar device in response to a control signal generated by a temperature sensing element. The degree of control depends on the requirement of the most temperature sensitive part or parts in the assembly. Heaters for the coolant are sometimes provided for exposed systems or for systems having extremely sensitive components where local operating temperatures must be held within close limits.

Liquid system control may be employed at the heat exchanger, to maintain a constant temperature of the fluid at the heat exchanger outlet regardless of load, or at the electronic system, to regulate the flow in response to a sensed internal temperature, or both. (See Figure 97) Controls may be generally categorized as intermittent, bypass, or continuous (also called proportional). (See Figure 98)

10.6.2.2 Intermittent control. The simplest form of cooling rate control is the intermittent type. This function is provided by a temperature sensing device located in the electronic equipment, which provides an on-off signal for the circulating pump, or for a line solenoid valve. If the control is applied to a valve or to valves in parallel, a relief valve must be provided for the pump for closed line conditions. Because of the cyclic variations in system pressure and/or electrical line load, this system is generally applicable to very simple systems only. (A thermostatic valve combines the functions of sensor and control in simple systems).

10.6.2.3 Bypass control. Improved temperature regulation can be obtained by allowing the pump to run continuously, and utilizing a temperature controlled bypass valve connected across the heat exchanger. The pump circulates liquid continuously through the electronic equipment, while the valve cycles the flow of coolant through the heat exchanger to control the fluid temperature within bounds set by the design and sensor.

10.6.2.4 Continuous control. Even more precise control may be obtained by use of a modulating or throttling valve in place of the on-off solenoid type valve. A modulating valve will distribute the flow between a bypass line and the heat exchanger line in proportion to a signal developed at the sensor. Mixing of the bypassed fluid with that passed through the heat exchanger will result in continuously controlled fluid temperature entering the electronics. The sensor used must provide a continuously varying signal, proportional to the monitored temperature, rather than a single on/off signal. Thermistors and resistance element sensors provide this type of signal, but suitable amplifiers must be provided. Signals from proportional sensors may be used, not only to activate modulating valves, but also to drive a temperature indicator for monitoring purposes, or to provide auxiliary signals. For example, if the temperature should continue to rise even with the valves wide open due to some system malfunction, a warning signal or electronic system shutdown signal may be provided from this same control system.

10.6.2.5 Considerations regarding monitored temperature. All thermal aspects in the vicinity of the temperature sensor must be carefully considered. The exact location and setting of the sensor should be determined by a systems test, rather than being based entirely on analytical procedures. Adequate margin between the control setting and the maximum allowable system temperature should be provided to allow for system variations and time degradation effects. Very often, the sensor cannot be placed directly on the component to be monitored, but must be placed adjacent to or downstream from the component. In this case, any thermal resistance and any time lag due to thermal capacity between the component and the sensor must be accounted for in determining the control setting.

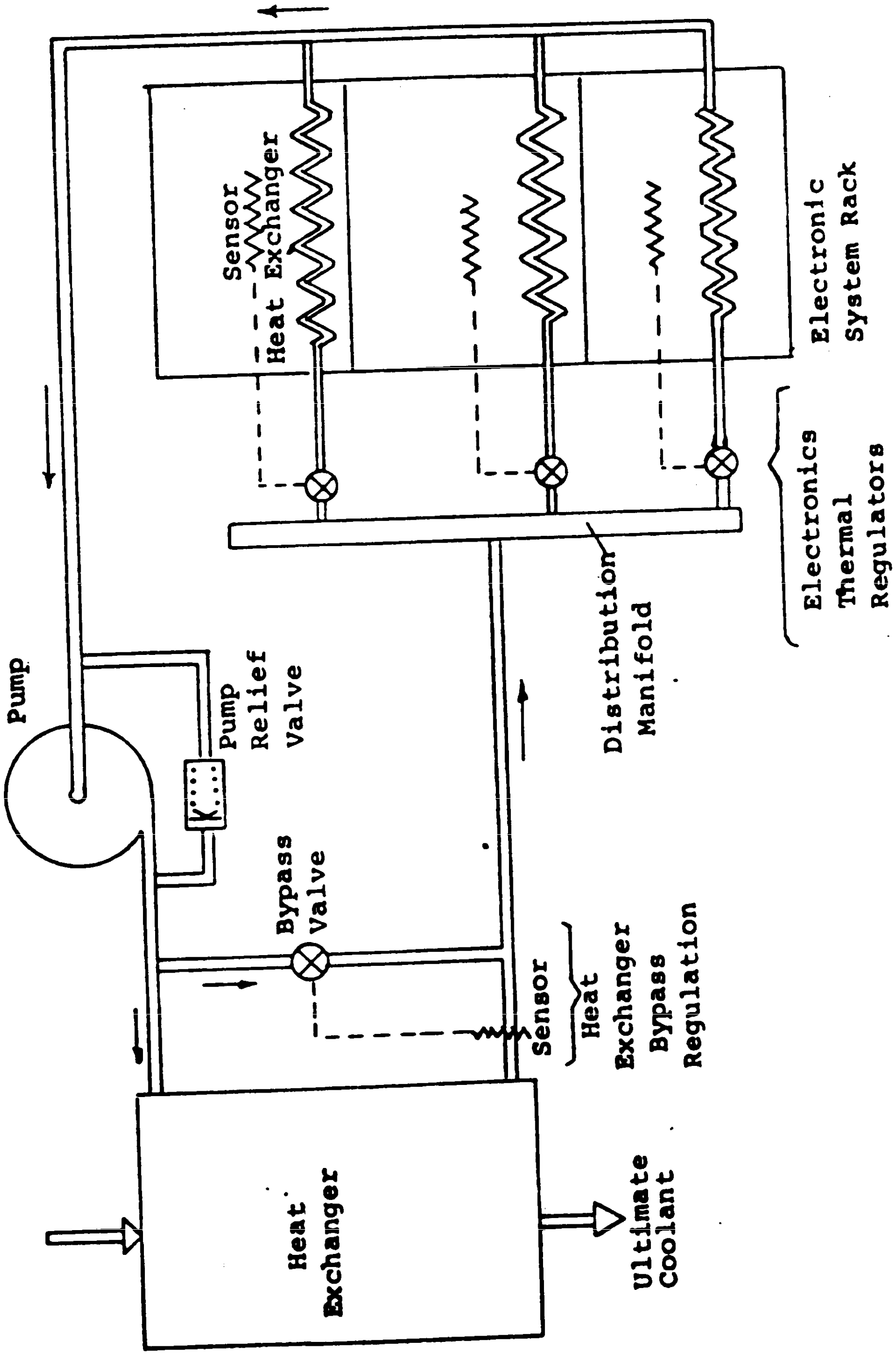


FIGURE 97. Liquid Cooling System Controls

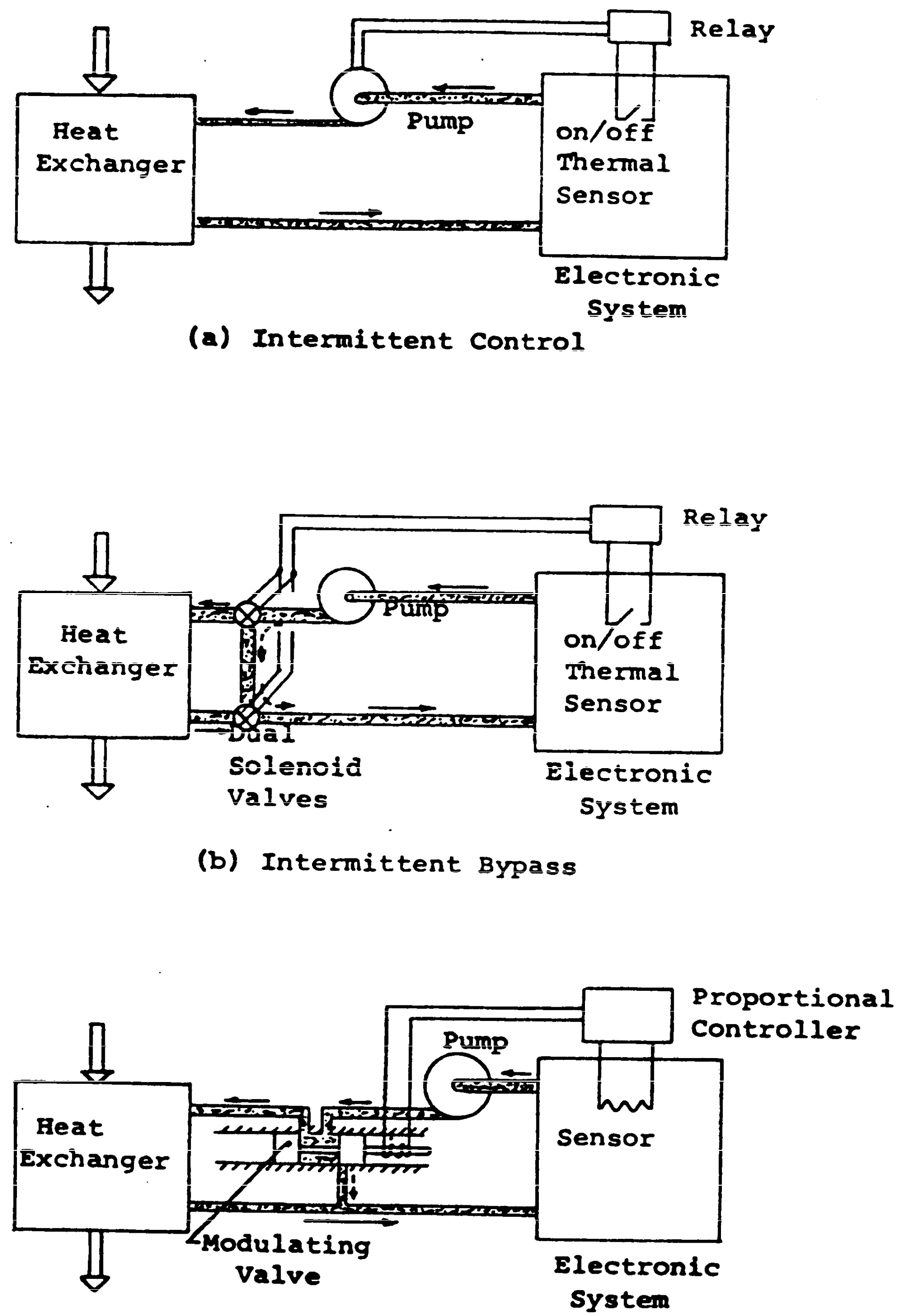


FIGURE 98. Types of Control

On/off sensors generally have two set temperatures, that is, the "on" temperature is somewhat higher than the "off" temperature. The on/off differential may range from a fraction of a degree for precise sensors to 10°C or more for cruder, industrial type sensors. This differential will in itself cause cyclic variations in temperature, with the cycle period dependent upon the rates of heat generation and removal and thermal capacitance of the system.

Proportional systems should be designed to prevent the type of oscillation about the control point known as "hunting." Generally, thermal time constants are so long as to prevent this type of operation, but its possibility should be recognized. One method of eliminating hunting is to provide a "dead band" in the proportioning signal.

If the control valve is remotely located from the sensor, there may be an appreciable time delay between application of the control signal and sensing the full effects of the increased coolant flow. This delay cannot be easily predicted, since it is a function of the coolant mass flow rate, coolant specific heat, heat generation, and heat rejection rates, and system thermal capacity.

Forced convection systems may incorporate flow sensing devices to prevent electrical operation when the pump (or blower) is inoperative. These safety devices may be used alone or in conjunction with thermal cutoff safety devices.

Forced convection systems frequently require that power for the cooling system prime mover (pump or blower) be applied independently of the system "on/off" switch. This allows continued circulation of the coolant after system shutdown, to prevent local "hotspots" due to thermal diffusion from high dissipation devices. A temperature sensing element is required to de-energize the coolant system when a suitable low temperature is reached.

10.6.3 Pumps and their selection.

10.6.3.1 General. The purpose of the pump in a liquid cooling system is to circulate the required rate of coolant flow against the total fluid friction heat of the cooling circuit. Variations in friction heat with variations in temperature or pressure are generally not as significant in liquid cooling systems as they are in air cooling systems, due to the relative stability of liquid properties as compared with air.

Pumps are almost always motor driven, with the electric motor matched to the pump to avoid overloading under any flow conditions, from blocked flow to open discharge. Pumps can be overloaded and damaged if the viscosity of the coolant becomes too high under low temperature starting conditions. With such coolant, all the coolant must be preheated with electrical heaters.

10.6.3.2 Pump selection. A variety of pump designs are available for liquid cooled systems. The more common categories include reciprocating, gear, vane, centrifugal, and axial flow types.

Reciprocating pumps are constant displacement devices, characterized by the capability of developing high system pressures at a high efficiency, but with relatively low volume flow rates for a given weight and volume. They are widely employed by hydraulic control systems, but because of their low flow rates, they are almost never applied to liquid cooling systems.

Vane pumps are similar in characteristics to reciprocating pumps, except that vane pumps tend toward slightly lower pressures and higher flow rates. Vane pumps are lower in cost than reciprocating pumps, with generally a slightly lower efficiency. Centrifugal action vane pumps require a minimum speed of about 600 RPM to insure contact between the vane tips and the cam ring to avoid excessive leakage. Like reciprocating pumps, the use of vane pumps is limited in liquid cooling applications. Both reciprocating and vane pumps are usually self-priming and suitable for handling non-lubricating liquids.

Gear pumps are also positive displacement pumps, characterized by low cost and a minimum of moving parts. Gear pumps are occasionally used in liquid cooling applications.

Centrifugal and axial flow pumps have a high capacity and moderate head characteristics. Since these pumps are not constant displacement devices, the flow volume may be controlled by a valve or variable orifice without developing excessive pressures or overloading the motor. Axial flow pumps approach fan-like characteristics with higher flow rates and lower heads than centrifugal pumps. These pumps can become air-bound, and must be primed. Priming can be accomplished by locating the pump below the liquid level so that the pump inlet is always flooded, or by using them in a fully closed system. Cooling systems should be pressurized to assure a positive liquid pressure at the impeller inlet, so as to avoid air leakage into the system at the pump shaft seal.

For the selection of a pump for a given cooling system, it is necessary to consider the following:

(1) The physical and chemical characteristics of the coolant liquid being used, such as specific gravity, viscosity, temperature, solids in suspension, abrasive material, thermal stability, and the corrosive and solubility effects on materials used in pump construction.

(2) The state of the fluid at the pump inlet, whether it floods the inlet, is full of air bubbles, whether the pump must prime itself, and, if so, the priming lift.

(3) The pump characteristics desired in terms of delivery required, differential pressure, inlet pressure, discharge pressure, the sum of the differential pressures across the electronic subassemblies, the heat exchanger, the line loss, and the duty cycle.

(4) The voltage, phase, and frequency desired for the pump motor, and environmental conditions, such as dust, moisture, fumes, and fire hazards under which it must operate.

(5) The electrical and mechanical noise characteristics of the pump motor.

(6) Special application priorities, such as minimum weight for aircraft, minimum power (efficient pump required), or maintenance philosophy.

10.6.4 Expansion spaces or tanks. In a totally enclosed liquid cooling system, provision must be made for expansion since practically all fluids increase in volume as temperature increases. Volumetric expansion rates of typical liquid coolants range from about .05%/°C to 0.2%/°C. With reference to a 25°C normal "ambient" temperature, volumetric changes of ± 16% might be experienced in an operating "ambient" temperature of -55°C to +55°C.. High temperature expansion must be based on the maximum coolant temperature, not the maximum "ambient" temperature. Expansion spaces should be sized so as to provide an air (or vapor) cushion even at the maximum coolant temperature anticipated. The primary object of the expansion space is to allow for the increase of fluid volume with temperature, but it may also provide ancillary functions, such as removal of air from the coolant, and isolation of hydraulic shocks due to, for example, the rapid closing of a valve. When subsystems, or "drawers," have integral cooling systems, each subsystem must be provided with an expansion volume. However, when a number of subsystems are served by the same cooling system, a single expansion device installed in the cold leg of the coolant flow loop is more appropriate, less expensive, and lighter in weight.

10.6.5 Piping and valving. Once the required coolant flow volume has been established, the pump/piping system must be sized such that the heat developed by the pump at the required flow rate equals or exceeds the friction heat developed at that flow rate in the piping, valves, heat exchanger, and electronic assembly system. Aircraft installations emphasize minimum pipe size, combined with high strength thin-wall tubing and welded joints, to minimize the line system weight. Shipboard and ground based equipment is generally more tolerant of weight, and larger, lower cost lines may be used. Welded, brazed, or soldered connections are still preferred to screw type where maintenance is not unduly sacrificed. Piping material must be carefully specified to avoid corrosion effects, either directly from the coolant, or more commonly, as the result of galvanic corrosion where different materials are in contact in the system. Water cooling systems shall be fabricated of basically high-purity copper materials. All materials used shall be electro-chemically compatable with high-purity copper. Materials shall be selected in accordance with MIL-W-21965B (SHIPS), "General Specification for Water Cooling of Shipboard Electronic Equipment", and with MIL-STD-1399 (NAVY), section 101, "Cooling Water for Support of Electronic Equipment". Section 10.6.6 contains additional discussion of corrosion effects.

For in-line valves, gate valves are preferred to globe valves because of their inherently lower pressure drop. The number of in-line valves should be kept to a minimum; their normal purpose is to permit servicing of an individual chassis without the need to drain the entire system. Drain valves should be provided at the lowest point of each leg of the system. Valves or pet-cocks should be installed at the uppermost point of each leg to facilitate drainage and to vent trapped air during system filling. Inlet and outlet connections of the quick disconnect, self-sealing type facilitate the removal of an individual unit without loss of coolant. Thus, draining of the unit and venting and refilling after reconnection are unnecessary during normal maintenance and repair procedures.

10.6.5.1 Filters. Filters are used to extract particulate matter from coolants to protect pump bearings and valve seats, and to prevent accumulation of flow restricting matter in bends, elbows, and fine passages. All throughflow (non-recirculating) systems should contain filters. Even sealed recirculating coolant systems are normally provided with filters, since all common coolants are subject to some form of sludge contamination over very long periods of time. Filters are generally of the replaceable cartridge type in liquid systems, and should be located for ease of maintenance. Very often, when electronic equipment is supplied for a large installation (for example, shipboard and many ground-land systems) the parent installation will provide a source of conditioned, filtered coolant. (See MIL-STD-1399)

10.6.5.2 Other components which may need to be supplied with liquid cooling systems include pressure regulators, air separators (often included as part of the expansion tank), sediment traps, and auxiliary heaters.

Warning systems in liquid cooling systems may use either (a) temperature sensing, wherein the temperature of the most critical component or components is monitored or (b) flow sensing, wherein the coolant flow presence is monitored. Temperature sensing is preferred, since it will react to malfunctions in the electronics, or overload conditions, as well as to loss of coolant. The sensor may activate an audible or visible warning (or both), or may actuate a relay to disable the electronic system or to reduce the system output to a lower level.

The heat loss in a piping system may be computed by the methods presented in section 9.3, since the loss computations presented therein are applicable to any fluid.

10.6.6 Corrosion effects. Corrosion is the chemical action of the environment on metals, resulting in their destruction or in the modification of their physical properties (usually detrimentally). Corrosion can be a serious problem in liquid cooling systems, but careful consideration in the design phase, coupled with an adequately planned and carried-out maintenance philosophy will minimize the problems and insure a satisfactory service life. The types of corrosion commonly encountered in liquid cooling systems are:

10.6.6.1 Atmospheric corrosion. Atmospheric corrosion is the reaction between oxygen and a metal, resulting in the formation of a thin oxide film on the surface of the metal. This film, left undisturbed, isolates the metal from the oxygen, preventing or greatly inhibiting further oxidation. Disturbance or removal of the film, however, will allow continued oxidation.

Copper, copper alloys, aluminum, and its alloys, and stainless steel form very thin, hard, impervious coatings. These coatings are, in fact, often enhanced by artificial methods for protective purposes, as in anodizing of aluminum or passivation of stainless steel. The oxide coating of iron, on the other hand, is much softer and cannot be relied upon for protection. While this form of corrosion has been termed atmospheric, it can also be present on the internal surface of a liquid cooled system, due to the presence of dissolved oxygen in the water or other coolant. This dissolved oxygen, along with the flow of coolant, can cause progressive interior deterioration (rusting) of iron pipes. This type of corrosion can be prevented by avoiding the use of iron, by suitable internal coatings for iron pipes, or by de-aerating to minimize oxygen content or adding inhibitors to the coolant.

Metal surfaces, such as piping, which are maintained at temperatures lower than the surrounding air, must be insulated to reduce the flow of heat and to prevent condensation. The insulating material should be impervious to moisture, and should be thick enough to keep the temperature of the outer surface of the insulation to a point slightly higher than the dew point of the surrounding vapor. Figure 99 gives the approximate thickness of insulation required to prevent condensation.

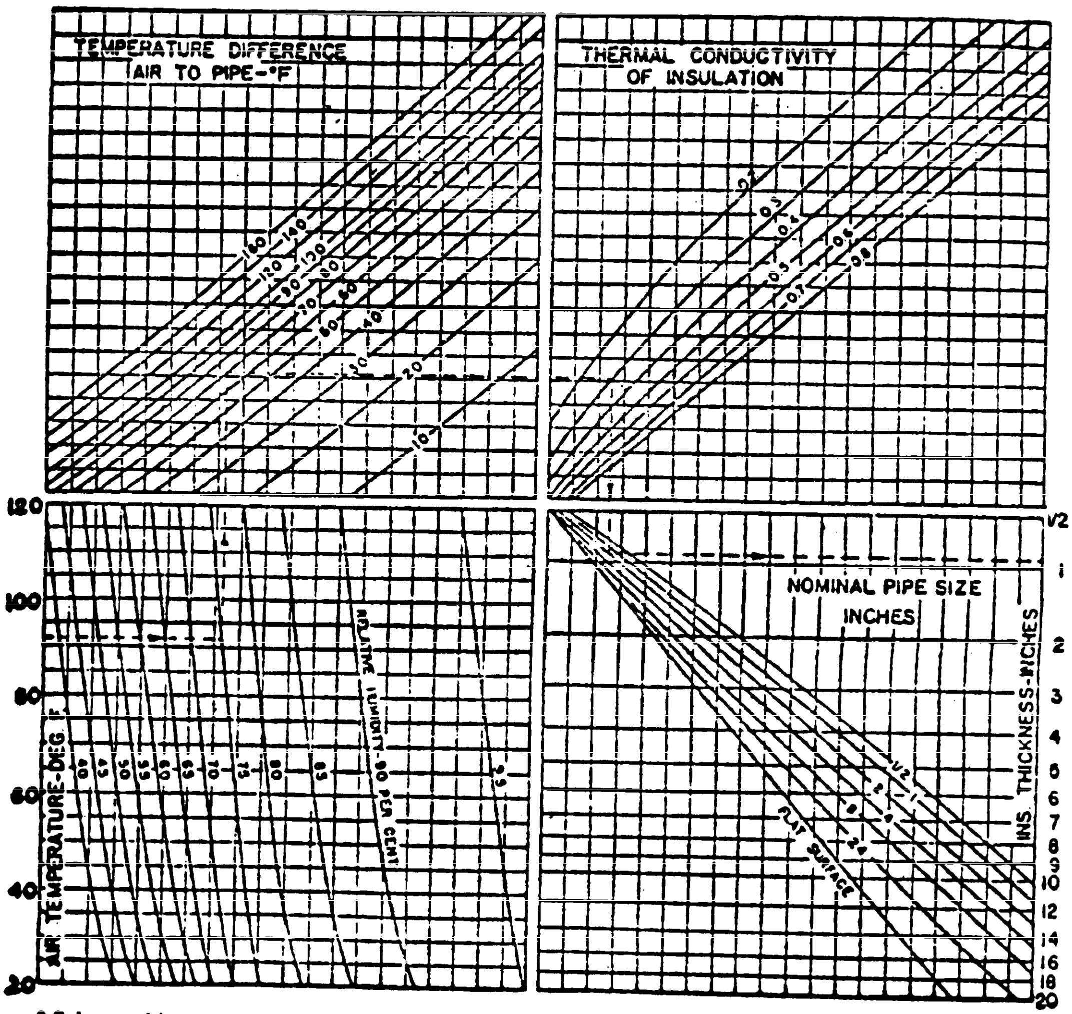
10.6.6.2 Galvanic corrosion. Galvanic corrosion is an electrochemical reaction brought about when two dissimilar metals are immersed in an electrolyte and an external contact is made. In essence, the dissimilar metals and the electrolyte form a miniature battery. The galvanic currents so induced will cause solution ionization of the less noble of the metals; if the galvanic potential is great enough, the less noble metal will eventually be eaten away.

The extent of the galvanic corrosion is dependent upon many factors, the most notable of which are the electrolyte involved and the relative dissimilarity of the metals forming the galvanic couple. Reference 57 gives a comprehensive chart indicating the effect of various couples in a sea water environment.

Galvanic corrosion can be prevented or minimized by a variety of methods. The surest method is to avoid the use of dissimilar metals in contact wherever possible. When other factors will allow it, an insulator placed between dissimilar metal surfaces will break the galvanic circuit. This insulator can often be an oxide of the base metal, or a paint or other protective coating. If dissimilar metals must be used, and use of insulators is not possible, the metals should be chosen as close together in the galvanic series as possible. The anode metal should be large with respect to the cathode, or more noble metal (the anode metal is the one which will suffer corrosive effects).

Differential aeration can also cause galvanic currents. If a small cavity or crack exists in the surface of a metal, oxygen cannot diffuse readily to the bottom of this crack, whereas there is an abundance of it at the surface. A current is produced between the aerated surface of the metal (cathodic) and the unaerated crevice which is anodic. Since the attack is concentrated is the small crevice, the rate of corrosion will be high. Such localized corrosion is insidious, and in designing cooling systems it is important to avoid such recesses.

10.6.6.3 Electrolytic corrosion. Electrolytic corrosion is similar in mechanism to galvanic corrosion, in that the corrosive action is initiated and sustained by an electric current. In electrolytic corrosion, however, the current is a direct result of the coolant contacting portions of the electronics at a high potential. This occurence should be avoided, if at all possible. All equipment should be provided with adequate electrical



* Solve problems as indicated by dotted line, entering chart at lower left hand scale

FIGURE 99. Insulation Thickness to Prevent Sweating
(From ASHVE Guide 1955)

bonding; piping systems should never be used as a ground return. In general, insulating non-metallic piping sections should be installed for electrical isolation between the equipment and shipboard cooling systems. Where the design makes it imperative for the coolant to directly contact areas of electrical potential, extreme precautions should be taken in the design to prevent both electrolytic corrosion and current leakage or shock hazards. (This might be a possibility in cooling high power tubes or power supplies).

10.6.6.4 Chemical corrosion. Most of the common metals corrode much more rapidly in an acid solution than they will in a neutral or slightly alkaline solution. Base metals such as iron, zinc, aluminum, and magnesium are very reactive with acid solutions.

A solution which is neither acid nor alkaline, therefore neutral, will have a pH value of 7. A cooling solution with a pH value between 7 and 9 (slightly alkaline) is satisfactory for cooling systems. The cooling medium should not be allowed to become excessively alkaline because metals such as aluminum and zinc, whose oxides are soluble in alkalies will corrode rapidly. Fresh water coolant on shipboard is normally properly treated by the ships cooling supply system.

NOTE: The pH value is the common logarithm of the reciprocal hydrogen ion concentration. In pure water this concentration is 10^{-7} mole per mole of water, i.e., 10^{-5} percent of the water is dissociated. Such water is considered to be chemically neutral. As hydrogen ion concentration decreases, acidity decreases and pH increases.

10.6.7 Maintenance aspects. Most water, unless chemically treated or distilled, will cause hard deposits to form on the inner surfaces of pipes and tubes. These deposits increase the friction factor and restrict the flow, and may eventually close the pipe completely. The nature and amount of hardness may be determined chemically and a suitable treatment will prevent, or at least greatly retard, this effect. Even distilled water is not completely safe. Piping and heat exchangers should be checked periodically for decrease in flow. Chemical treatment and mechanical cleaning may remove the salt deposits. NOTE: For shipboard systems, water treatment is taken care of by the ship designer, and properly treated water may be assumed to be available. (See MIL-STD-1399)

The designer of a liquid cooling system must consider maintenance aspects in the development of his equipment. Sufficient self sealing disconnects (or valves and disconnects) must be provided to ensure the ability to remove and repair individual subassemblies without draining the entire system. Replaceable components and modules mounted to cold plates should be removable without disturbing coolant seals. Direct liquid cooled modules should have drain provisions if they are repairable; small non-repairable (throwaway) modules may be totally sealed.

Water gauges or pressure indicators on expansion tanks can provide a means of checking liquid level. All sealed systems should be provided with some method of checking against leaks.

All accessory equipment, such as thermal devices, pressure regulators, automatic valves, and pumps, must, of course, be maintained in proper working condition.

10.7 Coolants.

10.7.1 Rating of heat transfer fluids.

10.7.1.1 General. A sound method of comparing the effectivenss of heat transfer fluids would be highly useful. However, heat transfer is such a complicated process, so many characteristics of materials are involved, and the effects of temperature on these characteristics vary so widely that no single rating criterion can be established.

It is possible, however, to establish specific criteria which are significant for certain specific operating conditions, based on those properties of the fluid which affect heat transfer. Such criteria are useful general guides, but are quantitatively valid only for the conditions for which such criteria are derived. After rating fluids by such criteria, the effects of other characteristics may be evaluated. Among these other characteristics are: freezing and boiling points, dielectric constant and loss factor, flash and ignition temperatures, toxicity, chemical stability and compatability, lubricity, availability, and cost.

10.7.1.2 Rating criteria. Several rating criteria, or figures of merit, have been published. Three useful formulas are given below. These figures of merit are not dimensionless; therefore, their numerical values depend on the units used. The same units must be used in computing values for each coolant which is to be compared. These figures of merit cannot be used for cross comparison between the different modes of operation, as between free and forced convection. For convenience, the terms used in these rating criteria are re-defined below.

ρ = mass density
C_p = specific heat at constant pressure
B = volumetric expansion coefficient
k = thermal conductivity
μ = dynamic viscosity

(1) For free convection, vigorous enough to attain a satisfactory degree of turbulence, fluids may be compared by relative values of the function:

$$n_1 = \left(\frac{\rho^2 g c_p B k^3}{\mu} \right)^{1/4} \qquad (10\text{-}34)$$

The rate of heat transfer is proportional to this function, for systems of the same geometry operating under the same conditions. Therefore, a good coolant has a large value of n_1.

(2) For forced convection at low Reynolds numbers, with laminar incompressible flow, fluids may be compared by relative values of the function:

$$\eta_2 = \frac{Pr^{0.4}\rho^{0.5}k}{\mu^{1.5}} \tag{10-35}$$

Since the ratio, rate of heat transfer/pumping power, is proportional to this function, all other conditions being the same, a good coolant has a large value of η_2.

(3) For forced convection with turbulent incompressible flow, the corresponding function is:

$$\eta_3 = \frac{Pr^{0.4}k}{\mu^{1.1}} \tag{10-36}$$

A good coolant has a large value of η_3.

10.7.1.3 Effectiveness of sample coolants. For laminar forced convection, Table XXVII gives computed figures of merit for some typical coolants. Since only relative values of the η_2 functions are significant, the units used for the parameters are not important. It is, of course, necessary always to express each parameter in the same unit. This computation is to be considered only as an example of the use of rating functions. A more detailed table of comparative values of coolant effectiveness is contained in the Appendix.

10.7.2 Selection of coolants.

10.7.2.1 Direct liquid coolants.

10.7.2.1.1 General. These liquids must be highly stable and nonvolatile at the selected operating temperature and must not undergo a change in state. When selecting a coolant, it is necessary to consider the change in the thermal and physical characteristics of the liquid over the entire operating temperature range as well as its chemical and electrical compatability with the metals and materials with which it comes in contact. With direct liquid cooling systesm, these properties include chemical inertness, dielectric constant, power factor, vaporization temperature, freezing temperature, flash point, vapor pressure, toxicity, coolant life, surface tension, and dielectric strength, as well as the rating criteria discussed above. In particular, a relatively high volumetric coefficient of expansion is actually desirable even though it requires means of accommodating the increased volume, because a change in liquid density is necessary to initiate convective flow of the coolant. Convection is, of course, the primary mode of heat transfer in liquid-immersed devices and experiments with coolants of relatively constant density (low expansion coefficient) show that large thermal gradients occur when the coolant is relatively motionless.

TABLE XXVII. Comparison of Coolants - Forced Convection Incompressible Flow

Fluid	Temperature	η_2
Water	200°F	2000.
Water	100	831.
Water - Methanol 68%	200	591.
Water - Ethylene glycol 62.5%	200	430.
DC550-112 Silicone or equivalent	200	1.38
DC550-112 Silicone or equivalent	100	0.392
Mineral oil	200	6.1
Mineral oil	100	0.895
Hydrogen	212	0.00101
Helium	212	.000228
Air	100	.0001
Air	500	.000037

There are a number of coolants which have been used in assemblies, power and communication transformers, chokes, capacitors, and other high voltage or high temperature parts. These fluids are, for the most part, hydrocarbons of one form or another and suffer from deficiencies such as high dielectric constant, molecular instability, high power factor (particularly at radio frequencies), and inability to withstand very low or very high temperatures.

Impurities cannot be tolerated in coolants used in liquid-filled assemblies. Even a small percentage of moisture can lead to electrolysis and rapid corrosion of wires and leads, especially those with DC circuits in excess of 25 volts. Extreme care must be used to prevent contamination of the coolant. The physical characteristics of coolants are presented in this section and in the Appendix.

10.7.2.1.2 Petroleum oils. Petroleum base oils can be used in liquid-cooled equipments whose "hot spot" temperatures do not exceed 175°C. The oils oxidize or rapidly decompose at higher temperatures and must be changed occasionally. Mineral oil exhibits lower electrical losses at high frequencies than the silicone fluids. Detailed fluid properties are listed in the Appendix.

10.7.2.1.3 Silicone fluids.

(a) Straight dimethyl fluids are known as the 200-series silicone fluids. It is not recommended that these fluids be operated at temperatures exceeding 150°C for optimum heat stability in direct liquid-cooled equipments. The 200-series fluids differ from each other only in viscosity. Viscosities ranging from .65 to 10 centistokes are available.

(b) The 500 and 700-series silicone fluids are blends of the dimethyl and phenolmethyl fluids. They are stable to 200°C for direct liquid cooling applications. These fluids are not available in as wide a range of viscosities as the 200-series fluids.

(c) Silicone fluids are superior to the hydrocarbons, especially in regard to their ability to operate at temperature extremes. High temperature operation is limited only by the cracking temperature of the fluid.

The dielectric constant and power factor are substantially constant up to at least 100 MC. The dielectric constant and the power factor decrease slightly with increasing temperature. In most instances these changes will be insignificant.

The thermal conductivity of silicone fluids is actually quite low, being intermediate between glass and air. For this reason, a rather low viscosity fluid is generally chosen so that circulation due to convection will be aided. Fifty centistoke silicone fluid has been used with considerable success.

The thermal expansion coefficient of all liquid dielectric materials is high. For example, DC-200 silicone fluid (or equivalent) increases in volume by 13% over a temperature range of from 25°C to 150°C. It is therefore, necessary to provide adequate space for expansion.

It has been reported that polystyrene and some of the related resins are attacked by chlorinated hydrocarbons. It has also been found that silicone fluids have an undesirable effect on some silicone protected parts. In general, however, silicone fluids are inert with respect to most commonly used electronic materials. Data can be obtained from the manufacturers.

10.7.2.1.4 <u>Silicate esters</u>. A fluid commonly used for electronic equipment cooling is the silicate ester (Coolanol or equivalent). This fluid has a useful temperature range of -50°C to 200°C. The manufacturer states that water is very slightly absorbed in Coolanol 45 (or equivalent) (0.08 weight per cent at 25°C) and that this amount does not affect the fluid. However, excessive amounts of water react with the fluid to form a precipitate or sludge. Closed and pressurized systems should be designed to prevent "breathing" of moisture-laden air and condensation. Although deterioration by moisture of the dielectric properties is not nearly as severe with Coolanol 45 (or equal) as with hydrocarbon oils, precautions to avoid water contamination should be taken when handling the fluid.

At room temperature, water quickly separates from Coolanol 45 (or equal) and forms a bottom layer. Water and sludge collecting at low points in the system indicate contamination. The manufacturer should be contacted for advice on the possibility and method of salvaging or decontaminating the fluid.

Generally speaking, Coolanol 45 (or equal) is compatible with all common materials with the exception of most silicones. Silicone elastomer, LS-53 (or equal) is completely compatible with the silicate ester. Probably the most widely used materials are Neoprene W (or equal), Poly FBS (or equal), and Buna N (or equal). Thermal limits of materials must be considered in any design.

Corrosion and oxidation data show that the silicate ester does not affect the common metals -- aluminum, iron, copper, silver alloys, brass, cadmium-plated steels, solders or brazing materials. The fluxes used in joining metals may affect the fluid, and these should be removed from any system which will contain a silicate ester. Detailed fluid properties are listed in the Appendix.

10.7.2.1.5 Freon (or equivalent). The Freon (or equal) products are organic compounds containing one to four carbon atoms and flourine. Chlorine, bromine, and hydrogen atoms may also be present. Their principal characteristics include non-flammability, a low level of toxicity, excellent thermal and chemical stability, high density, low boiling point, and low viscosity and surface tension. The Freons (or equivalent) are rarely used as direct liquid coolants because of their typically low boiling points (less than 100°C) at one atmosphere pressure. They are more extensively used as refrigerants, vapor cycle coolants, and vaporization cooling fliuds (see chapter 11 and Appendix).

10.7.2.1.6 Fluorinert (or rqivalent) liquids. Fluorinert (or equal) liquids are a series of dielectric coolants developed for electronic cooling applications. They are clear, colorless fluids, relatively dense and of low viscosity, and are compatible with almost all metals and plastics commonly used in electronic designs. Boiling points range from 31°C (88°F) to 174°C (345°F). Dielectric strength is in excess of 350 volts per mil. and also water and oil solubility are low. The Fluorinert (or equal) liquids are completely miscible with each other, allowing for "tailor-made" liquids (see also section 10.9.2.2). The Fluorinert (or equal) liquids are non-flammable and non-toxic under normal industrial conditions. Section 10.9.2.2 considers the Fluorinert (or equal) as applied to transformer cooling. Fluid properties are included in the Appendix.

10.7.2.1.7 Other coolants. The characteristics of other coolants are given in the Appendix. Only those fluids which are recommended for electronic heat removal applications are noted therein.

For a given cooling load, the temperature rise will vary inversely with the product of the weight rate of flow and the specific heat of the fluid. It is advantageous to have a high specific heat and density so as to secure a high weight rate of flow with small tubes and low flow velocity.

10.7.2.1.8 Comparison of coolants. An interesting comparison of several fluids in free convection, based on experimental work, is given in Table XXVIII.

Table XXVIII shows the great advantage obtained in using a liquid rather than air as a medium of free convection. In the experiments the enclosure walls had to be maintained at -18°C using air as the fluid, whereas, for transformer oil, the wall temperature could be raised to 111°C for the same fluid temperature and heat dissipation. Also, the tube surface temperature for air was 235°C and it was only 143.5°C for the transformer oil.

Liquids of high volatility are not recommended for use as direct liquid coolants. Vapor jackets, with high resistance to heat flow, may form around heat sources. Vaporization cooled equipments require special designs to avoid vapor lock.

TABLE XXVIII

Comparative Heat Transfer Data for Air, Silicone Fluid (or equivalent), Transformer Oil, and Freon (or equivalent); Free Convection

Fluid temperature	130°C
Unit heat dissipation	3 watts/sq. in.
Total heat dissipation from enclosure	115 watts

	Component Part Surface			Enclosure Wall	
Fluid	Temp. Rise Over Fluid °C	Temp. °C	Surface Heat Transfer Coefficient Watts/sq.in. (°C)	Temp. Drop (Below Fluid °C)	Temp. °C
Air(sea level	105	235	0.029	148	-18
Silicone Fluid (or Equivalent)	19.1	149.1	0.157	31	99
Transformer Oil	13.5	143.5	0.222	19	111
Freon -113 (or Equivalent)	5.6	135.6	0.535	7	123

10.7.2.2 Indirect liquid coolants. The two major factors that determine whether or not a coolant is suitable for indirect cooling are its corrosive tendencies and thermal properties. Since the coolant does not come in direct contact with electronic parts, the question of its liquid dielectric properties does not enter into the problem.

Water is generally recognized as the most effective liquid coolant. Its chief faults are its freezing and boiling temperatures. Antifreeze and corrosion inhibiting additives may be added without a serious reduction in heat transfer characteristics.

10.7.2.3 Antifreeze solutions. Antifreeze solutions are used in recirculating cooling systems which are exposed to freezing temperatures. There are two general types of antifreeze materials, those with a glycol base and those with an alcohol base (methanol and ethanol). The glycol base antifreezes have a higher boiling point and are referred to as permanent type antifreezes.

The presence of alcohol in the system without an inhibitor will loosen the protective film of the metallic surfaces. If an alcohol base antifreeze is continued in use after its corrosion inhibitor has lost its effectiveness, an inhibitor should be added. Borax can be used as an inhibitor for an ethanol base antifreeze; a 0.5% solution of borax

(0.7 ounces per gallon of solution) is adequate. For a methanol base antifreeze, either borax or sodium chromate may be used as an inhibitor.

10.8 Design of liquid cooling systems.

10.8.1 Direct immersion, sealed assemblies. The design problems encountered in a liquid-immersed assembly include hermetic sealing, provision for expansion of the liquid coolant, vapor pressure, strength of the container, ease of repair, removal of the coolant from the parts, orientation of parts and the effects of the coolant on the function of high frequency circuits. The orientation and mounting of parts must be given special consideration since they should be located so as to achieve maximum convection.

As mentioned in section 10.4.2 it is desirable to expose the largest possible surface of the heat producing parts to the coolant and to direct the free convection currents of the fluid around these parts. Parts should be mounted, as in air, to promote convective cooling. Metallic conduction paths of low thermal resistance from the heat-producing parts to the surface of the case are not as important in direct liquid cooled assemblies as in most other types of assemblies, since adequate cooling is usually obtained by convection. The parts may be supported by solid insulators or insulating materials so long as the fluid is permitted to circulate freely around the parts.

The construction of this type of equipment must be given special consideration. Viscous coolants can provide a damping action which assists in resisting vibration and shock, dependent upon the viscosity of the fluid. Cylindrical parts should be mounted vertically and the entire surface should be exposed to the coolant fluid. This provides maximum cooling at the hot spots and tends to minimize electrolysis. Holes can be provided in the subchassis to direct the flow of the free convection currents around the heat-producing parts.

10.8.1.1 Experimental results. A liquid cooled assembly was fabricated from a cast aluminum assembly originally designed for natural cooling. This assembly consisted of two cast aluminum housings, i.e., upper and lower, with a gasket-type seal at their joint. Thermatrons (Reference 63) were installed as heat sources and the assembly was liquid filled. The test conditions in order of increasing plate temperature at constant power were:

1. Assembly filled with Dow Corning #550 fluid, (or equivalent), water in heat exchanger.

2. Assembly filled with mineral oil (Nujol) (or equivalent), water in heat exchanger.

3. Assembly filled with mineral oil (Nujol) (or equivalent), no water in heat exchanger.

4. Assembly filled with air, water in heat exchanger.

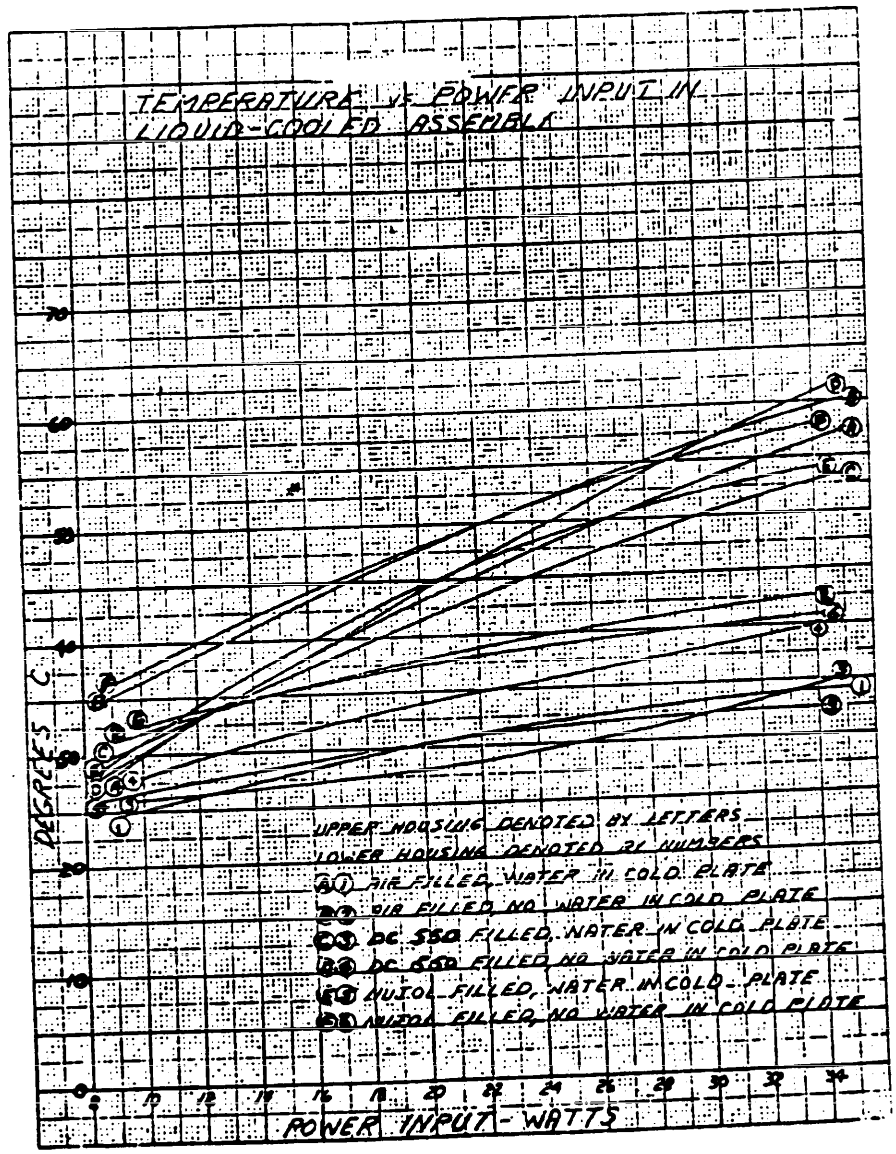


FIGURE 100. Temperature vs. Power Input In Liquid-Cooled Assembly

5. Assembly filled with air, no water in heat exchanger.

6. Thermatron in free air, not mounted on a heat exchanger.

Figure 100 is a plot of the upper and lower housing temperatures in test conditions 1, 2, 3, 4, and 5. These data show clearly that effective heat removal is accomplished by cooling the cold plate with water. Whether the case was filled with air or oil had little effect on case temperatures. The oil made a very slight improvement in case temperature rise, but conduction through the oil did cause severe heating of the terminal board, as evidenced by warpage. Considerable trouble was also experienced with oil leakage, particularly around the pins swaged into the connector which carried the leads into the lower housing.

It was concluded that conductive cooling to a water-cooled heat sink, with an appropriate direct liquid coolant inside the housing, is the most desirable arrangement.

10.8.2 Indirect liquid cooling systems.

10.8.2.1 Cabinets and enclosures; air-to-liquid systems. One method of indirect liquid cooling commonly employed, especially on shipboard rack or cabinet mounted equipment, utilizes a self contained, recirculating forced air system within the cabinet. The air is cooled in a forced air-to-liquid heat exchanger in the cabinet and re-introduced to the electronics. The liquid portion of the heat exchanger is generally the ships fresh water cooling supply. While these systems have much merit, the heated air will contact the cabinet enclosure outer surfaces, and by virtue of these exterior surfaces being heated, some heat transfer to the space or room containing the equipment will occur by free convection and, to a lesser extent, radiation. In practice, it has been found that the most carefully designed system of this type will transfer about 75% of the dissipated heat into the water, with the remainder being dissipated into the space. More commonly, 60% or less of the heat is absorbed by the water cooled heat exchanger. (High power water cooled tubes are an exception; careful design usually results in almost all the heat being transferred to the water.)

In a space, equipments dissipating several kilowatts can represent an intolerable rate of heat transfer into the space, resulting in personnel discomfort, air conditioner overload, and possible failure of other lower powered equipment dependent on space or cabin temperatures. Even liquid cooled chassis (cold-plates) will normally allow escape of some heat into the space, although the percentage of heat lost with a properly designed cold plate is generally not as high as with the forced air-liquid heat exchanger system.

The equipment designer should strive for maximum heat transfer within a controlled path, and minimize heat loss by parallel paths into the local environment. Methods of reducing heat loss into ship's spaces include insulating the enclosure; bleeding cool air through small orifices across the inside surfaces of front panels and enclosure walls, using water cooling channels in the enclosure walls to intercept the stray heat, and directing the air flow properly over the electronic parts so that the heat transferred to the air is maximized.

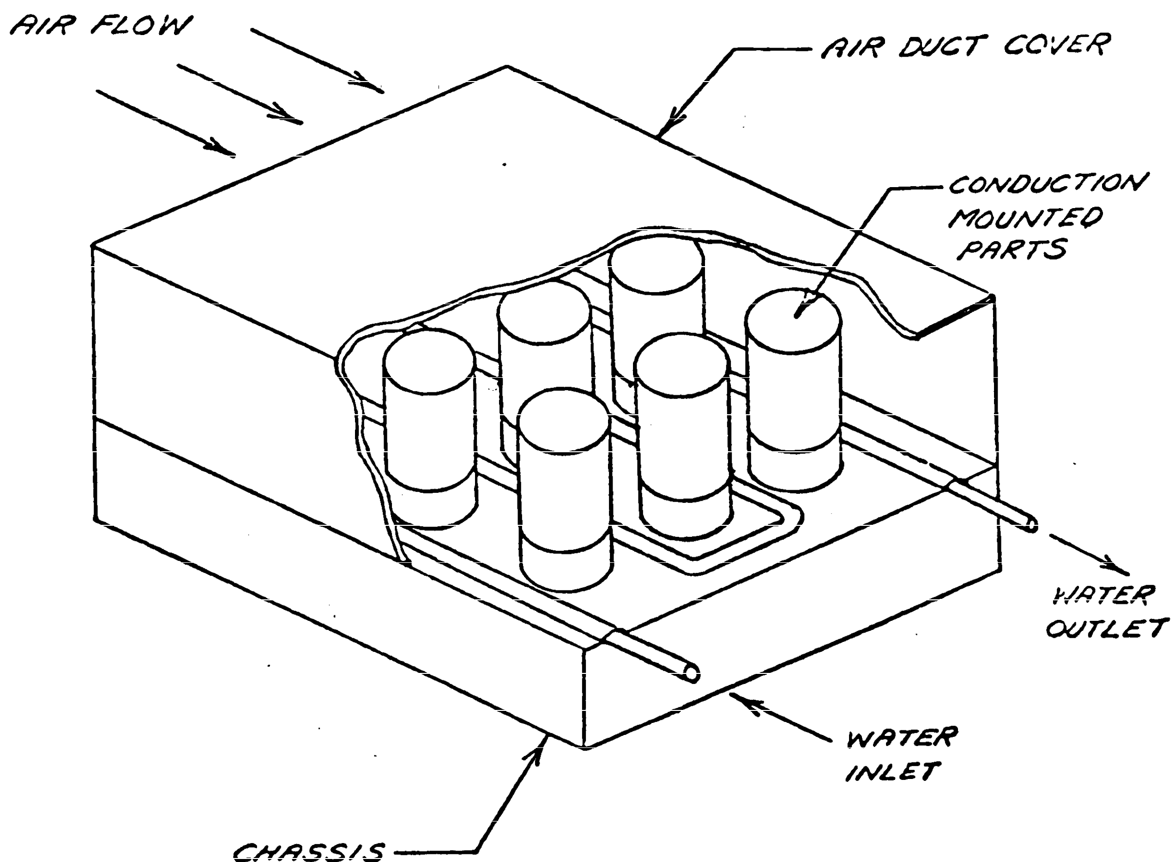


FIGURE 101. Cross Section of a Combined Cooling Chassis

10.8.2.2 Composite cooling systems. It is sometimes advantageous to combine fcrced air cooling and indirect liquid cooling, part of the heat produced by the heat sources flowing by natural means to the liquid-cooled panels and part being carried away (usually to a heat exchanger) by the cooling air flowing across or parallel to the heat sources. Figure 101 shows a cross section of a typical combined cooling chassis.

10.8.2.3 Ultimate sinks for liquid cooling. Large bodies of water provide an attractive ultimate heat sink because of their excellent heat transport properties and because of the high heat transfer coefficients attainable in the systems final heat exchanger. Typical liquid ultimate heat sinks include sea water for shipboard systems, natural ground water for ground based systems, and liquid fuel in airborne and spacecraft systems. A specific system may be cooled by a closed liquid loop; assuming that the liquid loop capability is not exceeded, it may be considered as a local or intermediate sink for that system, even though it in turn, delivers its heat to the ultimate sink. For example, shipboard systems are frequently cooled by a recirculating fresh water coolant supplied by the ship. This fresh water coolant is subsequently cooled by a fresh water-to-sea water heat exchanger. For purposes of the electronic system thermal design, the fresh water loop may be considered the intermediate sink. Similarly, many van or trailer installations employ indirect liquid cooling for the electronics, with the coolant subsequently delivering the heat to the ambient air by means of a large liquid-to-forced air heat exchanger as part of the van. Here also, the liquid coolant represents an intermediate sink for the electronics.

Mobile ground based systems occasionally employ natural water (lake, rivers, streams, etc.) as an ultimate sink for cooling high powered equipment. This water is never used directly in the system, but instead, is used to cool an intermediate loop. Consequently, characteristics of the natural water coolant are not usually of direct importance to electronic system designers.

High speed aircraft and spacecraft are faced with high thermal resistances between the vehicle and the ultimate sink (the atmosphere or space) with a relatively high temperature sink. Consequently, designers of these vehicles are constantly looking for other modes of heat disposal. Expendable coolants and vehicle fuel are two possible ultimate sinks. Expendable coolants obviously impose a weight penalty, but are sometimes used for short duration, high thermal intensity missions, such as the final strike of an attack aircraft. The combination of the weight-specific heat product and the heat of vaporization of the expendable coolant can provide for the absorption of extremely large quantities of heat. (See chapter 11)

10.8.3 Agitation of coolants. Agitation of coolants enhances the natural convection process by forced local circulation of the fluid. This tends to reduce the boundary layer thickness and is analogous to pumping the fluid. The heat transfer process in direct liquid cooling with agitation is difficult to analyze since it is not a true free convection process and the forced circulation is not usually vigorous enough to establish clear flow patterns. The recommended analytical approach is to apply the general free convection relationship modified for the higher heat transfer rate produced by agitation. Usually, experiments simulating the thermal configuration will determine empirical data sufficient for design purposes.

The agitation can be deliberately introduced into the fluid or be produced by an existing environmental effect. For example, with oil immersed high power transformers, the motion of the cores due to magnetostrictive effects and the motion of the conductors due to magnetic forces at double the operating frequency significantly enhances the natural convection. Also, the vibration of liquid cooled devices mounted adjacent to rotating machinery enhances the natural convection. On the other hand, agitation can be introduced into the fluid by a low frequency vibrator or, as is discussed in the next subsection, by ultrasonic agitation. It is important to note that severe agitation or vibration can damage electronic parts and, consequently, care must be exercised in utilizing such techniques.

10.8.4 Ultrasonic agitation (insonification) of coolants. This technique involves the utilization of an ultrasonic transducer, operating usually in the 25 to 40 Khz range, to insonify the liquid coolant and disturb the boundary layers at the interfaces between the heat transfer surfaces and the liquid. Thus, the heat transfer coefficient can be enhanced both at the liquid to heat source interface and at the liquid to heat exchanger surfaces.

This technique is relatively unexplored and has found little use to date. At least one practical application is known. In this instance, ultrasonic agitation was used in a vaporization cooling application to control the size of the vapor bubbles and reliably extend the maximum heat flux into the range of normally unstable boiling. This permitted the operation of a ferrite core computer memory at up to 50 watts/cu. in., previously impossible to achieve.

References 64 through 70 inclusive discuss details of ultrasonic agitation. In brief, significant improvements in heat transfer coefficient can be achieved dependent upon the fluid, the amplitude of insonification, the frequency, and the overall configuration. Up to 370% increases have been reported. Currently, there is no unified theory to predict the expected improvement in the heat transfer coefficient as a function of the measurable variables. Operation at or beyond the threshold of cavitation generally maximizes the heat transfer coefficient. Liquids with physical properties which facilitate cavitation (see section 10.8.4.4) show the greatest increase in heat transfer coefficient. However, this does not mean that ultrasonic agitation cannot be used with relatively low amplitude insonification or with liquids that "resist" cavitation.

In some experiments a critical power level has been observed beyond which no appreciable improvement was found. In other tests a critical power level was found below which the cooling improvement diminished rapidly.

10.8.4.1 Configuration. The arrangement typically can be similar to that of an ultrasonic cleaner or if the cooling imrpovement is to be maximized, the transducer can be mounted adjacent to the surface to be cooled and oriented so that the energy is "beamed" on that surface. Both piezoelectric and magnetostrictive transducers have been used. The attitude of the surface being cooled is apparently not significant.

Also, since the acoustic field in the liquid propagates over considerable distances and penetrates into holes and crevices, heat sources with unusual shapes can be effectively cooled.

10.8.4.2 Electrical and acoustical power and frequency. Electrical power inputs to transducers range from 0.01 to 0.5 of the power dissipated by the heat sources being cooled. Acoustic power density at the surfaces of transducers is usually less than 1 watt/sq. in.

Higher frequencies have been reported that have resulted in increased heat transfer coefficients. However, the effects of the increasing directivity of the transducers with increasing frequency were not accounted for. This may have caused increased acoustic power densities at the heat transfer surfaces at the higher frequencies.

10.8.4.3 Fluids. Ultrasonic agitation is most effective in water. With fluids, such as toluene, kerosene, petroleum oils, and freons (or equivalent), the improvement in convective heat transfer coefficient is approximately one half or less than that obtained with water. With evaporative heat transfer significant gains have been observed with all of the above fluids. Ultrasonic agitation also delays the onset of vigorous boiling.

In natural convection the Nusselt number is increased by the ultrasonic agitation to a maximum of about four times the magnitude obtained with natural convection. At frequencies below 125 Khz the increased heat transfer coefficient results from cavitation. The Nusselt number for a given magnitude of the Grashof-Prandtl modulus correlates with the parameter.

$$\sqrt{\frac{I}{I_c}}\left(\frac{C}{C_s}\right)^{1.5} \qquad (10\text{-}37)$$

Where:

I = the sound intensity at the transducer

I_c = the cavitation threshold intensity of the gas-saturated liquid

C = the gas content of the liquid

C_s = the gas content of the liquid under saturated conditions

This parameter can be considered as the ratio of the sound pressure to the sound pressure required to cause cavitation. The factor $(C/C_s)^{1.5}$ provides the variation of cavitation threshold pressure with gas content.

10.8.4.4 Cavitation. Cavitation in liquids consists of formation of breaks or cavities filled with vapor of the liquid or with gas which is normally dissolved in the liquid. When these cavities collapse, they form strong local compression shocks, reaching thousands of atmospheres pressure.

Actually, vaporous cavitation is merely a form of boiling of the liquid; the only difference is that during ordinary boiling, the bubbles are filled with vapor, while in cavitation they may be filled either with vapor or with gas.

Experimental measurements have shown that for samples of tap water, and even for triple distilled and filtered water, the tensile pressures required to pull the water apart (cavitation threshold), are roughly three orders of magnitude less than the calculated value for idealistically pure water. This discrepancy has been explained by the nucleation theory of cavitation. Several hypotheses have been suggested as mechanisms by which "weak spots" are generated in a liquid.

1. Thermal fluctuations in the liquid give rise to tiny vapor bubbles which then function as cavitation nuclei. If the vapor pressure of the liquid is greater than the external pressure, the bubbles will grow; if, however, the vapor pressure is less than the external pressure, they will diminish in size.

2. Another possible supposition is that the liquid does not actually break down in its actual volume or body, but where it forms an interface against a solid surface, as in the case of a suspended particle. In as much as the liquid can contain particles with varying degrees of wettability, the strength of the liquid can fluctuate within great limits. Moreover, solid particles can have microdefects into which water cannot penetrate.

3. Under certain conditions stable gas bubbles can remain for long periods of time in water. These bubbles can also act as cavitation nuclei. For example, freshly drawn tap water has many of these bubbles.

When intense periodic pressure waves are transmitted through a fluid, the water can be ruptured at one of these weak spots if the negative pressure exceeds that of the cavitation threshold. Depending upon the type of nuclei at which the rupture takes place, two types of cavitation result. In the fluid itself, as opposed to the fluid and boundary, the weakest spot is usually a gas if stabilized gas bubbles are present. Therefore, with increasing negative pressures, gas bubbles act as nucleation centers and the resulting phenomenon is referred to as gaseous cavitation. If, on the other hand, the weak spots are tiny vapor bubbles, vaporous cavitation occurs.

The cavities formed in both cases are not stable and the walls of the void collapse during succeeding overpressure cycles. With a spherical cavity containing a vacuum in a liquid, the instantaneous collapsing radial implosion velocity is proportional to $1/R^2$, where R is the instantaneous radius of the collapsing cavity. Thus, when $R \to o$ the instantaneous radial velocity approaches ∞. It should be noted that the terminal velocity at the instant the inrushing walls are maximum can never reach ∞, because in gaseous cavitation the entrapped gas cushions the implosion, while in vaporous cavitation, both minute traces of gas and uncondensed water vapor cushion the impact and reduce the magnitude of the propgated shock wave. Because of the greater cushioning effect of entrapped gas in gaseous cavitation, the shock waves tend to be less intense than those of vaporous cavitation.

10.9 Design methods for predicting thermal resistances using liquid cooling.

10.9.1 General. It is difficult to formulate generalized thermal resistances for liquid cooled systems because of the wide variation in systems to which these cooling methods are applied. In general, however, any application must consider at least three generic types of thermal resistance.

The first of these might be termed the "local" thermal resistance. This is the thermal resistance associated with the transfer of heat from the particular component, module, or subassembly under consideration into the "mainstream" of the thermal network. A given design problem will contain as many local thermal resistance computations as there are components to be thermally evaluated.

The second major class of thermal resistance is that associated with the disposal of the heat generated by the system into an ultimate sink. Normally, this will consist of some type of heat exchanger for liquid cooled systems, although in rare instances, expendable coolants might be employed.

The third major thermal resistance is that within the liquid cooling system. Liquid cooling systems may involve either or both of the local and system thermal resistances, and may be employed in conjunction with other cooling techniques, e.g. - natural methods of conduction, convection, and radiation; forced air cooled systems; or special cooling techniques.

The following sections will provide some design guidelines for predicting the thermal resistance of the liquid cooled portions of systems, and will include design examples to illustrate the methods employed.

10.9.2 Direct liquid cooling.

10.9.2.1 The general formulas of section 8.1.2 are applicable as follows:

$$q/A_s = C\ k/L\ (Gr\ Pr)^m\ \Delta T \qquad (10\text{-}38)$$

or

$$q/A_s = C\ k/L\ (aL^3)^m\ \Delta T^{\ m+1} \qquad (10\text{-}39)$$

Where $a = (Gr\ Pr)/(L^3 \Delta T)$

The constants C and m must be evaluated from experimental data. For (Gr Pr) between 10^3 and 10^9, m assumes a value of 1/4. For (Gr Pr) greater than 10^9, m assumes a value of 1/3. Liquid cooled systems may fall into either category, dependent on the properties of the fluid used, the characteristic length, and the temperature differential. For small parts at normal temperature differentials, m will generally be 1/4. (Reference 8 states that (Gr Pr) is likely to exceed 10^9 if the height of the surface in a liquid is more than 0.15 ft.) Values of Gr, Pr, and occasionally a, may be found in tabular form in various handbooks and manufacturers publications for commonly used fluids. These are usually given as functions of temperature. Evaluation of Gr and Pr should be made at the average of the surface and the bulk fluid temperature (unless the tabulated data indicate some other reference temperature).

Table XXII lists values of L for common shapes, and may be used for irregularly shaped electronic parts if the most similar condition is selected. Table XXIII lists values to be used for the constant C.

10.9.2.1.1 Design example - Transistor on small plate. In a laboratory experiment, a transistor with a maximum collector dissipation of 20 watts at 25°C surface temperature was mounted on a small copper plate and immersed successively in silicone "200" fluid (or equivalent), toluol, carbon tetrachloride, methyl alcohol, distilled water, and Freon 113 (or equivalent). The fluid which proved best for heat removal by free convection was distilled water (not electrically practical). The others in order of thermal merit were: methyl alcohol, Freon 113 (or equivalent), toluol, carbon tetrachloride, and silicone "200" (or equivalent). A correlation of experimental data with the free convection equation, for small parts such as transistors immersed in liquid coolants, results in:

$$Nu = 1.47\ (Gr\ Pr)^{0.25} \qquad (10\text{-}40)$$

It appears that the heat was dissipated directly from the transistor to the fluid rather than from the transistor to the copper plate and thence, to the liquid because the diameter of the transistor could be used instead of the equivalent dimension of the vertical plate in correlating the free convection equation. Actually, the area of the plate was small compared to that of the transistor. Investigations with larger plates indicated that the area of the plate must be considered when it exceeds 25% of the total transistor area.

10.9.2.1.2 Design example - P. C. card component. Determine the approximate surface temperature of an electronic part mounted on a circuit card within a fluid filled sealed enclosure. The 1.0 x 1.0 x 0.1 inch element dissipates 20 watts. Other circuitry within the housing dissipates an additional 80 watts. The housing is a three inch cube, held at a uniform temperature of 50°C. The fluid is FC-75 (or equivalent). Ignore conduction heat paths into and through the circuit card, and assume there is no impediment to natural convection circulation currents.

The thermal analog is given in Figure 102.

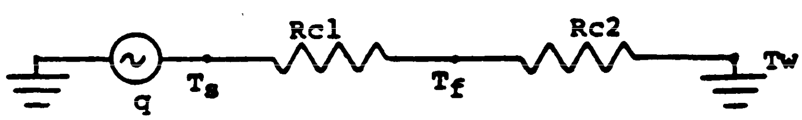


FIGURE 102. Thermal Analog

(1) Determine R_{c2}, the convection thermal resistance between the fluid and the enclosure.

$$\frac{1}{R_{c2}} = h_c A = CA\frac{k}{L}\ (Gr\ Pr)^m$$

Gr and Pr should be evaluated at the mean temperature of the fluid and the wall. Since the fluid temperature, T_f, is unknown, an estimate and possible iteration must be made. Assume a 10°C temperature differential, so that the average film temperature is $50 + \frac{10}{2} = 55°C$. Then,

$a = 24.70 \times 10^9$ un./cu. ft. -°C for FC-75 at 55°C (See Appendix)

$(Gr\ Pr) = a\ L^3 \Delta T = (24.70 \times 10^9)(.25)^3(10) = 3.86 \times 10^9$

Use $m = 1/3$, since $(Gr\ Pr) > 10^9$

From Table XXIII, C = 0.55 for vertical surfaces (use a single constant value of C for all six contacting surfaces, assuming that the top surface and bottom surface are the equivalent of a vertical surface of the same total area). The thermal conductivity of FC-75 (or equivalent) is 0.0195 watts/ft.-°C.

Then,

$$h_c A = \frac{(0.55)(6)(.25)^2(.0195)}{.25} (3.86 \times 10^9)^{1/3}$$

$$h_c A = 25.2 \text{ watts/°C}$$

and,

$$\Delta T = \frac{Q}{h_c A} = \frac{100}{25.2} \approx 4°C$$

Since ΔT is only 4°C instead of the assumed 10°C, recalculate:

$$T_m = 50 + \frac{4}{2} = 52.0°C$$

$$a = 26.10 \times 10^9 \text{ units/cu. ft.-°C}$$

$$(Gr\ Pr) = (26.1 \times 10^9)(.25)^3(4) = 1.63 \times 10^9$$

$$h_c A = \frac{(0.55)(6)(.25)^2(.0195)}{.25} (1.63 \times 10^9)^{1/3} = 18.9 \text{ watts/°C}$$

and,

$$T = \frac{Q}{h_c A} = \frac{100}{18.9} = 5.28°C$$

This is close enough to the assumed 4°C ΔT that further refinement of the values of a and k are unnecessary.

Thus, the fluid temperature, T_f, = 50 + 5.28 ≈ 55°C.

(2) Determine R_{c1}, the convection thermal resistance between the part and the fluid.

$$\frac{1}{R_{c1}} = h_c A = CA \frac{k}{L} (Gr\ Pr)^m$$

A similar iteration procedure is required, because the part surface temperature T_c is unknown. Assume a 40°C temperature differential.

Then,

$$T_m = 55 + \frac{40}{2} = 75°C$$

$a = 32.9 \times 10^9$ units/cu. ft.-°C for FC-75 (or equal) at 75°C

$$(Gr\ Pr) = aL^3\ \Delta T = (32.9 \times 10^9)(\frac{1}{1728})(40) = .762 \times 10^9$$

$m = 1/4$ since $(Gr\ Pr) < 10^9$

$h_c A = CA \frac{k}{L} (Gr\ Pr)^m = .548$ (use C = 1.45 for small parts)

$$\Delta T = Q/h_c A = \frac{20}{.548} = 36.5°C$$

This is slightly less than the 40°C T assumed, but close enough that fluid properties may be considered constant.

$$R_{c1} = \frac{\Delta T}{Q} = 1.83°C/watt$$

The part surface temperature is 55 + 36.5 = 91.5°C (ANSWER)

These computed temperatures should be used cautiously, because of the many unknown variables associated with free convection in direct liquid cooling. <u>A generous safety factor should be allowed in the design process.</u> Thus, while the computations show a total temperature rise of 91.5 - 50 = 41.5°C from the housing to the element, at least 25% should be added for unknown factors; i.e., a minimum rise of 50°C should be anticipated. Verification tests should be conducted as early as possible on breadboards or thermal mockups.

10.9.2.2 <u>Liquid-cooled transformers</u>. Considerable work has been performed in the field of miniaturizing transformers, using direct liquid cooling. Several design techniques using Fluorochemicals in immersion-type cooling have been developed. The size and weight of many electronic-type transformers may be considerably reduced by utilizing Fluorochemical liquids and/or gas and conduction design techniques, at operating temperatures up to 200°C. Fluorochemical liquids have electrical properties which are equivalent to the lower-operating-temperature transformer oils. These properties, together with excellent heat transfer characteristics of the conduction plates and Fluorochemical liquids, lead directly to smaller and lighter transformer designs. By employing the solubility of certain gases in selected Fluorochemical liquids, internal pressures may be minimized and the need of bellows, pellets, etc., may be eliminated.

10.9.2.2.1 Volatile liquid immersion. Reactor immersion in a volatile fluid such as a Fluorochemical provides a low resistance to heat transfer from the heat source to the case surface. The transformer should be mounted in a sealed container together with a suitable Fluorochemical and expansion space. The low thermal resistance is obtained from the high autoconvection characteristic of the Fluorochemicals and from the latent heat of the volatile liquid vapors as they come into contact with the container surface in the expansion space. The transfer of heat through the expansion space, which in ordinary oil-immersed transformers is not too effective, is considerably increased. It should be noted that the more volatile liquids provide a lower thermal resistance. However, the more volatile fluids produce a greater internal pressure. This cooling technique, vaporization cooling, is discussed in more detail in chapter 11.

10.9.2.2.2 Coolant mixtures. The Fluorochemical liquids presently available and suitable for transformer designs vary in volatility. In most instances, the operating temperature is such that the liquid which is otherwise most suitable for cooling would not be used because its vapor pressure is in excess of desired container design limitations. A liquid of lower volatility, however, gives a higher thermal resistance. It is possible by a proper mixture of two Fluorochemical liquids to calculate an optimum dielectric coolant consistent with the desired pressure and thermal limitations.

The following is the formula used for this calculation:

$$\frac{W_B}{W_A} = \frac{(p_o - p_A)\, M_B}{(p_B - p_o)\, M_A} \qquad \text{(10-41) (D.E.)}$$

Where:

W_A = weight of less volatile liquid, lb.
W_B = weight of more volatile liquid, lb.
M_A = molecular weight of less volatile liquid
M_B = molecular weight of more volatile liquid
p_A = vapor pressure of less volatile liquid at the operating temperature, lb./ft.^{2}
p_B = vapor pressure of more volatile liquid at the operating temperature, lb./ft.^{2}
p_o = maximum vapor pressure of the optimum liquid at the operating temperature, lb./ft.^{2}

It is possible to mix two or more gases to provide a gas solubility coefficient which will produce the desired pressures consistent with adequate dielectric strength over the specified operating temperature. Liquid combinations in conjunction with gas combinations to give optimum heat transfer and adequate dielectric protection within desired pressure limits may be selected.

10.9.2.2.3 Partial fill and use of "wicking action" of liquid Fluorochemicals. The exceptional wetting action of liquid fluorochemicals may be used to conduct these liquids through an inorganic wick to the interstices of a reactor winding. Upon reaching the "hot spots," the liquid vaporizes and transfers heat from the winding by the evaporative cooling. The vapors condense on the inner surface of the container and ultimately return to the mass of liquid to complete the cycle. This method may minimize weight and cost through the selection of the minimum amount of liquid. Variations of this cooling method are presented in chapter 12, section 2, "Heat Pipes."

10.9.2.2.4 Fluorochemical liquid and gas combinations at reduced pressure as dielectric coolants. The use of a Fluorochemical gas at less than atmospheric pressure in the expansion space over the liquid offers advantages in relation to heat transfer, and is practically as good as dielectric as air at one atmosphere pressure. With a transformer container having a large surface area at the top to facilitate condensation of vapors, any air or gas present in the case at the time of sealing displaces a portion of the vapor during operation; thus, reducing the effective cooling. By reducing the quantity of gas at the time the unit is sealed, the entire upper surface of the container provides condensing area for the vapor and the heat dissipation is increased. The use of this technique is also a practical means for the control of internal pressure within mechanical design limitations. The use of Fluorochemical gases in such a combination may provide necessary dielectric protection when the liquid vapors are inadequate, i.e., low temperature starting conditions, etc.

10.9.2.2.5 Liquid and/or gas Fluorochemical fill in conjunction with heat conductor plates. Conduction plates of copper have been used successfully in many dry-type transformers of moderate voltages to transfer heat from the windings to an external radiator. The copper conducting plates are an integral part of the coil construction and consist of copper foil of sufficient thickness for proper transfer of the heat to be dissipated. The plates may serve as electrostatic shields in the coils or as thermal conductors around the core. From the thermal conductivity coefficient of copper (9.6 watts-in. per in.^{2}-°C), it is obvious that the copper of relatively small cross section can reduce the gradient between the winding and the external radiator. When such a transformer is mounted on a heat sink, heat transfer is considerably increased.

Although considerable heat can be removed by this method, resulting in a reduction of size, this technique is sometimes limited because of dielectric and corona problems at higher voltages. Higher voltage operation is possible when Fluorochemical gas or partial liquid fill is used in a hermetically sealed transformer.

Fluorochemical liquid and heat conduction techniques may be applied to many transformer designs. One or more of the above mentioned techniques may be used in an optimum transformer design for a specific set of operating conditions.

Figure 103, which summarizes the experimental results obtained with various liquids, shows that considerably more power may be dissipated in a transformer when volatile liquids surround the coil.

10.9.2.2.6 Direct liquid cooling of electron tubes. The heat rejected from glass bulb, metal, and external anode electron tubes can be removed effectively by direct immersion. The radiation which penetrates the bulb of glass tubes is almost entirely absorbed by the coolant within a distance of 0.25 in. from its outer surface.

Figure 104 presents tube plate temperatures for various fluids.

10.9.3 Indirect liquid cooling.

10.9.3.1 General. Through the use of indirect liquid cooling, heat sources can be arranged in the flow system independent of their physical locations, since the small interconnecting lines that are required can be installed and routed almost as readily as electrical wiring. If judiciously accomplished, this minimizes the flow rate and pumping power requirements and allows part placement for greatest compactness. Because of the predominance of conduction heat transfer to the liquid over all other incidental modes, the heat flow paths are well defined, and the interaction of parts is small.

One of the most important gains achieved by indirect liquid cooling equipment is the resulting flexibility in the installation. The small space requirements of liquid conveyance make the use of remote heat sinks economical. As within a single unit, the flow rate in a transfer system combining several units of different temperature limits can be minimized by arranging them in the order of increasing temperature limits.

Individual cooling devices may be used for the major heat-dissipating parts, particularly special parts such as TWT's. Such devices must envelope the part surface as completely as possible so as to provide a heat flow path of least thermal resistance between the heat source and coolant. Heat transfer from the tube alone is not considered adequate for high power tubes. Means must be provided for maintaining allowable pinseal temperatures by conduction through an adequate insulator to a cooled metal portion of the chassis (or by forced air).

The cold plate shown in Figure 105 will readily cool power transistors. Typically, a coolant flow of 0.9 lbs./min. had a 15°C rise.

For maximum compactness and flexibility of application, the equipment should not incorporate within itself the means for liquid circulation or ultimate heat exchanger. These can be provided more efficiently and compactly in a separate unit or as part of an overall cooling system. Parts

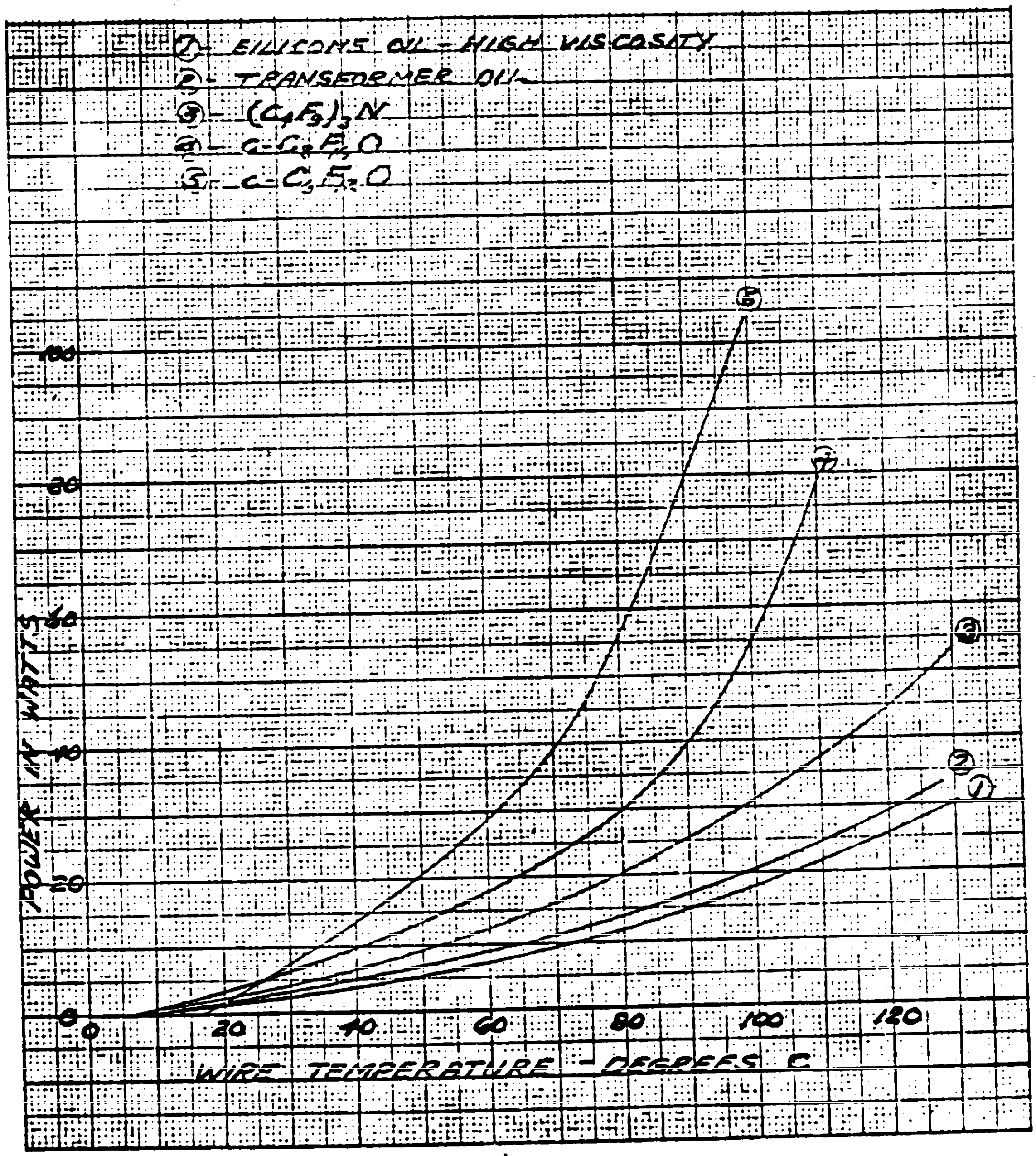


FIGURE 103. Effect of Coolant on Transformer Cooling Transformer Power vs. Wire Temperature With Various Coolants

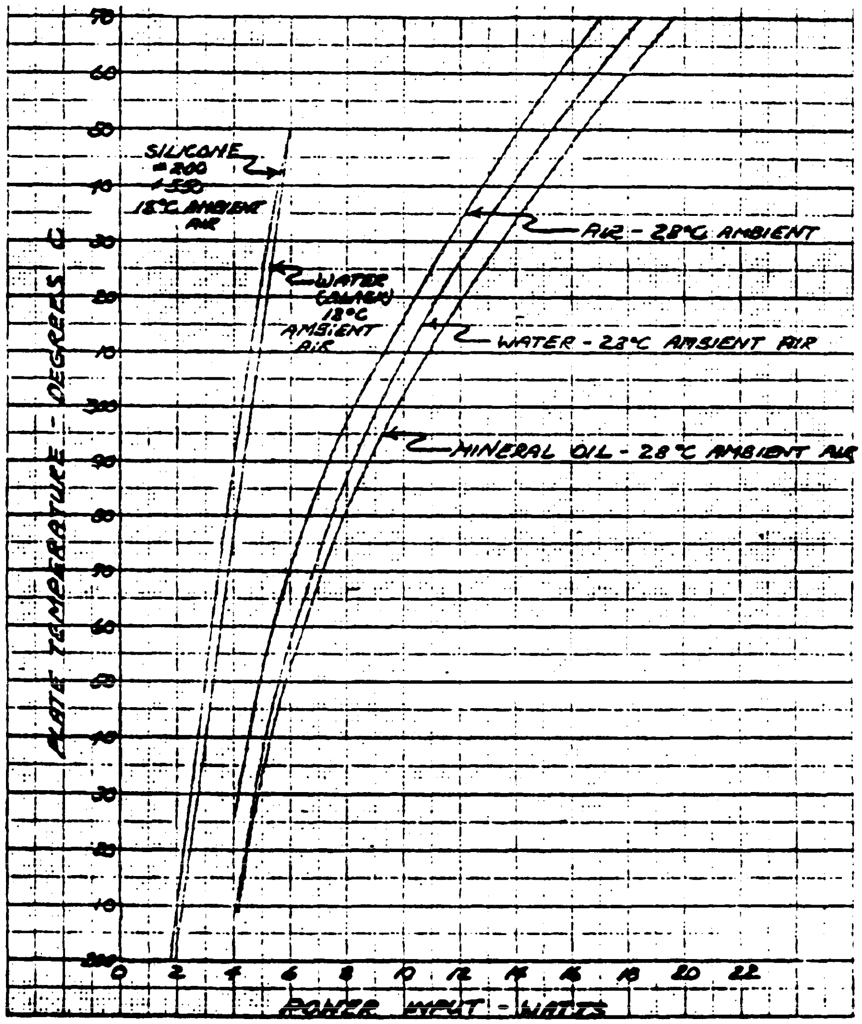


FIGURE 104. Effect of Coolant on Plate Temperature For a 6AQ5 Tube

Power Input vs. Plate Temperature Tube Vertical Base Up

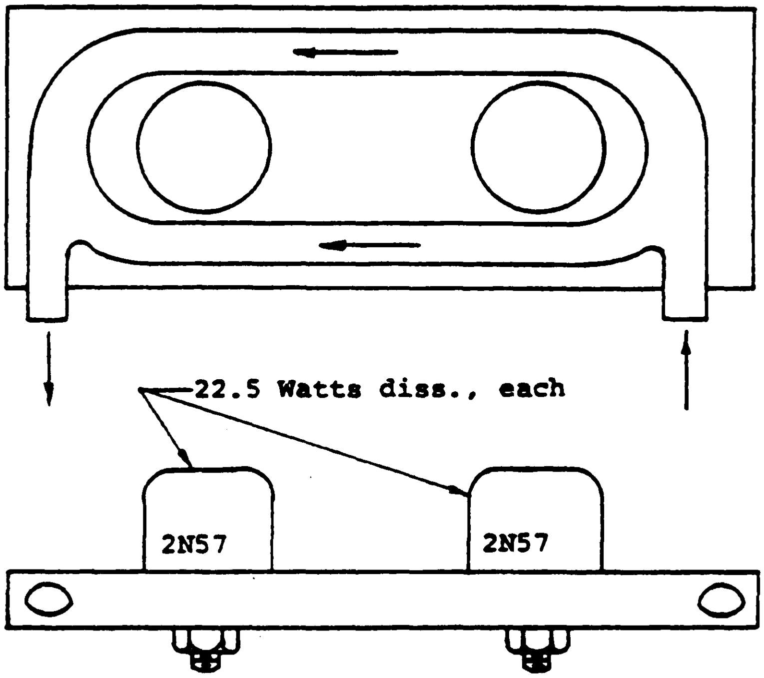


FIGURE 105. Transistors Mounted on A Water-Cooled Chassis

should be arranged according to the criteria of least total space requirements and best electrical interconnection since cooling means can readily be provided at all locations within the unit's case.

Even though parts may not be designed specifically for liquid cooling, suitable means can be provided for conducting their heat to the liquid-cooled surfaces on which they are mounted. In this manner, the liquid may become the primary means of heat extraction from an electronic equipment.

10.9.3.2 Cold plate chassis-experimental example. A standard type 17" x 13" x 3" aluminum chassis was modified by replacing the deck with an expanded metal cold plate of similar dimensions. Heat producing parts such as tubes, transformers, and resistors were mounted on the plate and dissipated a total of 537 watts. Each part was mounted so that the thermal resistance to the chassis was minimized. The resistors were held in metal clamps as shown in Figure 106. Note that a large surface of metal-to-metal contact was formed between the clamp and chassis to facilitate heat transfer by conduction. These clamps were shaped to contact the resistor bodies closely.

The two transformers had to be reworked to improve heat conduction from the bottom of the core stacks and the mounting flanges. The mounting flanges were "shimmed" up so that the transformer could straddle a flow tube as shown in Figure 107, thus, providing the maximum amount of surface contact with the chassis.

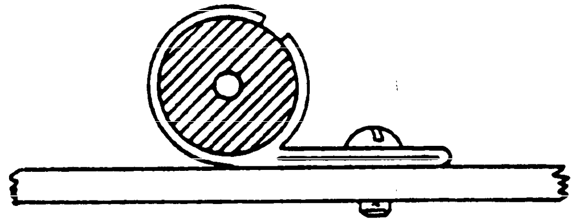


FIGURE 106. Resistor Clamp

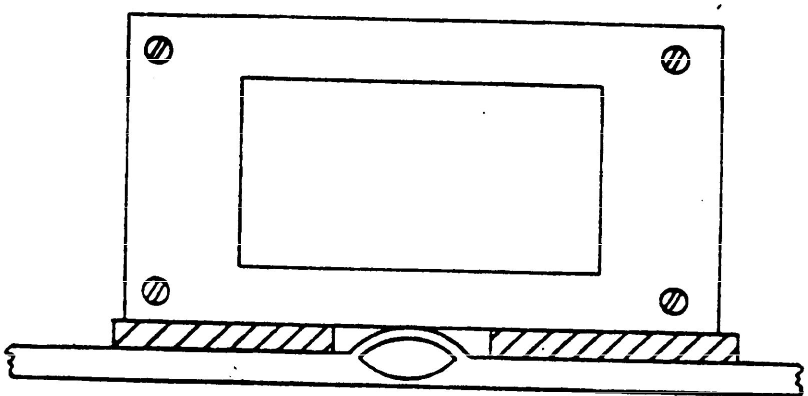


FIGURE 107. Transformer Location on Cold Plate Chassis

The main consideration in the thermal design of water-cooled chassis is to provide metallic conduction paths from the heat producing parts to the chassis. Chapter 8 presents a detailed discussion of conduction heat transfer.

Data for this chassis were obtained while operating under simulated shipboard conditions.

10.9.3.3 Figure 108 is a design curve applicable to all similarly constructed cold plate chassis, computed for 35°C inlet water and 2.8°C water temperature rise. This curve is a plot of approximate water flow rate required vs. total power input to the equipment mounted on the chassis, and is useful as a guide in estimating cooling water requirements.

For this particular chassis the power input is 537 watts, and from Figure 109 the required water flow rate is 0.56 gallon per minute (4.65 lb./min.). Figure 110 shows the observed temperatures of certain parts mounted on this chassis vs. water flow rate. It is seen that flow rates in excess of 0.1 gallon per minute result in small decreases in component temperatures. However, it is seen from Figure 108 that a flow rate of 0.46 gallon per minute is required for a cooling water temperature rise of 2.8°C, as required by NAVSHIPS. Figure 108 therefore, calls for 20% more water than

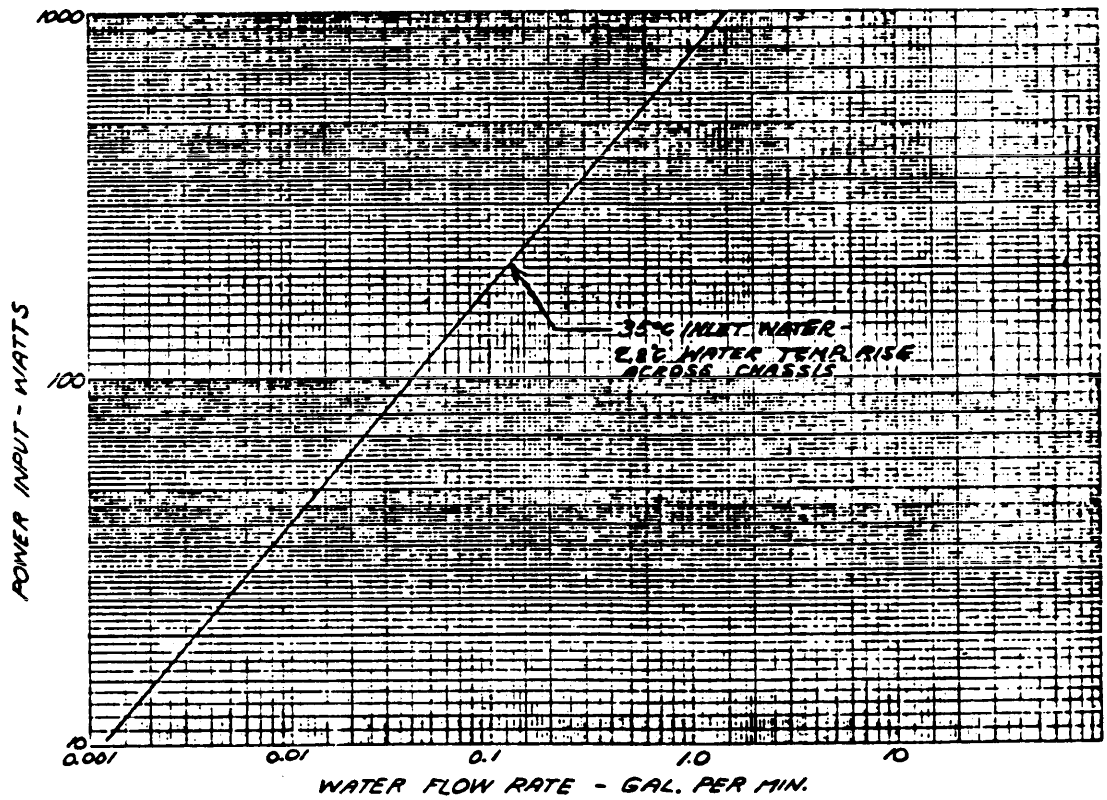


FIGURE 108. Design Curve for Cold Plate Chassis

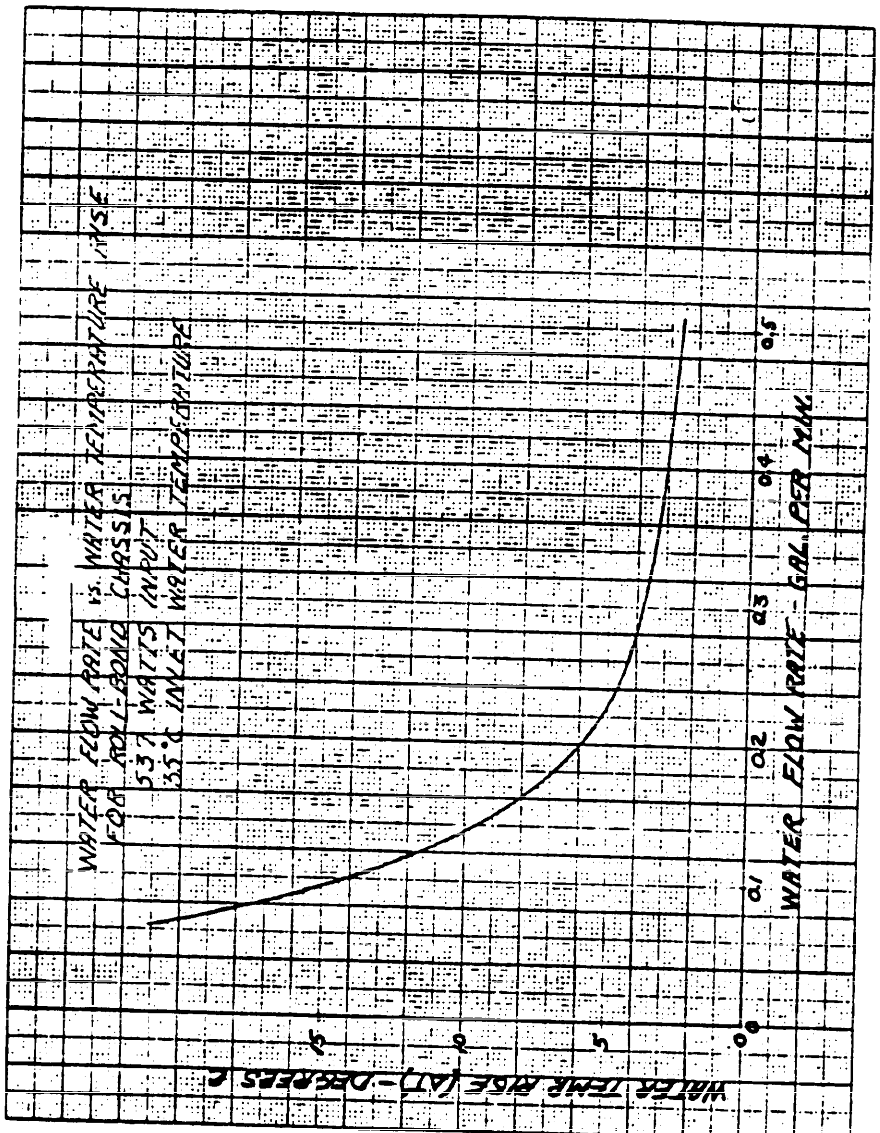


FIGURE 109. Water Flow Rate vs. Water Temperature Rise for Roll-Bond Chassis

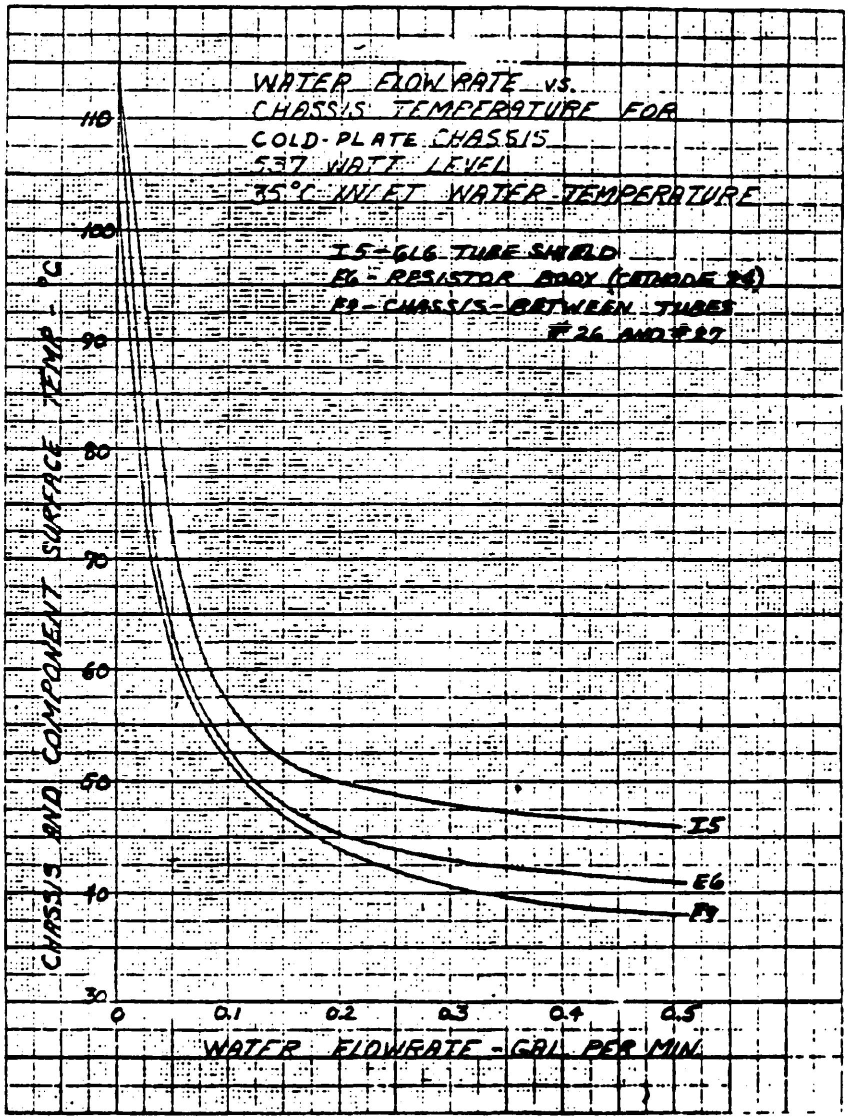


FIGURE 110. Water Flow Rate vs. Chassis Temperature For Cold Plate Chassis

is required for this chassis. The discrepancy is on the safe side, as the water flow rate can be reduced by a control valve.

Figure 111 is the measured heat transfer efficiency curve for this chassis. The percentage of the heat removed by the water increases slowly with water flow rates above 0.2 gallon per minute, approaching 69% at 0.46 gallon per minute. The remaining 31% is removed by radiation and convection to the circulating air and the water-cooled cabinet walls.

10.9.4 Design example - Shipboard cabinet. Figures 112, 113, and 114 schematically depict a system consisting of a cabinet, five electronic system chassis, and a blower and heat exchanger, designed for shipboard operation. The installation is to be in a confined space with limited air conditioning and the design requirements specify the use of the ship's fresh water cooling supply because heat loss to the compartment air is to be minimized to avoid compartment overheating.

A combination forced air and indirect liquid cooling system is to be designed, consisting of the following elements: (a) A recirculating self-contained forced air system, providing parallel forced air cooling of the five chassis. (b) A forced air-to-fresh water coolant heat exchanger, (c) Indirect liquid cooling of three high power tubes (4-250A) on a single chassis. (d) One indirect liquid cooled (cold plate) chassis for temperature sensitive components. (e) Water cooled cold plate panels in the cabinet walls to prevent heat loss to the compartment by convection and radiation.

Table XXIX summarizes some design data for the cooling system. The first line indicates the dissipation in watts for each drawer. (This system was later designed and thermally evaluated.)

The 6AQ5, 6L6, and power supply chassis were designed for forced air cooling. The second line of Table XXIX indicates the air flow rate established by test for each of these drawers, while line three indicates the air pressure drop. The flow rates indicated represent an optimum condition. The blower required for the conditions shown uses a 0.2 HP, 115V, 60 cycle, single phase synchronous air cooled motor, dissipating 500 watts. Line four indicates the heat removed by the forced air in watts.

The chassis with the 4-250A tubes presented a unique heat transfer problem. The primary dissipative elements on this chassis were the three high power 4-250A's designed for forced air cooling. While the forced air cooling system was adequate to maintain the tube temperature within acceptable limits, with these particular tubes the anodes normally operated at "cherry red" temperatures. Such high temperatures can result in overheating of adjacent components by radiating heat directly through the glass envelope. A simple shield reflected the radiated heat back to the tube, raising its temperature. Consequently, a water cooled shield was designed to absorb this heat.

The cold plate chassis design is that described in section 10.9.3.2, and 10.9.3.3.

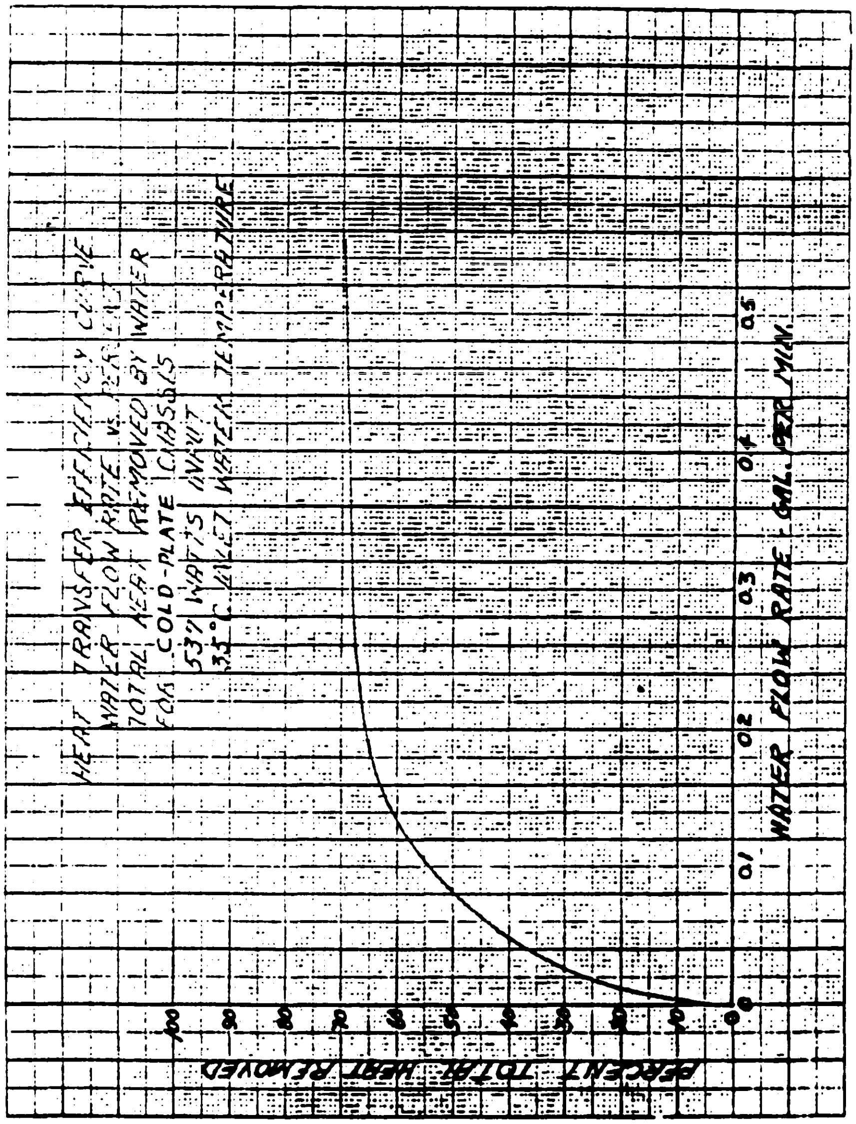


FIGURE 111. Heat Transfer Efficiency Curve Water Flow Rate vs. Percent Total Heat Removed By Water For Cold Plate Chassis

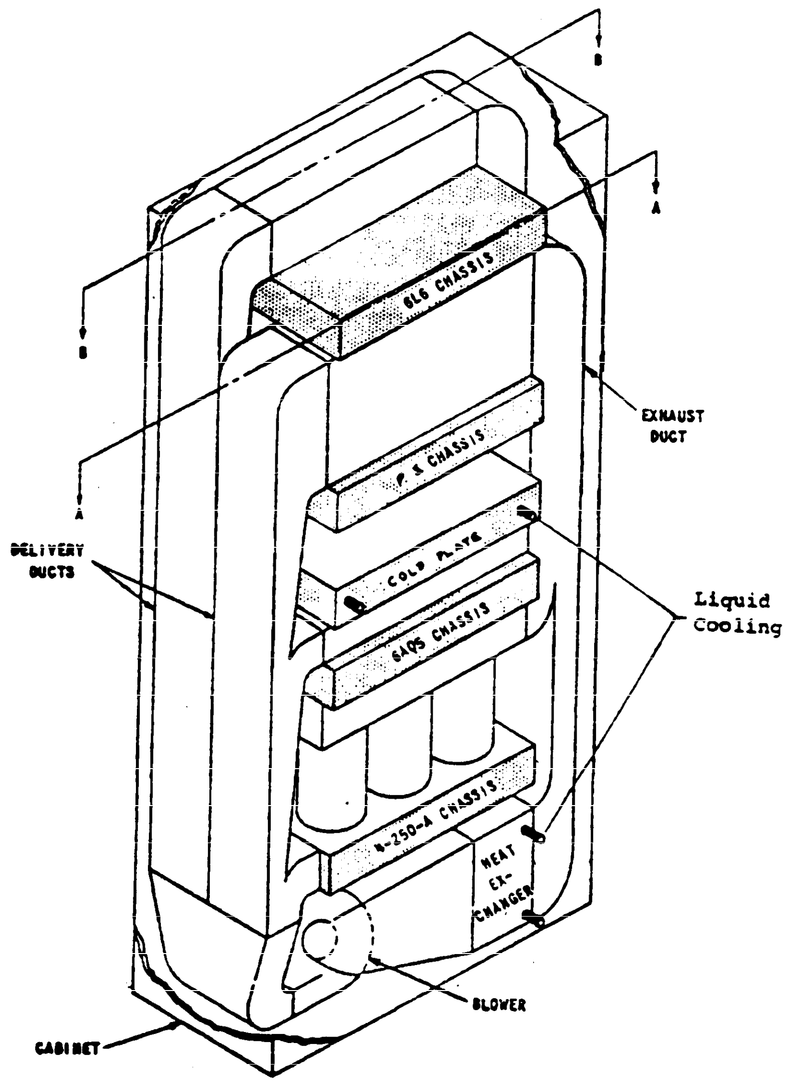


FIGURE 112. Rack-Type Forced Air Cooled Cabinet

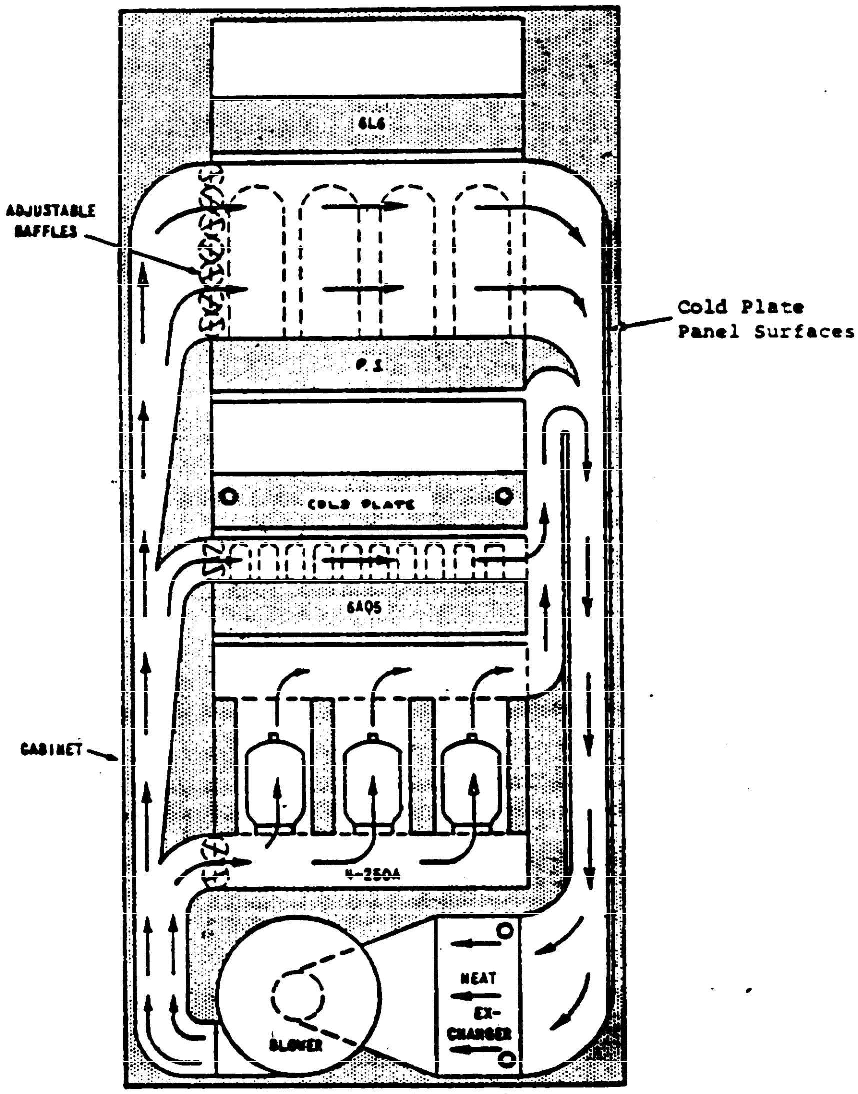


FIGURE 113. Section A-A

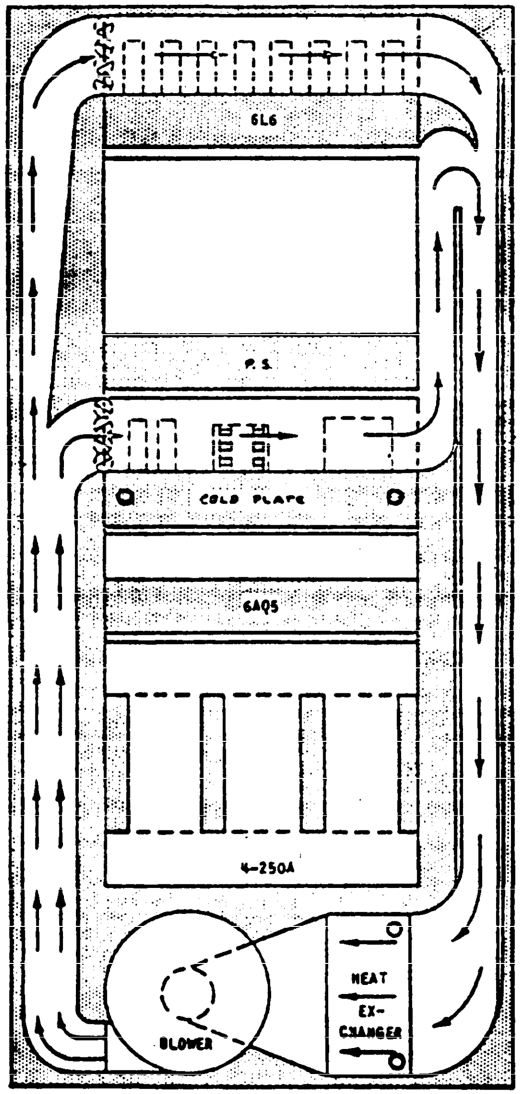


FIGURE 114. Section B-B

TABLE XXIX. Chassis Design Data

	6AQ5 Chassis	6L6 Chassis	4-250A Chassis	Cold Plate Chassis	Power Supply
Power input watts	700	676	1030	537	600
Air flow rate lb./min	4.0	4.0	9.0	1.37	3.2
Air pressure drop-in. H_2O	0.45	0.12	1.5	0.5	0.20
Heat removed by air-watts	425	405	365	166	390
Water flow rate gal./min.	-	-	0.18	0.46	-
Water pressure drop - psi	-	-	1.97	5.93	-
Heat removed by water - watts	-	-	285	371	-
Radiation and convection losses - watts	275	271	380	-	210
Air temp. rise across chassis-°C	14	5	8	16	12
Water temp. rise across chassis-°C	-	-	2.8	2.8	-
Maximum tube temperature -°C	114	96	125	68° (Tube Shield)	-

The entire system therefore utilizes indirect liquid cooling in four areas, each of which will be described in more detail in the following categories:

1) Cold plate chassis design
2) Water cooled external panels design
3) Heat exchanger design
4) Air flow ducting

10.9.4.1 Cold plate chassis. Experimental data for this chassis indicate that, with 35°C inlet water temperature, a flow rate of 0.46 gallon/min. is needed to maintain a 2.8°C water temperature rise across the chassis when operating at maximum dissipation (537 watts). At this flow rate, the chassis is 69% efficient in conducting heat to the water, which means 371 watts are absorbed by the water. The average tube shield temperature reached a maximum of 68°C.

The water pressure drop across the chassis was calculated as follows:

$$H_1 = f \frac{L}{D} \frac{V^2}{2g}$$

$$\frac{L}{D} = \frac{L}{D}_{\text{length}} + \frac{L_e}{D}_{\text{turns}}$$

The free flow cross section is A_c = 0.0354 in.^{2} (manufacturer's data). For this particular cold plate chassis, D = 0.212 in. Straight length of tube = 130 in.

45° elbows = 16	K = 0.42
60° elbows = 8	K = 0.58
90° elbows = 13	K = 0.9
180° turns = 2	K = 2.2

For water at the mean temperature of 36.4°C

ρ = 62.0 lb./ft.^{3}
μ = 1.75 lb./hr.-ft.

For a flow rate of 0.46 gallon/min. (3.99 lb./min.):

$$V = \frac{m}{A_c} = \frac{3.99(144)}{0.0354(62.0)} = 262 \text{ ft./min.} = 4.37 \text{ ft./sec.}$$

$$R_e = \frac{DV\rho}{\mu} = \frac{0.212(262 \times 60)(62.0)}{12(1.75)} = 9{,}830$$

Aluminum was considered a smooth material, so from the Moody diagram, Figure 39: F = .031

for 45° elbow - $L_e = \frac{.42(.212)}{.031} = 2.87$ in.

for 60° elbow - $L_e = \frac{.58(.212)}{.031} = 3.97$ in.

for 90° elbow - $L_e = \frac{KD}{f} = \frac{(.9).212}{.031} = 6.15$ in.

for 180° close return bend - $L_e = \frac{212(.212)}{.031} = 15.1$ in.

$$\frac{L}{D} = \frac{130}{.212} + \frac{13(6.15)}{.212} + \frac{8(3.97)}{.212} + \frac{2(15.1)}{.212} + \frac{16(2.87)}{.212}$$

$$\frac{L}{D} = 613 + 377 + 150 + 142 + 216$$

$$\frac{L}{D} = 1498$$

$$H_1 = f \frac{L}{D} \frac{V^2}{2g}$$

$$H_1 = \frac{(.031)(1498)(4.37)^2}{144} = 13.75 \text{ ft. of water}$$

$$\Delta p = \frac{(13.75)(62.0)}{144} = 5.93 \text{ psi}$$

The measure pressure drop through the chassis during tests was between 5 and 6 psi showing good correlation with theoretical calculations.

10.9.4.2 Water cooled external panels. Combined forced air and water cooling accounted for 68% of the heat dissipated within the cabinet, leaving 1136 watts to be transferred by other means (mostly radiation) to the fresh water supply. In presently designed equipment, this heat would ordinarily be absorbed by the cabinet walls and reradiated to the compartment space. To prevent this, water cooled cold plate panels were mounted on the sides, back, and top of the enclosure with a small air space between the outside cabinet walls for insulation. The cold plate panels also formed the outside walls of the delivery and exhaust ducts, thus, helping to cool the air. The inside walls of the cold plate panels were painted black to help promote radiation absorption. In order to decrease air friction within the ducts, the cold plate was flat on the inside with raised tubing on the outside. The tubing in the cold plate side panels was run vertically to reduce the number of return bends and the attendant pressure drop. The sizes of the panels were:

sides (2) = 22 in. x 60 in.
back = 23 in. x 60 in.
top = 22 in. x 23 in.

total area = 31.4 ft.^{2}

The panel tubing was connected in series with the inlet and outlet connections located near the inlet and outlet of the heat exchanger.

The required water flow rate was computed for a panel temperature equal to the average of the inlet and outlet water temperatures. The water required to absorb 1200 watts with a temperature rise of 2.8°C was found as follows: for water @ 36.4°C

$$\rho = 62.0 \text{ lb./ft.}^3$$
$$\mu = 1.75 \text{ lb./hr.-ft.}$$
$$c_p = 31.6 \text{ watt-min./lb.-°C}$$

since $q = mc_p\Delta T$

$$m = \frac{1200(60)}{31.6(2.8)} = 814 \text{ lb./hr.}$$

Add 10% for film resistance between aluminum and water.

$$m = 895 \text{ lb./hr.}; \; Q = 1.8 \text{ GPM}$$

$$Q = \frac{895}{3600(62.0)} = 4.01 \times 10^{-3} \text{ ft.}^3\text{/sec.}$$

The water pressure drop in the cold plate panels is given by:

$$H_1 - f\frac{L}{D}\frac{V^2}{2g} + K\frac{V^2}{2g}$$

The cold plate panels (Figure 115) have tubes with cross sectional flow areas equivalent to a semicircular tube of:

$$d = 0.75 \text{ in.}$$

$$A_c = \frac{1}{2}\left(\frac{\pi d^2}{4}\right) = .221 \text{ in.}^2$$

$$V = \frac{m}{A_c} = \frac{4.01 \times 10^{-3}\,(144)}{.221} = 2.61 \text{ ft./sec.}$$

Since $R_e = \frac{VD\rho}{\mu}$

Where D = 4 (Hydraulic Radius)

$$= 4\left(\frac{A_c}{\text{perimeter}}\right)$$

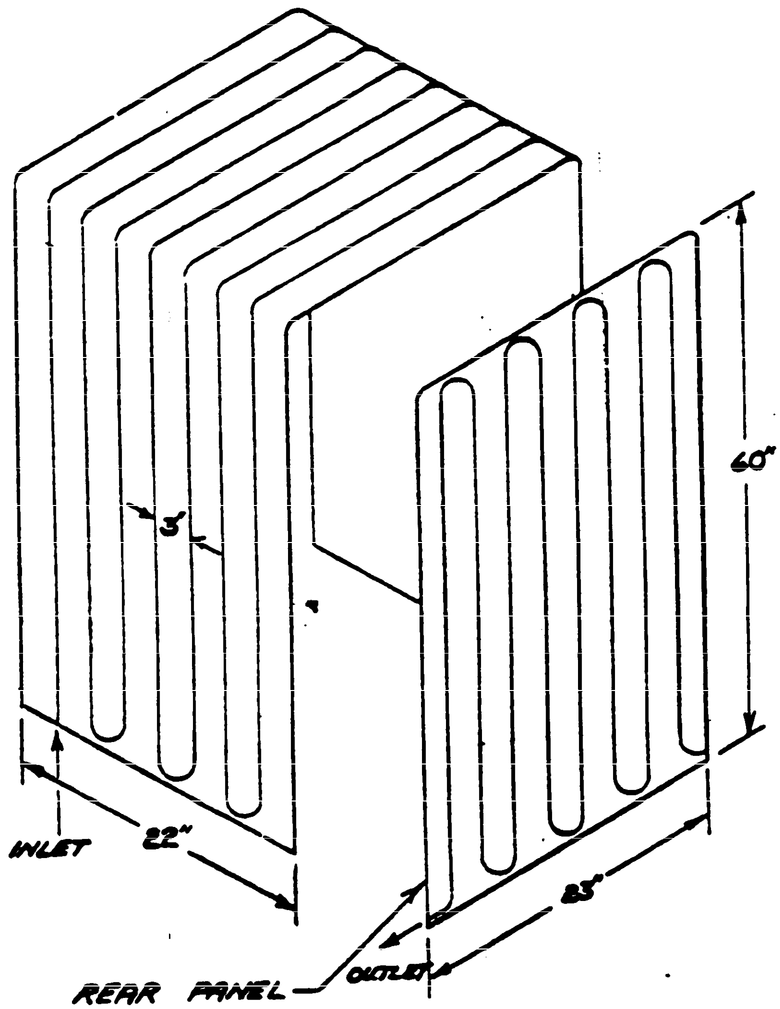


FIGURE 115. Cooling Panel on Cabinet

$$D = 4 \frac{.221}{\frac{\pi(.75)}{2} + .75} = .458 \text{ in.}$$

$$R_e = \frac{3600(2.61)(.458)(62.0)}{12(1.75)} = 12{,}700$$

From Moody Diagram (Figure 39) for smooth pipe:

$$f = .029$$

Total length of tube - 1500 in.

13 - 180° close return bends	K = 2.2
18 - 90° elbows	K = 0.9

$$H_1 = (.029)\frac{1500}{.458}\frac{(2.61)^2}{64.4} + 13(2.2)\frac{(2.61)^2}{64.4}$$

$$+ 18(0.9)\frac{(2.61)^2}{64.4}$$

$$H_1 = 10.0 + 3.0 + 1.2 = 14.7 \text{ ft. } H_2O$$

$$\Delta p = 6.4 \text{ psi}$$

10.9.4.3 Heat exchanger design. The heat exchanger must cool the hot exhaust air to a temperature approaching that of the water. It is desirable to have a heat exchanger with high effectiveness; however, this is possible only at the expense of increased size and weight of the exchanger or by increased coolant flow. Thus, the selection of a heat exchanger involves a compromise. (See section 10.4)

Theoretically, the counter-flow arrangement affords the highest possible effectiveness. Unfortunately, advantages of this arrangement cannot be realized in practice because of the difficulty of header design problems but has lower effectiveness.

10.9.4.3.1 Detailed design of heat exchanger. The total heat load to be removed by forced air cooling was:

425 watts-6AQ5 chassis
405 watts-6L6 chassis
166 watts-cold plate chassis
365 watts-4-250A chassis
390 watts-power supply
500 watts-blower motor
2,251 watts-total heat load

The following air flow rates for the individual chassis are required at 40°C:

4.0	lb./min. -	6AQ5 chassis
4.0	lb./min. -	6L6 chassis
1.4	lb./min. -	cold plate chassis
9.0	lb./min. -	4-250A chassis
3.2	lb./min. -	power supply
21.6	lb./min. -	total plus 10% for circulation between chassis and leakage = 23.8 lb./min.

$$\Delta t = \frac{Q}{c_p m}$$

$$\Delta t \text{ for air (through cabinet)} = \frac{2251}{7.6(23.8)} = 12.4°C$$

Therefore, the inlet air temperature to the heat exchanger will be 52.4°C and the outlet temperature desired is 40°C. The Δt across exchanger = 12.4°C with 1428 lb./hr. air flow.

The fresh water which will flow through this heat exchanger will come from the ship's fresh water supply. Inlet water has a maximum temperature of 35°C, which is 17.4°C below the inlet air temperature. A 2.8°C water temperature rise and approximately 5 to 6 psi pressure drop are permitted through the exchanger. The water leaving the exchanger is to be at 37.8°C, or 2.2°C below the leaving air temperature.

For water @ 36.4°C:

$$c_p = 31.6 \text{ watt-min./lb.-°C}$$

$$m = \frac{q}{c_p \Delta t} = \frac{2251}{(31.6)(2.8)} = 25.4 \text{ lb./min.}$$

$$Q = \frac{m}{\rho'} = \frac{25.4}{8.3} = 3.06 \text{ gallon/min.}$$

$$\text{or } \frac{3.06}{2.25} = 1.36 \text{ gallon/min./kw.}$$

Where ρ' = 8.3 lbs./gallon

The flow in the heat exchanger tubes should be turbulent to insure efficient heat transfer. Figure 116 illustrates the variation in Reynolds number obtained in tubes with inside diameters from 0.25 to 0.75 inch with flow rates of 400 lbs./hr. to 800 lbs./hr. The curves are drawn for an absolute viscosity of 1.75 lb./hr.-ft., corresponding to 36.4°C, the average temperature for the water in the system.

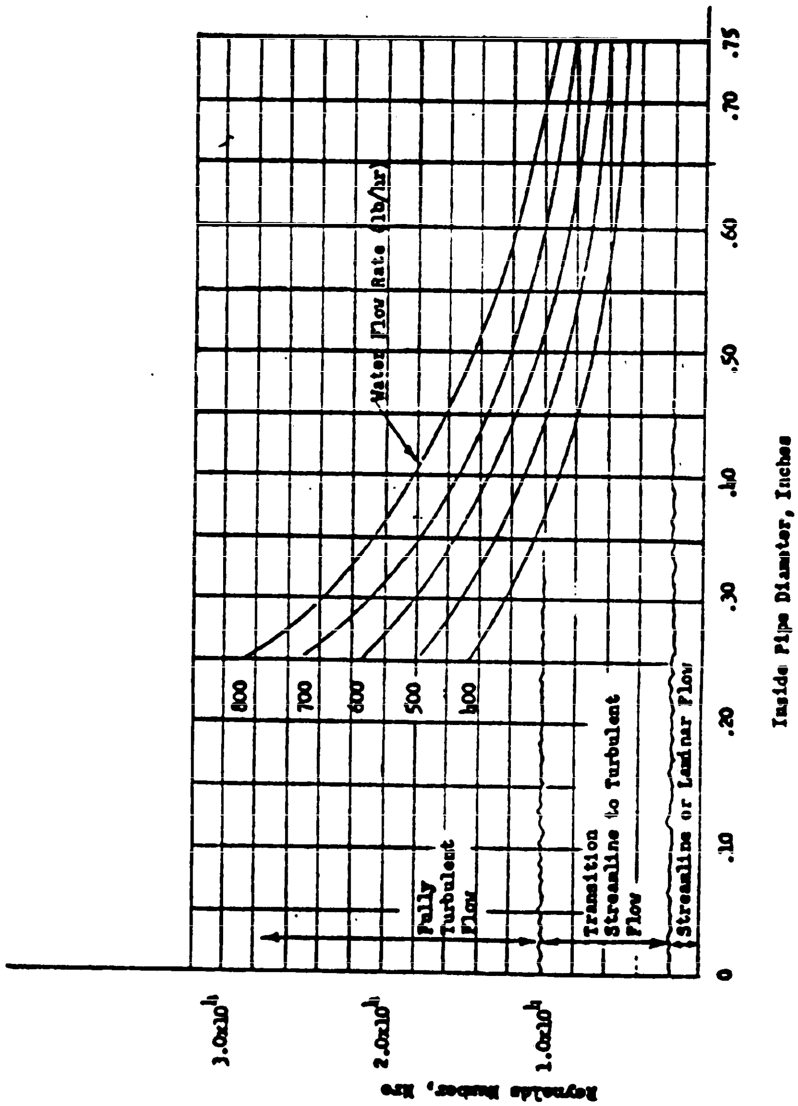


FIGURE 116. Reynolds Number vs. Water Flow Rate

Two design criteria must be considered when the heat exchanger dimensions are calculated:

(1) It is desirable that the air flow at a high velocity over the surface of the coils. Reynolds number values above 10,000 insure turbulent flow, thus eliminating a stagnant air film on the surface of the coil and increasing the local surface heat transfer coefficient. Surface air velocities are usually limited to 800 ft./min. because of increased noise at speeds in excess of this figure.

(2) The heat transfer coefficient for the air side of an air-water heat exchanger tube will generally be significantly less than that for the water side. Consequently, it is necessary to increase the surface area of the air side of the tubes to obtain good efficiency. This leads to finned tubing and plate-coil type of exchanger surfaces.

The length of the heat exchanger in the direction of the air flow is determined by the coil characteristics, air and water design flow rates, heat to be dissipated, size of tubing, and grouping of tubes in air flow path.

The heat exchanger will be designed for the following conditions:

air flow	- 1428 lb./hr.
air velocity	- 2000 ft./min.*
heat to be removed	- 2251 watts
air inlet temperature	- 52.4°C
air outlet	- -40°C
water inlet temperature	- 35°C
water outlet temperature	- 37.8°C
water flow rate	- 3.06 gallon/min.

The following problem is the design of a heat exchanger for the rack-cabinet under discussion, using the recommended ϵ -NTU** approach, which is described in Reference 9 and 54. All tables, figures, and equations referenced in this problem are from Reference 54.

*The air velocity should normally be limited to 800 ft./min. In this example, a high velocity was used because of space limitation.

**Exchanger effectiveness - Number of Transfer Units.

<u>air side</u>

air flow rate	- 1428 lb./hr. = 23.8 lb./min.
air inlet temperature	- 52.4°C
air outlet temperature	- 40°C
air velocity	- 2000 ft./min.

<u>water side</u>

water flow rate	- 3.06 gallon./min. = 25.4 lb./min.
inlet temperature	- 35°C
outlet temperature	- 37.8°C

The exchanger will be fabricated of 5/8" I.D. aluminum tubing with 7.75 fins per inch, and the fins are the continuous type as illustrated on page 117, Figure 101, Reference 54.

The objective of this problem is to predict for the specified conditions, and the basic heat transfer and flow friction characteristics of the surface: 1) the intercooler effectiveness, 2) area needed and, 3) the pressure drops for both water and air. This prediction will be based on clean surfaces with no allowance for fouling.

The ϵ - NTU solution of the problem requires the determination of the following factors in the order given:

(1) Bulk average temperatures (air and water)
(2) Exchanger effectiveness - ϵ
(3) Capacity rate ratio - C_{min}/C_{max}
(4) Number of thermal units - NTU
(5) Overall coefficient of heat transfer - U
(6) Air side area - A
(7) Heat exchanger dimensions
(8) Pressure drops (water side and air side)

10.9.4.3.2 Determination of bulk average temperatures. Figure 117 shows the temperature distribution in the heat exchanger (drawn as for true counterflow, although the actual exchanger has a crossflow arrangement).

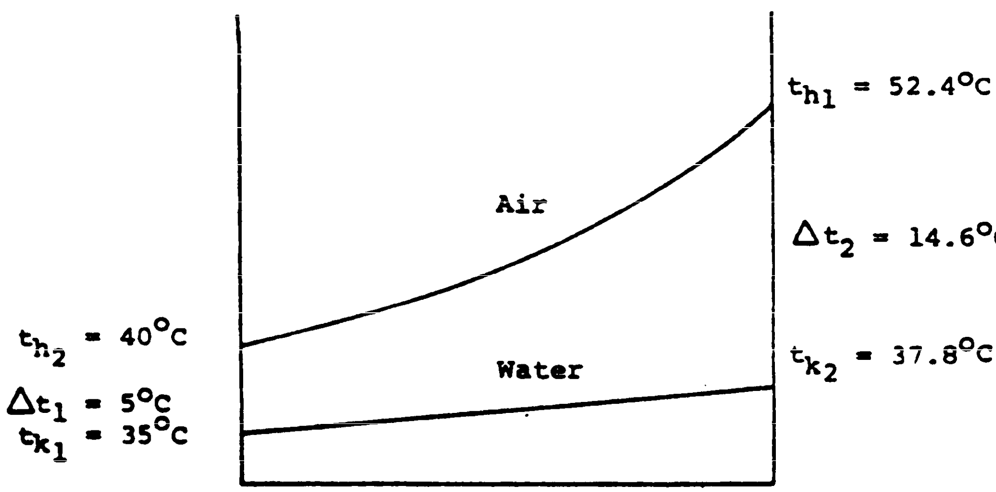


FIGURE 117. Temperature Gradients in Heat Exchanger

Since the water temperature rise is small, a linear rise may be assumed and an arithmetic mean may be used.

Since the air temperature rise is large, the logarithmic mean should be used.

The average bulk temperatures are:

$$t_k = \frac{t_{k_1} + t_{k_2}}{2} = \frac{35 + 37.8}{2} = 36.4°C$$

$$t_h = t_k + \frac{\Delta t_2 - \Delta t_1}{\ln \frac{\Delta t_2}{\Delta t_1}} = 36.4 + \frac{(14.6 - 5)}{\ln \frac{14.6}{5}}$$

$$= 36.4 + \frac{9.6}{1.07} = 45.4°C$$

10.9.4.3.3 Determination of exchanger effectiveness - ϵ.

For air @ 45.4°C	c_p = 7.6 watt-min./lb.-°C
Air side capacity rate	$C_h = m_h c_p$ = 23.8(7.6) = 181 watt/°C
For water @ 36.4°C	c_p = 31.6 watt-min./lb.-°C
Water side capacity rate	$C_k = m_k c_p$ = 25.4(31.6) = 803 watt/°C

$$\epsilon = \frac{C_k}{C_h} \frac{t_{k_2} - t_{k_1}}{t_{h_1} - t_{k_1}} = \frac{803(37.8-35)}{181(52.4-35}$$

$$\epsilon = \frac{803(2.8)}{181(17.4)} = 71.4\%$$

10.9.4.3.4 Determination of capacity rate ratio.

$$\frac{C_{min}}{C_{max}} = \frac{C_h}{C_k} = \frac{181}{803} = 0.225$$

10.9.4.3.5 Determination of number of thermal units - NTU. From Figure 5 or Table IV of Reference 54 (crossflow exchanger with both fluids unmixed), the NTU corresponding to ϵ = 71.4% and C_{min}/C_{max} = 0.225 is:

$$NTU = 1.5$$

10.9.4.3.6 Determination of U. The air side surface characteristics of the type of heat exchanger chosen for this problem are:

Air Side (Table 14, Reference 54)

flow passage hydraulic radius,	$HR_h = 3.17 \times 10^{-3}$ ft.
total air side transfer area/total volume	$\alpha_h = 157$ ft.^{2}/ft.^{3}
fin area/total area	$A_f/A = 0.905$
free flow area/frontal area	$\sigma h = 0.497$
fin metal thickness	$\delta = 0.016$ in. $= 0.0013$ ft.
Fin material, aluminum	$k = 4.4$ watt-in./in.^{2}-°C
Fin length, 1/2 dist. between tubes	$L_f = 0.0341$ ft.

Assuming the air is dry @ 45.5°C, then:

$\mu = 0.0470$ lb./hr.-ft.
$c_p = 7.6$ watt-min./lb.-°C
$P_r = 0.70$
$\rho = 0.070$ lb./ft.^{3}

Reynolds Number (air side)

Air velocity will be limited to 2000 ft./min.
Cross section free flow area - $A_c = \frac{m_h}{V}$

$$A_c = \frac{23.8}{(0.070)(2000)} = .17 \text{ ft.}^2 = 24.5 \text{ in.}^2$$

$$G_h = \frac{m_h}{A_c} = \frac{23.8}{.17} = 140 \frac{\text{lb.}}{\text{min.-ft.}^2} = 8{,}400 \text{ lb./hr.-ft.}^2$$

$$Re = \frac{4HR_hG_h}{\mu} = \frac{4(3.17 \times 10^{-3})8400}{0.0470} = 2266$$

Stanton Number (air side)

From Figure 101 of Reference 54 with $Re_h = 2266$

$StPr^{2/3} = 0.0067$

$Pr = 0.70$; $Pr^{2/3} = 0.788$

$St = \frac{0.0067}{0.788} = 8.50 \times 10^{-3}$

Unit Film Conductances (air side)

$$h_h = c_pG_hSt = 7.6(140)(8.50 \times 10^{-3}) = 9.04 \frac{\text{watts}}{\text{ft.}^2\text{-°C}}$$

Fin Effectiveness (Use Figure 94)

For straight fins, $m = 2h_h/k\delta$

$$k = 4.4 \text{ watt-in./in.}^2\text{-°C} = 52.8 \frac{\text{watt-ft.}}{\text{ft.}^2\text{-°C}}$$

$$m = 2(0.04)/52.8(0.0013) = 16.2 \text{ ft.}^{-1}$$

$$mL_f = 16.2(0.0341) = 0.552$$

From Figure 94 $\eta_f = 0.92$

Overall Surface Effectiveness

$$\eta_o = 1 - \frac{A_f}{A}(1 - \eta_f)$$

$$\eta_o = 1 - 0.905(1-0.92) = 1-0.07$$

$$= 0.93$$

The water side characteristics are from Figure 101 of Reference 54.

Outside diameter	= 0.676 in.
Inside diameter	= 0.625
Frontal area associated with one tube	= 2.63 in.^{2}
Free flow area of one tube	A_c = 0.307 in.^{2}
Inside periphery of one tube	= 1.96 in.
Free flow area/frontal area	σk = 0.117
Water side transfer area/total volume	αk = 8.95 ft.^{2}/ft.^{3}
Water side flow passage hydraulic radius	HR_k = 0.013 ft.

Reynolds Number (water side)

Flow rate $m_k = 25.4$ lb./min

For water @ 36.4°C

$$\mu = 1.75 \text{ lb./hr.-ft.}$$
$$\rho = 62.0 \text{ lb./ft.}^3$$
$$c_p = 31.6 \text{ watt-min./lb.-°C}$$
$$Pr = 4.72$$
$$A_c = 0.307 \text{ in.}^2 = 2.13 \times 10^{-3} \text{ ft.}^2$$

$$Gk = \frac{m_k}{A_c} = \frac{25.4}{2.13 \times 10^{-3}} = 11{,}900 \text{ lb./min.-ft.}^2 = 7.14 \times 10^5 \text{ lb./hr.-ft.}^2$$

$$Re = \frac{4HR_kG_k}{\mu} = \frac{4(0.013)(7.14)10^5}{1.75}$$

$$= 21{,}200 \text{ (turbulent flow)}$$

Stanton Number (water side)

From Figure 28 of Reference 54 for turbulent flow inside of circular tubes, with Re_k = 21,000 and Pr = 4.72:

$$10^4 St \left(\frac{\mu w}{\mu m}\right)^n = 14 \text{ (use n=0.20 since water is being used for cooling)}$$

Since at high Prandtl numbers, wall temperature variations are negligible.

$$\left(\frac{\mu w}{\mu m}\right)^{0.20} \approx 1$$

$$St = 0.0014$$

Unit Film Conductance (water side)

$$h_k = St\, G_k c_p = (0.0014)\ \frac{7.14 \times 10^5}{60} \quad (31.6)$$

$$h_k = 526 \text{ watts/ft.}^2\text{-°C}$$

Overall Coefficient of Heat Transfer U

$$\frac{1}{U} = \frac{1}{n_o h_h} + \frac{1}{\frac{\alpha k}{\alpha h} h_k}$$

$$\frac{1}{U} = \frac{1}{0.93(9.04)} + \frac{1}{\frac{8.95}{157} 526}$$

$$\frac{1}{U} = 0.119 + .033 = 0.152$$

$$U = 6.58 \text{ watt/ft.}^2\text{-°C}$$

10.9.4.3.7 Determination of air side area - A.

$$A = NTU\ \frac{C_{min}}{U} = NTU\ \frac{C_h}{U} = \frac{1.5(181)}{6.58}$$

$$A = 41.3 \text{ ft.}^2$$

10.9.4.3.8. Determination of heat exchanger dimensions.

Frontal area $A_{fr} = \frac{A_c}{\sigma_h} = \frac{24.5}{0.497} = 49.3 \text{ in.}^2$

The height of the heat exchanger is predetermined by the duct height (4.5 in.).

$$\text{width} = \frac{A_{fr}}{\text{height}} = \frac{49.3}{4.5} = 11.0 \text{ in.}$$

Exchanger volume $V = \frac{A}{\alpha_h} = \frac{41.3}{157} = 0.263\ \text{ft.}^3 = 454\ \text{in.}^3$

$$\text{depth} = \frac{V}{A_{fr}} = \frac{454}{49.3} = 9.2\ \text{in.}$$

The exchanger, therefore, is of the following dimensions:

4.5 in. high x 11.0 in. wide x 9.2 in. deep

10.9.4.3.9 Determination of pressure drops

Water side

$$H_1 = f\frac{L}{D}\frac{V^2}{2g} + K_t\frac{V^2}{2g}$$

Number of tubes in heat exchanger $= \frac{\text{depth x height}}{\text{frontal area/tube}}$

$$= \frac{9.2(4.5)}{2.63} = 15.7$$

Therefore, there are 16 tubes, 11.0 in. long, with 15 close return elbows ($K = 2.2$).

Total length of tubes, $L = 16(11) = 176$ in.
Loss coefficient $K_T = 15(2.2) = 33$

$$Q = \frac{m_k}{\rho} = \frac{25.4}{62.0} = \frac{.41\ \text{ft.}^3}{\text{min.}} = 6.82 \times 10^{-3}\ \frac{\text{ft.}^3}{\text{sec.}}$$

$$V = \frac{Q}{A_c} = \frac{6.82 \times 10^{-3}}{2.13 \times 10^{-3}} = 3.2\ \frac{\text{ft.}}{\text{sec.}}$$

$D = 0.625$ in. (inside diameter)

From Moody Diagram (Figure 39) with Re = 21,000, for smooth pipes, $f = 0.026$.

$$H_1 = 0.026\ \frac{176}{0.625}\ \frac{(3.2)^2}{64.4} + 33\ \frac{(3.2)^2}{64.4}$$

$H_1 = 1.16 + 5.25 = 6.41$ ft. of water

$\Delta p = 2.77$ psi

The pressure drop in the tubing connecting the heat exchanger to the fresh water supply has not been included in the 2.77 psi. A flow-regulating valve may be required at the inlet to the exchanger to permit a 3.06 GPM flow rate. The pressure drop due to a regulator valve and the connecting tubing will not raise the overall pressure drop above the 5 psi maximum.

Air side

Neglecting entrance and exit losses which would be small for this type of surface and exchanger geometry, the air pressure drop is given by equation 28a of Reference 54.

$$\Delta p = \frac{G^2}{2g} \; v_1 \left[(1 + \sigma_h{}^2) \left(\frac{v_2}{v_1} - 1 \right) + f \frac{A}{A_c} \left(\frac{v_m}{v_1} \right) \right]$$

The maximum head loss through the ducts and chassis will be 2.0 in. of water. Assuming 1.0 in. of water head loss through the exchanger and standard atmospheric pressure:

inlet: $p_1 = \dfrac{34 + \frac{1}{12}}{34} \; 14.7 = 14.74$ psia

outlet: $p_2 = 14.7$ psia

specific volume: (v)

inlet: $v_1 = \dfrac{RT_1}{p_1} = \dfrac{53.34(586)}{14.7(144)} = 14.73$ ft.^{3}/lb.

outlet: $v_2 = \dfrac{RT_2}{p_2} = \dfrac{53.34(564)}{14.7(144)} = 14.21$ ft.^{3}/lb.

mean specific volume: (v_m)

$$\frac{v_m}{v_1} = \frac{p_1}{p_{avg}} \times \frac{T_h}{T_1}$$

where: $T_h = 574°R$, $p_{avg} = \dfrac{p_1 + p_2}{2} = 14.72$ psia

$$\frac{v_m}{v_1} = \frac{14.74}{14.72} \times \frac{574}{586} = 0.981$$

$$\frac{A}{A_c} = \frac{\text{exchanger depth}}{\text{flow passage HR}} = \frac{9.2}{12(3.17 \times 10^{-3})} = 242$$

$$(1 + \sigma_h{}^2) = 1 + (0.497)^2 = 1.247$$

From Figure 101 of Reference 54 with $Re_h = 2262$, $f = 0.017$

$$\Delta p = \left(\frac{140}{60}\right)^2 \frac{14.73}{64.4}\left[1.247\left(\frac{14.21}{14.73} - 1\right) + 0.017(242)(0.981)\right]$$

$$\Delta p = 1.24\ (-0.044 + 4.04) = 4.96\ \text{lb./ft.}^2 = 0.0345\ \text{psi}$$

$$H_1 = 0.965\ \text{in. of water}$$

The above calculation makes use of the method of successive approximations. Since the calculated pressure drop is so close to the assumed pressure drop, a second calculation is not required.

Figure 118 is a schematic diagram of the piping system needed within the cabinet.

10.9.4.4 Ducting. The blower will be delivering 23.8 lb./min. @ 40°C, or 338 ft.^{3}/min. to the chassis and the maximum allowable velocity is 2000 ft./min. for aluminum duct work; therefore, the minimum duct cross section will be:

$$A = \frac{338(144)}{2000} = 24.3\ \text{in.}^2$$

The delivery duct will be 10 in. wide x 2.5 in. deep and will diverge to a 19 in. x 2.5 in. duct as shown in Figure 114. The exhaust duct will be handling 23.8 lb./min. of 52.4°C air, or

$$\rho \text{ for air @ } 52.4°\text{C} = .0674\ \text{lb./ft.}^3$$

$$Q = \frac{23.8}{.0674} = 353\ \text{ft.}^3/\text{min.}$$

$$\text{minimum duct area} = \frac{353(144)}{2000} = 25.4\ \text{in.}^2$$

or a 10 in. x 2.5 in. duct.

10.9.5 Design example, airborne. A TWT (traveling wave tube) in an airborne electronic system uniformly dissipates a total of approximately 6230 watts from the collector in a continuous duty mode. The collector consists basically of three parts, as shown in Figure 119: (1) An inner copper cone which contains 24 integral slots machined into its outer surface for cooling channels, (2) a thin copper manifold separator cone which seals the open sides of the cooling channels to form rectangular passages, and (3) an outer stainless steel cone, forming an annular coolant passage with the manifold separator.

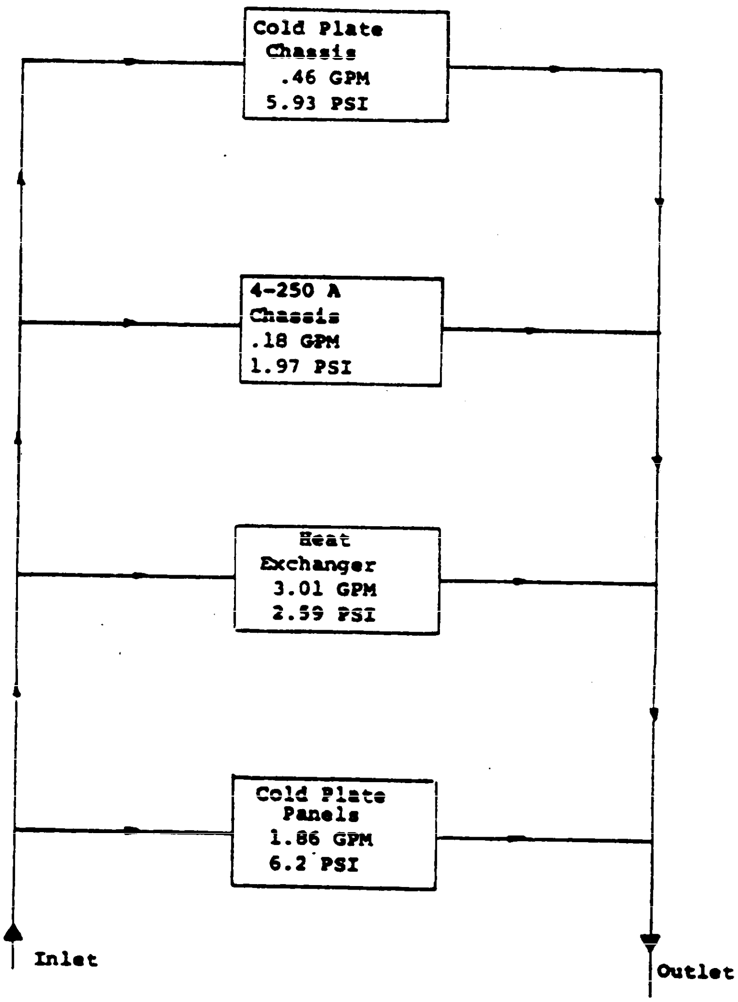


FIGURE 118. Schematic Diagram of Piping System

Coolant is available in the form of Coolanol 35 (or equal) with the inlet temperatures ranging from -20PC to +110°C. Coolant is supplied by a constant pressure pump.

Design limitations are a maximum of 250°C collector temperature, to achieve the required tube reliability. In addition, the Coolanol 35 (or equal) has a temperature limit of approximately 220°C (423°F). Exceeding this coolant temperature for any extended period of time will result in coking deposits on the surface of the coolant passages. These deposits increase the thermal resistance at the collector surface, and also increase the pressure drop across the coolant passages. For a constant pressure pump the latter influence decreases the coolant flow rate.

The problem is one of determining the maximum collector and coolant temperatures under conditions of varying coolant inlet temperatures to insure that the maximum allowable temperatures (collector and coolant) are not exceeded. (NOTE: in the actual design upon which this example is based, numerous other parameters were varied, such as the number of collector cooling slots, coolant flow direction, and varying width annular passages. For simplicity, only the final design is presented herein).

The properties of Coolanol 35 (or equal) vary with temperature, so that different flow rates will result from varying inlet temperatures with a constant pressure pump. Flow rates of the coolant were determined in model tests, and are shown in Figure 120 as a function of coolant inlet temperatures.

The equations on which the analysis was based are:

(a) For coolant flowing through slots:

(1) Turbulent flow

$$h = 0.023 \frac{k}{D} (Pr)^{.4} (Re)^{.8}$$

(Reference equation 10-11)

(2) Laminar flow

$$\frac{hD}{k} = 3.65 + \left(\frac{0.0668 \ (D/L) \ RePr}{1 + 0.04 \ [(D/L)(Re)(Pr)]^{0.67}} \right)$$

(b) For annular flow:

$$h = 0.023 \ (c_p) \ G \ Pr^{-0.67} \ Re^{-0.2} \left(\frac{D_2}{D_1} \right)^{0.45}$$

(Reference 25. Note similarity to equation 10-14)

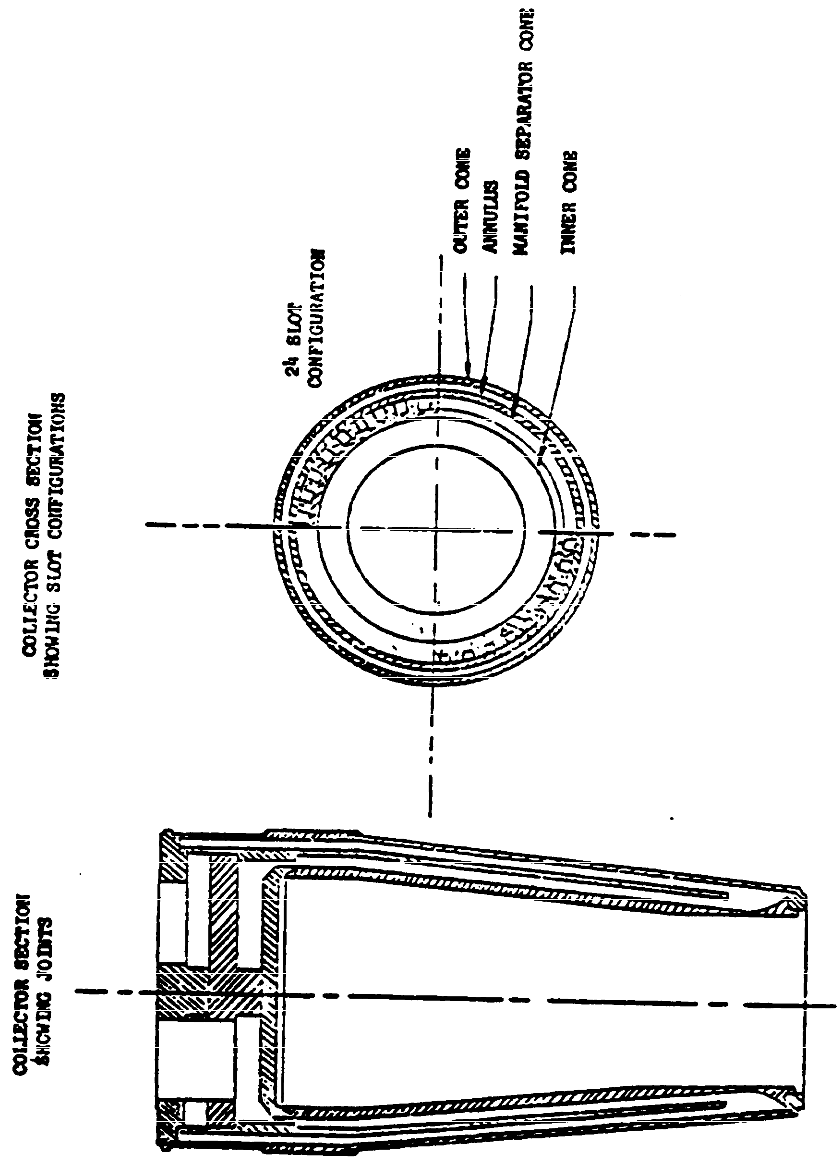


FIGURE 119. Collector Sections

(In all equation, D is the equivalent diameter, or hydraulic radius.)

All values of h were modified by the factor $\frac{\mu_B}{\mu_W}^{.14}$

Where μ_B is the dynamic viscosity at bulk coolant temperatures, and μ_W is the dynamic viscosity at the collector wall temperature. This term is negligible for most problems, but should be included when coolant properties are strongly temperature dependent or when large temperature differences exist between the wall and the bulk fluid. (Reference 26, chapter 9)

Note that flow within the slots may be either laminar or turbulent, dependent on flow rates (which are variable) and fluid properties (which also vary with temperature). Both types of flow may occur simultaneously at different points within the passage, or the fluid properties change due to temperature increase.

This expamle illustrates the complexity often encountered in heat transfer problems. The varying coolant inlet temperatures cause variations in flow rates and coolant properties. These in turn affect the heat transfer coefficient, and even the basic equation by which this coefficient is derived. Because of the complex interaction of variables, and the number of parameters to be evaluated, this problem was programmed for computer solution, using nodal analysis techniques. A mathematical thermal model of the TWT was constructed, consisting of 23 nodal points selected in a manner to depict the temperature distribution of the collector and the coolant bulk temperature. The computer continually re-evaluated the convection coefficients as temperatures varied, for all applicable node interactions. Results are shown in Figure 121, depicting the temperature distribution for a typical run, and Figure 122, which shows the variation in collector temperature with coolant inlet temperature for a particular configuration.

Certain trends are apparent from Figure 121. Beginning with the lowest coolant inlet temperature, collector temperatures are high due to the low flow rates. As the coolant inlet temperature rises, the flow rate increases and the collector temperature drops. At about 0 to 10°C, the increasing coolant temperature is more effective than the rate of increase in flow, and collector temperatures rise again. A peak is reached at a coolant inlet temperature of about 60°C beyond which turbulent flow develops, yielding increased convection coefficients and reduced collector temperatures. Another collector temperature rise is encountered beyond about 90°C, again due to the increasing coolant inlet temperature.

This design was acceptable in maintaining collector and coolant temperatures below the allowable limits.

10.9.6 Design applications - spacecraft. An electronic system designer rarely has direct and independent control over the heat transfer aspects of his system when applied to a spacecraft. Since weight, volume, and power are at such a premium in these applications, and since the ultimate sink is limited by vehicle radiation effects, the thermal design is generally controlled by the vehicle designer or at least, must be very closely

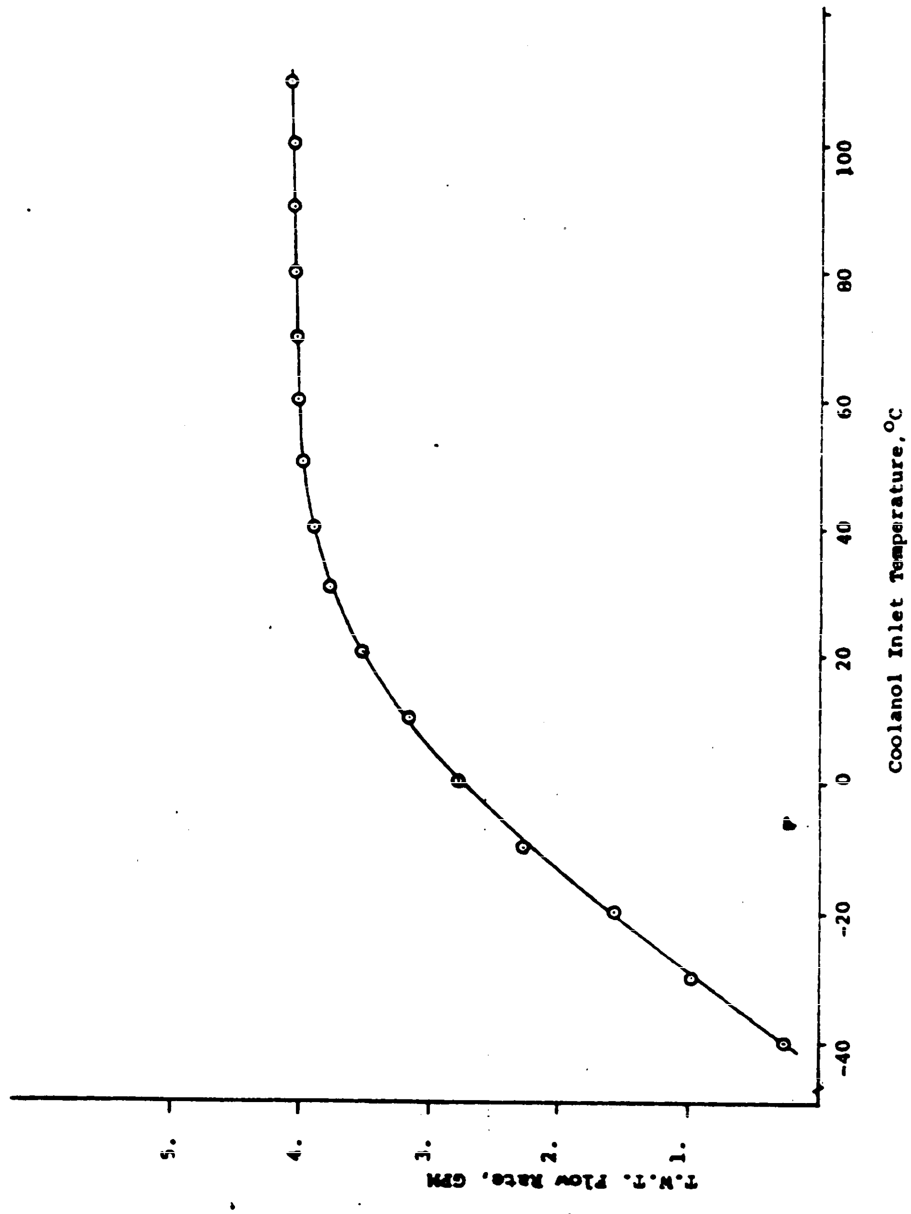
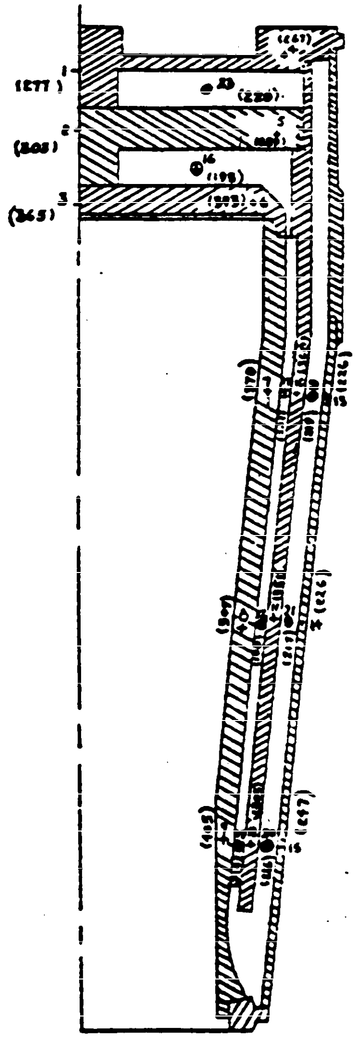


FIGURE 120. Coolant Flow Rate vs. Inlet Temperature



Legend

✚ Denotes Copper
⊕ Denotes Coolant
Numbers in ( ) Represent Steady State Temperature in °F at the associated nodal point.

Operation Conditions

Power Dissipation: 6000 Watts
Coolant Temperature: 194°F (90°C)
Coolant Flow Rate: 3.6 Gal/Min
Number of Slots: 24
Annulus: Restricted
Flow thru Slots first

FIGURE 121. Thermal Mapping - Run No. 1

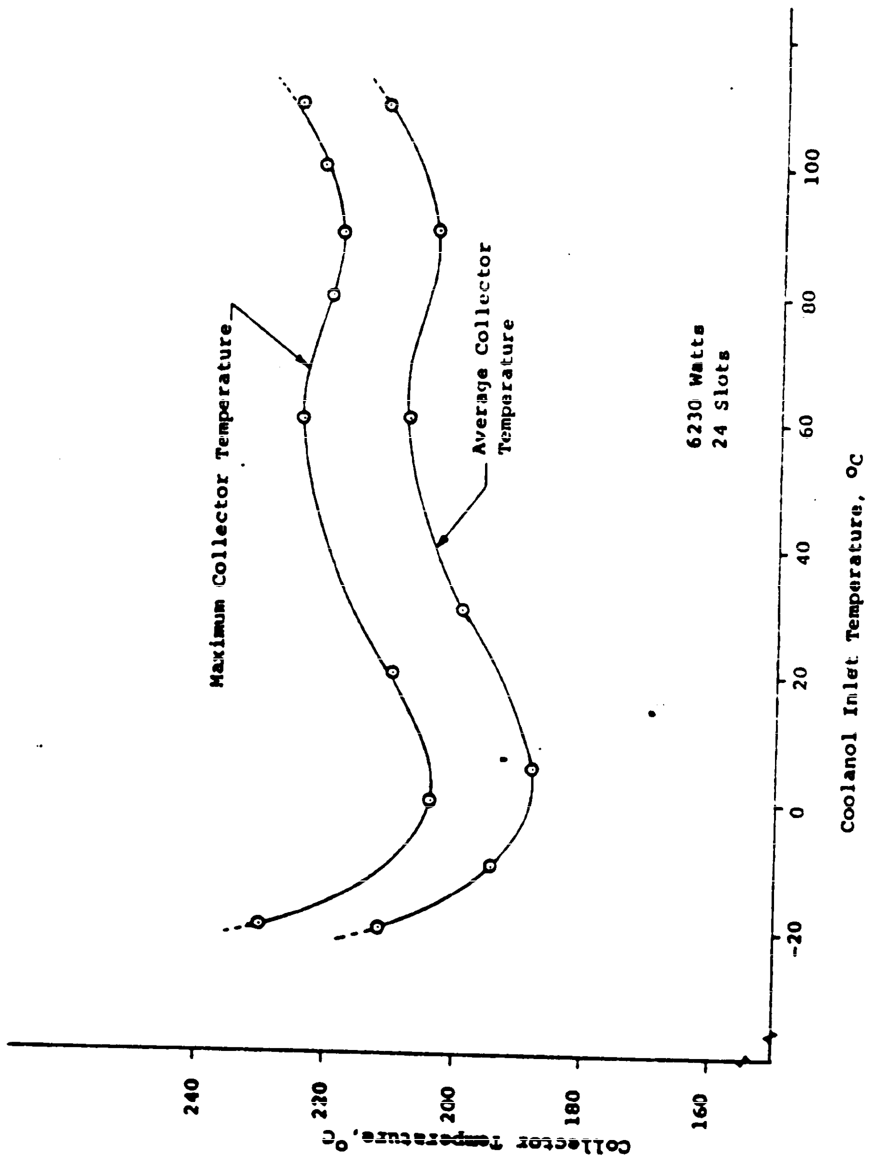


FIGURE 122. Typical Computed TWT Collector Temperatures

coordinated with the vehicle designer. Thus, indirect liquid cooling systems, if used, will be incorporated into the vehicle design, and may serve several electronic systems. The electronic system designer in such an application is given a cold plate surface to which his equipment is mounted. The temperature of the cold plate will be given in the design specification; the heat removal capacity of the cold plate surface must be jointly determined by the vehicle designer and the electronic system designer. Excellent conduction coupling from the package to the cold plate is important in all indirect liquid cooling systems, but is even more critical in spacecraft applications becuase of the absence of free convection air cooling.

Internal to the electronic package, a high conductivity thermal path is required from each significantly dissipative component to the system housing. Convection is non-existent in space, and radiation heat transfer generally requires too high a component temperature to be effective. Direct conduction paths are essential. Auto-convection currents in liquid filled (direct liquid cooled) modules will not exist in the zero-gravity of space. Any benefits derived from liquid filled housings will be due to liquid thermal conduction paths only. Of course, auto-convection will be re-established in a re-entrant spacecraft when the vehicle once again enters a gravitational field. Since re-entry is often the most critical thermal phase of a space vehicle's mission, direct liquid cooling may be a justifiable consideration to insure operation in this phase. In any space application, weight penalties associated with any given cooling system must be carefully evaluated.

11. THERMAL DESIGN OF VAPORIZATION COOLED ELECTRONIC EQUIPMENT

11.1 Theory

11.1.1 Boiling heat transfer, bubble size and velocity. As indicated in Figure 5 , chapter 7, vaporization cooling (boiling) is the most effective heat removal method known. Heat removal rates can be as much as 10,000 times greater than those achievable by natural means. Thermal resistances as low as 0.01°C/watt/sq. in. are possible under optimum conditions.

The effectiveness of boiling heat transfer is due to two related effects. First, the process makes use of the latent heat of vaporization of the liquid. For example, water at atmospheric pressure has a nominal specific heat of 0.527 watt-hr./lb./°C (1.0 BTU/lb./°F). Thus, one pound of water will absorb 52.7 watt-hrs. (180 BTU) of heat energy in raising its temperature from 0 to 100°C. However, about 284 watt hours (970 BTU) of heat energy will be absorbed in changing the water from the liquid to the vapor state, with no additional change in temperature. When such large quantities of heat can be absorbed into a coolant with no temperature change, the stabilizing effect on the cooling system is obvious.

Secondly, the local heat transfer coefficient at the hot solid-to-coolant interface is increased enormously due to the local turbulence and mass tranport effects of bubble formation. In chapter 7, expressions for the heat transfer rate to a fluid in natural convection (direct liquid cooling) were derived, in the form

$$q/A = K_N \, (\Delta T)^{1.25} \qquad (11\text{-}1)$$

where K_N is a constant dependent on fluid properties, and the orientation and characteristic dimension of the submerged hot surface. A similar equation can be written for boiling heat transfer within the nucleate boiling region:

$$q/A = K_B \, (\Delta T)^n, \text{ where } K_B \text{ is a boiling constant.} \qquad (11\text{-}2)$$

The value of n for boiling heat transfer is between 3 and 4. Because of the large exponent on ΔT, major changes in heat flux density, q/A, can be accommodated with small variations in the temperature of the heat rejecting surface.

A distinction should be made between the processes normally termed local boiling and pool boiling. Local boiling refers to the state wherein the heat source surface temperature is above the liquid boiling temperature, but the bulk of the liquid is below its saturation (boiling) temperature. Under these conditions, vapor bubbles will be formed at the surface, break away, and disappear again by condensation in the bulk of cooler liquid. There will be no net generation of vapor at the surface of the liquid. Heat transfer rates on the order of 5000 watts sq./in. may be obtained under optimum conditions in this process.

Pool boiling refers to the state wherein the bulk of the liquid is at its saturation temperature. In this process, bubbles will break the surface and a net generation of vapor will occur. Pool boiling often occurs in closed vaporization cooled electronic equipment, due to the methods used in initial filling of the system. In order to avoid contamination of special coolants, atmospheric air and water vapor are generally excluded during the filling process, either by drawing a vacuum on the system while filling, or by filling the system with the coolant at a high temperature and allowing contraction of the coolant as it cools (section 10.2.6, chapter 10). In either case, the result is a system partially filled with liquid coolant, with the remainder of the volume occupied by coolant vapor at a pressure corresponding to the partial pressure of the saturated vapor at the prevailing temperature. As the system is heated or cooled, the vapor pressure will respectively increase or decrease, but will always be the saturated vapor pressure. Consequently, the liquid coolant is always at its "boiling point," as determined by the pressure variation of the vapor phase.

The "ΔT" referred to in boiling heat transfer computations refers to the temperature differential between the surface of the heat source and the bulk temperature of the liquid, the saturated liquid temperature. Because of the high heat transfer coefficients developed during boiling this temperature differential is generally small. If, in a given design the amount of heat transfer is increased, the temperature differential will increase also, as more and more bubbles are formed at each site, and more bubble formation sites become active. Eventually, a peak heat flux will be reached, at which time the surface of the heat source will be blanketed with vapor. The insulating effect of the vapor will result in a decrease in the overall heat transfer coefficient from the surface to the fluid bulk, so that the unit rate of heat transfer will decrease despite an increase in the temperature differential (see Figure 123). A still further increase in the temperature differential will result in radiation heat transfer effects becoming prominent, with a rise in unit heat transfer rate with increasing temperature differential, until the temperature differential becomes excessive (burn-out occurs). The transition region between the critical point and the establishment of film boiling is an unstable region, wherein the temperature differential is insufficient to maintain a stable film, but is in excess of that required to develop nucleate boiling. Oscillations between the nucleate boiling and film boiling regimes may occur in this region, and in extreme cases, rapid generation of vapor with explosive force may occur.

The film boiling regime is rarely used for cooling of electronic systems because of the high temperature differentials involved. The only practical example is in the use of cryogenic fuels (e.g., - liquid oxygen) as coolants, where boiling occurs at very low temperatures and film boiling is unavoidable. Applications to electronic system cooling are rare; specific cryogenic cooling techniques will be considered in chapter 12, section 12.9. For normal electronic system cooling, film boiling is not applicable.

The nucleate boiling region, up to and including the critical point, is therefore the regime most pertinent to cooling of electronic equipment. Despite the attractively high heat transfer coefficients attainable with nucleate boiling, and despite the extensive research being conducted in the field of boiling heat transfer, the process has yet to yield to either general descriptive equations, or general correlations of boiling heat transfer data. What data does exist has for the most part been derived for specific combinations of variables (heater material and surface condition, coolant, pressure, etc.), and is not applicable beyond the range of test data. Rohsenow's equation is a convenient method of correlating empirical data (Reference 58):

$$\frac{c_p\,\Delta t}{h_{f_g}\,p^{1.7}} = C\left[\frac{q/A}{\mu_l h_{f_g}}\sqrt{\frac{g_c\,\sigma}{g(\rho_l-\rho_v)}}\right]^{0.33} \qquad \begin{matrix}(11\text{-}3)\\(\text{D.E.})\end{matrix}$$

Where:

c_p =	specific heat of saturated liquid	BTU/(lb-°F) or	watt-min/(lb-°C)
q/A =	heat flux	BTU/(hr-sq ft)	watts/sq in
h_{f_g} =	latent heat of vaporization	BTU/lb	watt-min/lb
g_c =	conversion factor	4.17×10^{8} lb-ft/(lb_f-hr^2)	1.390×10^{6} lb in/(lb_f-min^2)
g =	gravitational acceleration	ft/hr^2	in/min^2
ρ_l =	saturated liquid density	lb/ft^3	lb/in^3
ρ_v =	saturated vapor density	lb/ft^3	lb/in^3
σ =	surface tension of liquid-vapor interface	lb_f/ft	lb_f/in
Pr_l =	saturated liquid Prandtl number	No units	No units
μ_l =	liquid viscosity	lb/(ft-hr)	lb/(in-min)

and C is a dimensionless constant which must be empirically determined for each system. Note that consistent units must be used, and that the value of g_c will change, dependent on the units. Constant C is a function of such items as roughness of the heating surface and wetability of the surface by a particular fluid. These types of factors cannot generally be determined analytically, which is why emperical data must be used, and why the data are applicable to only one particular combination of heater-fluid. The advantage of the correlation is that is a value of heat flux q/A, and temperature differential, ΔT, can be determined in a single test, then the performance at any other heat flux and pressure can be predicted (within the nucleate boiling region). Some typical values for the constant C are given in Table XXX.

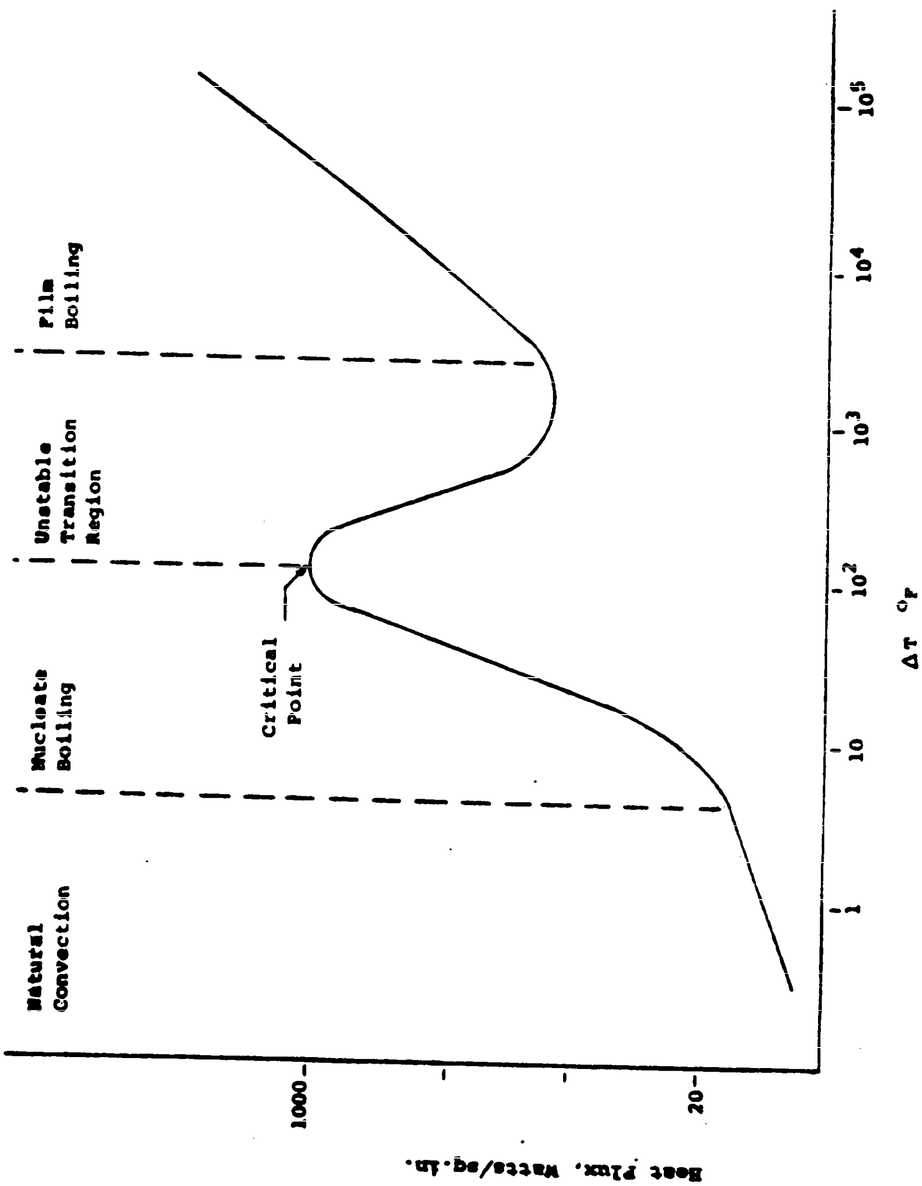


FIGURE 123. Typical Boiling Curve, Water at Atmospheric Pressure

TABLE XXX. Constant "C"

Fluid - Heater	C
Water - copper	0.013
Carbon tetrachloride - copper	0.013
35% $K_2\ CO_2$ - copper	0.0054
n Butyl alcohol - copper	0.00305
50% $K_2\ CO_3$ - copper	0.00275
Isopropyl alcohol - copper	0.00225
n Pentane - chromium	0.015
Water - platinum	0.013
Benzene - chromium	0.010
Water - brass	0.006
Ethyl alcohol - chromium	0.0027

A simple formula may be used for rough approximations of the temperature rise to be expected in nucleate boiling with a given heat flux. (Reference 59)

$$\Delta t = C'(q/A)^n \qquad (11\text{-}4)$$

Where:

q = dissipation, watts
A = dissipative area, sq. in.
Δt = temperature difference between the surface and the saturation temperature of the bulk fluid
C' and n are empirical constants

The exponent n for nucleate boiling is 0.293. The constant C' varies dependent on the surface characteristics and fluid. For most organic fluids, C' has a nominal value of 5.2, with maximum and minimum values of 11.2 and 2.4, respectively. Equation 11-4 should not be used to compute a heat flux, q/A, from a given Δt, because of the small value of the exponent. That is, relatively small changes in Δt will produce large variations in q/A.

The theoretical treatment of the boiling process begins with a study of vapor bubbles. The volume of a bubble at the moment of breakaway can be predicted theoretically, and has been verified experimentally. This breakaway volume, V_1 may be expressed by:

$$V_1 = (0.0119\beta a)^3 \qquad (11\text{-}5)$$

where β = the outer contact angle between the bubble and the heating surface at this instant, in degrees.

a = Laplace's constant = $(2\sigma/g(\rho_1-\rho_v))^{1/2}$

If a spherical shape is assumed, the bubble diameter at breakaway is:

$$D_1 = 1.034 \, (\sigma/\rho)^{1/2} \tag{11-6}$$

for β = 50°, and assuming ρ_v is negligible with respect to ρ_1.

Tests have determined that the product of bubble diameter and frequency of bubble formation is constant, at least for a given fluid. Designate this product by v_B (equal to 183.7 in./min. (280 meter/hr.) for water or carbon tetrachloride).

A dimensional analysis of the boiling process may be set up as:

$$N_u = \frac{h \, D_1}{k} = f\left(\frac{n \, A_1}{A} \cdot \frac{V_2}{V_1}\right) \tag{11-7}$$

Where:

N_u = Nusselt number

D_1 = bubble diameter at heating surfaces

n = number of bubble formation sites

A_1 = Cross section area of bubble at heating surface

A = total heating surface area

V_2 = bubble volume at the liquid surface

V_1 = bubble volume at the heater surface

f represents some function

Subsitute the values:

$A_1 = 1/4 \, \pi \, (D_1)^2$

$A_2 = 1/6 \, \pi \, (D_2)^3$

$V_1 = 1/6 \, \pi \, (D_1)^3$

Then,

$$\left(\frac{nA_1}{A} \cdot \frac{V_2}{V_1}\right) = \frac{3}{2} \, \frac{n}{A} \left(\frac{(\pi/6) D_2{}^3 f}{D_1 f}\right) \tag{11-8}$$

where f = frequency of bubble formation.

The term $(\frac{n}{A}) (\frac{\pi}{6}) D_2{}^3 f$ represents the total volume of vapor formed per unit time. This vapor generation is due to the heat extracted from the surface during vaporization, and is equal to

$\left(\frac{q}{A\rho_v h_{fg}}\right)$, so that

$$\frac{nA_1}{A} \cdot \frac{V_2}{V_1} = \frac{3}{2} \cdot \frac{q}{(A\rho_v \; h_{f_g} \; D_1 \; f)} \qquad (11\text{-}9)$$

Substituting equation 11-6 and 11-9 into equation 11-7, and recognizing that D_1 in equation 11-9 is a constant, and that all constants may be grouped, then

$$\frac{h}{k} \sqrt{\frac{\sigma}{f}} = f\left(\frac{q}{v_B \; A\rho_v \; h_{f_g}}\right) \qquad (11\text{-}10)$$

Empirical evaluation of the relationship, using water and carbon tetrachloride, results in the formula

$$\frac{h}{k} \sqrt{\frac{\sigma}{f}} = 30 \left(\frac{q}{Av_B \; \rho_v \; h_{f_g}}\right)^{0.8} \qquad \begin{matrix}(11\text{-}11)\\(\text{D.E.})\end{matrix}$$

Equation 11-11 is limited to conditions of atmospheric pressure and to those systems wherein the bubble outer contact angle at breakaway is 50°. Minor variations in pressure (as are likely to be found in electronic equipment cooling) will generally have a negligible effect. If bubble contact angles other than 50° can be determined experimentally (as by photographic analysis and measurement), then the experimentally determined values may be substituted in equation 11-5 and subsequent equations to determine a modifier for the constant of equation 11-11.

11.1.2 Peak heat flux, vapor lock. While it is not possible to analytically determine the exact relationship between heat flux and temperature differential in nucleate boiling, the determination of the peak heat flux is possible. The peak heat flux, as shown in Figure 123 is the heat flux at the critical point, where the nucleate boiling process ends and the transition region toward film boiling begins. Determination of the peak heat flux is very often of more use to the designer than an exact mathematical relationship between heat flux and temperature differential in the nucleate boiling regime, since the major design problem is one of avoiding operation in the transition or film boiling regions; that is, of assuring operation in the nucleate boiling region.

The reason the peak heat flux can be determined is that is is not dependent upon surface conditions, bubble dynamics, nucleation sites, etc., as is the boiling process equation. Rather the peak heat flux is dependent upon fluid dynamics. The boiling process is characterized by the evolution of vapor bubbles at the heating surface, and eventual breakaway of these bubbles. As a bubble leaves the heating surface, its volume is replaced by an inrush of liquid. This turbulent liquid motion as bubbles are replaced is one of the major reasons for the high heat transfer rates associated with the process. As the heat flux is increased, bubble evolution becomes more rapid, and more nucleation sites become active. There is a continually increasing relative velocity between the upward rising vapor stream and the downward flowing liquid, as heat flux is increased. Eventually, a point is reached where any further increase in this relative velocity would result in the inability of the liquid to reach the heating surface. This point is termed the hydrodynamic crisis, and determines the peak heat flux that can be attained within stable fluid dynamics operation. The peak heat flux is given for horizontal surfaces by (Reference 60):

$$\left(\frac{q}{A}\right)_{MAX} = 0.13\ \rho_v\ h_{f_g} \left(\frac{\sigma\ (\rho_l - \rho_v) g\ g_c}{\rho_v^2}\right)^{1/4} \left(\frac{\rho_l}{\rho_l + \rho_v}\right)^{1/2} \qquad \text{(11-12) (D.E.)}$$

where $(q/A)_{MAX}$ is the peak heat flux in BTU/(hr.-sq. ft.) or watts-sq. in., and all other terms are as defined for equation 11-3.

If assumptions are made that operation occurs under conditions of normal earth gravity, and that the saturated vapor density, ρ_l, then equation 11-12 may be simplified to:

$$(q/A)_{MAX} = K\ h_{f_g}\ \rho_v^{1/2}\ (\sigma\rho_l)^{1/4} \text{ watts/in.}^2 \qquad \text{(11-13) (D.E.)}$$

If h_{f_g} is expressed in BTU/lb., ρ_v and ρ_l in lb./cu. ft., and σ in lb._f/ft. (6.85×10^{-5} x dynes/cm), then k is 5.4 for horizontal surfaces and 4.2 for vertical surfaces.

Either equation indicates that the peak heat flux varies linearly with the latent heat of vaporization of the fluid, so that in this respect at least, water is the most suitable of the common fluids.

The peak heat flux is also dependent on the pressure, since an increase in pressure causes an increase in the vapor density and in the boiling point. Changes in the boiling point in turn affect the heat of vaporization and the surface tension. For every given liquid, there is a pressure which maximizes the peak heat flux. For water, the optimum pressure is about 1500 psi, and the peak heat flux is about 2500 watts/sq. in. (1,200,000 BTU/hr.-sq. ft.) (Reference 61).

In any design utilizing boiling heat transfer, generous margins of safety should be allowed. Empirical and analytical data yield average values of heat flux. Boiling, however, is a summation of many localized phenomena, and while the average value of heat flux may be below critical, local variations due to material imperfections, surface variations, or non-uniform heating may lead to significantly higher localized heat fluxes, with associated "hot spots" of high temperature rise.

11.1.3 Condensation. A vaporization cooling system utilizes the high heat transfer coefficients associated with nucleate boiling to remove heat from a heat source. This process results in the net generation of coolant vapor. Except for expendable vaporization cooling systems (section 11.4), this vapor must be re-converted into liquid and returned to the bulk of the liquid used in the boiling process. The extraction of heat from the vapor, with the attendant change of phase to liquid, is the function of a condenser.

Unlike the boiling process, equations for the mean heat transfer coefficient in condensation have been analytically derived, and experimental results indicate reasonably good correlation with theory. There are in general two forms of the condensation process, usually termed film condensation and drop condensation. Film condensation refers to a process wherein a continuous film of liquid (condensate) coats the condenser surface. As the liquid film is removed (usually by gravity), it is continually replaced by condensing vapors. Drop condensation refers to a process wherein the vapor condenses on the cool surface in drop form, and drops are removed as they grow larger due to continued condensation or coalescense with adjacent drops. Due to (a) the direct contact of the vapor with the condenser surface, (b) the larger ratio of surface to volume of drops as compared to a film, and (c) the more rapid removal of liquid in drop condensation than in film condensation, the heat transfer coefficient in drop condensation is normally several times larger than that for film condensation. Which process will occur in a given system is dependent primarily on the wetability of the surface by the coolant involved. While drop condensation is the more desirable process, it cannot be reliably maintained in practice, and is not conducive to analytical treatment due to the random nature of drop formation and removal. Film condensation will be assumed for design purposes; if drop condensation should occur, either locally or generally, the error will be conservative.

The mean coefficient of heat transfer for film condensation on a vertical surface (plane or tube) is given by:

$$h_v = 0.943\left(\frac{\rho_1\ (\rho_1-\rho_v)g\ k^3(h_{f_g} + 0.375\ c_p\Delta t)}{\mu_1 L\Delta t}\right)^{1/4} \qquad \begin{matrix}(11\text{-}14)\\ (\text{D.E.})\end{matrix}$$

Where:

h_v = mean coefficient of heat transfer for vertical surface watts/(sq. in.-°C)

ρ_1 = liquid density, lb./cu. in.

ρ_v = vapor density, lb./cu. in.

g = gravitational acceleration, in./min.^{2}

k = liquid conductivity, (watts-in.)/(in.^{2}-°C)

h_{f_g} = latent heat of condensation (evaporation) of coolant, (watt-min.)/lb.

c_p = liquid specific heat, (watt-min.)/(lb.°C)

Δt = temperature difference between condensor wall and vapor, °C

μ_1 = dynamic viscosity, lb./in.-min.

L = height of condensing surface, in.

In most design problems, $\rho_l >> \rho_v$ and $h_{fg} >> (0.375c_p \Delta t)$, so that equation 11-14 may be simplified with little error to

$$h_v = 0.943 \left(\frac{\rho_l^2 g \; k^3 h_{fg}}{\mu_l L \Delta t} \right)^{1/4} \qquad \text{(11-15) (D.E.)}$$

It is recommended that equation 11-15 be used except where the characteristics of an unusual coolant or operation at high pressures invalidate the simplifying assumptions made, in which case, equation 11-14 must be employed.

The mean film coefficient h_i for a surface inclined at angle θ to the horizontal may be obtained by

$$h_i = h_v \; (\text{Sin } \theta)^{1/4} \qquad \text{(11-16) (D.E.)}$$

where h_v is the coefficient determined by equation 11-14 or 11-15 as if the surface were vertical.

The heat transfer coefficient for film condensation on a horizontal tube may be determined by

$$h_{HT} = 0.725 \left(\frac{\rho_l^2 \; g \; h_{fg} \; k^3}{\mu_l D \Delta t} \right)^{1/4} \qquad \text{(11-17) (D.E.)}$$

where D = tube diameter, inches.

Comparison of equations 11-15 and 11-17 shows that

$$\frac{h_{HT}}{h_v} = 0.770 \left(\frac{L}{D} \right)^{1/4} \qquad \text{(11-18) (D.E.)}$$

Thus, for example, a 1/2 inch O.D. tube, one foot long, will condense 1.7 times more vapor in a horizontal position than in a vertical position. Anytime the length of a tube is greater than 2.87 times its outer diameter, the coefficient will be higher if the tube is horizontal rather than vertical.

For banks of horizontal tubes, where the condensate from one tube drops onto the tube directly below it, equation 11-17 should be modified to

$$h_{HT} = 0.725 \left(\frac{\rho_l^2 \; g \; h_{fg} \; k^3}{\mu_l n D \Delta t} \right)^{1/4} \qquad \text{(11-19) (D.E.)}$$

where n = the number of horizontal tube layers.

These general equations (11-14 through 11-19) are valid for film condensation of vapors at rest, which is usually valid for electronic system cooling. Coefficients may be increased by adding de-wetting agents to promote drop condensation, or by providing a high vapor velocity to remove the liquid film more rapidly. Quantitative analysis of these effects is complex and beyond the scope of this handbook.

While condensation heat transfer coefficients may be determined analytically with reasonable accuracy, empirical tests should always be conducted in the design process to verify computed data, especially for combinations of coolant and condensor surface which are new or which have not previously been evaluated.

11.1.4 Ultrasonic agitation of boiling liquids. There are several methods available to increase the boiling heat transfer coefficient and the peak heat flux, but application data is extremely limited.

One such technique is the utilization of an ultrasonic transducer, operating usually in the 25 to 40 KHz range, to insonify the boiling coolant and aid the breakaway of vapor bubbles from the heat transfer surface. The process appears to delay the onset of boiling; that is, boiling will start at a lower temperature difference without ultrasonic agitation than with it. The peak heat flux is also increased with ultrasonic agitation, but the relationship is presently undefined. The technique is relatively unexplored and has found little use to date. At least one practical application is known, which permitted the operation of a ferrite core computer memory at unit heat concentration levels of up to 50 watts/cu. in. Because of the limited knowledge of the process, this technique should be used only in critical applications, and extensive verification tests are imperative to demonstrate design adequacy. Additional information on this technique is given in chapter 10, sections 10.8.3 and 10.8.4.

Somewhat similar effects have been obtained in limited laboratory tests by application of high electrostatic fields.

Additives to the coolant may also be used to increase the peak heat flux. For example, the addition of Hyamine 1622 to water in concentration as low as 1% has increased the peak heat flux by 40% in laboratory tests. These additives seem to work primarily be increasing the coolant surface tension.

11.2 Direct vaporization cooling.

11.2.1 Immersion of electronic parts. In a direct vaporization cooling system, the heat dissipating or temperature sensitive electronic parts are immersed in the liquid phase coolant. In this respect, it is similar to the direct liquid cooling process described in chapter 10, section 10.2. Similar considerations of the effect of immersion of electronic parts apply, but in boiling heat transfer, several additional factors are encountered. (1) During boiling, the coolant at any given point may exist alternately in the liquid and vapor phases, as vapor bubbles form, grow in size, and are released. (2) The temperatures involved in boiling heat transfer are generally higher than those encountered in normal liquid cooling. The operating temperature is fixed within a narrow range by the boiling point of the fluid at the prevailing pressure. (3) The boiling process by its nature involves extreme agitation and local turbulence of the coolant. (4) Parts must be placed carefully to insure that there is sufficient room for bubble formation and escape, and that no vapor traps exist.

With these differences in mind, an evaluation of coolant-electronic system compatability may be made similar to that for direct liquid cooling, including thermal, electrical, chemical, and mechanical aspects. Boiling heat transfer differs thermally from direct liquid cooling in that it accommodates a wide variation of heat flux over a narrow temperature range. The operating temperature is therefore normally determined by the properties of the coolant. Consequently, the coolant must be selected such that its boiling temperature at the system pressure is suitable for reliable operation of the electronic circuitry.

Electrical compatability of the circuit-coolant is concerned primarily with coolant resistivity and with coolant dissipation factor for high frequency circuits and dielectric strength for high voltage circuits. Evaluation of these effects is complicated by the two-phase nature of the coolant, i.e., vapor and liquid. If the intermittent presence of vapor should cause circuit malfunction, it may be necessary to utilize an indirect vaporization cooling scheme.

Chemical compatability of the coolant and circuit may be affected either by the two-phase nature of the coolant, or by the higher operating temperature normally involved. Vapors are often more active chemically than their liquid counterpart. Compatability of both liquid and vapor phase must be checked with component cases, circuit board materials, rosin, potting compounds, solders, and any other material within the module.

Mechanical support of components must be capable of withstanding the forces generated by the turbulent action of the boiling fluid. These forces can be appreciable, and packaging techniques normal for natural air convection or direct liquid cooling may be inadequate structurally for a boiling heat transfer process.

11.2.2 Forced circulation of vapors and liquids. Present data indicate that once the pool boiling regime has been established in a vaporization cooled system, the forced convection cooling effect obtained by mechanical circulation of the coolant has a negligible effect on the surface temperatures of the heat sources. The agitation caused by the bubbles is much more effective than the turbulence caused by forced convection alone. Forced convection of the fluid will delay the onset of boiling at a given heat flux. Higher fluid velocities require a higher surface temperature to initiate the boiling process because of the forced convection cooling of the surface. However, once pool boiling begins, the relationship between the heat flux and the excess temperature is independent of fluid velocity. Consequently, mechanical agitation or circulation of the liquid coolant is not effective in increasing the heat transfer coefficient in a pool boiling process.

In a condenser, the condensate layer itself presents a major thermal resistance. Any effect tending to reduce film thickness can result in substantially higher surface conductances than those predicted by equations 11-14 through 11-19. High vapor velocities by use of fans or blowers is one such method. Analytical evaluation of high vapor velocity effects is impractical, and model tests are necessary in establishing design parameters. Care must be taken that the vapor velocity is such as to reduce the film thickness. An upward velocity on a plane vertical wall adds a retarding force to the viscous shear, and causes the film thickness to increase. Similarly, a horizontal velocity may cause a reduction in film thickness at the vapor inlet end, but an increase in film thickness at the opposite end, with no net effect on overall conductance.

11.2.3 Pressure effects-temperature control. An increase of pressure in a boiling liquid system causes an increase in the vapor density, and an increase of the boiling point temperature. Changes in the boiling point temperature, in turn, result in changes in the heat of vaporization and in the surface tension. All of these factors affect the peak heat flux, as given in equation 11-12 or 11-13. The net result is that the peak heat flux increases at first, with increasing system pressure, then passes through a maximum, and finally decreases to zero at the critical pressure (since $h_{fg} = 0$ at $p = p_c$). There is therefore an optimum pressure for each liquid coolant, at which the peak heat flux is a maximum. This optimum pressure generally occurs at approximately 1/3 the critical pressure. For water, the optimum pressure is about 1500 psi, with a corresponding peak heat flux of about 2440 watts/sq. in. The saturation temperature at this pressure is over 300°C, which is too high for most electronic cooling systems. Because of the high pressures and temperatures involved, optimization of peak heat flux by increasing pressure is not generally applicable to the cooling of electronic systems; however, even slight increases in system pressure will yield appreciable increases in peak heat flux. The effect of moderate increases in system pressure may be evaluated by multiplying the heat transfer coefficient computed at standard atmospheric pressure by the factor $(p/p_a)^{0.4}$, where p and p_a are the prevailing and atmospheric pressures, respectively.

The relationship of system pressure to operating temperature conditions is unique for pool boiling heat transfer. Since the fluid bulk is at its saturation temperature during the boiling process, its temperature is precisely defined by the vapor pressure. A pressure relief valve may be set to maintain the bulk fluid temperature at any point within the range of the fluid used. For example, using a water system vented to standard atmospheric conditions (14.7 psia), the boiling water will be at 100°C (121°F). If the system pressure is allowed to build up, the boiling water temperature will be 227.9°F at 20 psia, 250.3°F at 30 psia, etc. (Saturated pressure-temperature values for water are readily available from steam tables, such as Reference 62. Pressure-temperature saturation curves for commercial coolants are generally available from the manufacturer). A pressure relief valve will, of course, allow escape of some vapor, which must be replenished with makeup coolant, or condensed separately and returned to the system by a pump.

Without a pressure relief system, the capacity of the condenser must be matched closely to the boiling process. If the condenser is not capable of liquifying the vapor evolved, the system pressure will increase, with a corresponding increase of bulk fluid temperature. An oversized condenser results in poor system economy, both in initial cost and in operation. The system operating temperature can theoretically be controlled by regulating the flow of coolant through the condenser to maintain a specific vapor pressure. Such control systems, however, tend to be complex and the thermal time lag involved in the condensor operation makes the design critical.

In all systems, a safety pressure relief valve should be provided to allow release of excessively high pressure should any malfunctions occur.

11.2.4 Condensers and evaporators. A condenser may take a variety of forms, some of which are indicated in Figure 124. In its simplest form, Figure 124a, the housing itself forms the condenser. The external housing surfaces are cooled by ambient air by means of free or forced convection. These cooled housing surfaces form a relatively low temperature heat sink upon which the vapor condenses. If the exterior housing surface is not adequate to maintain the housing below the vapor saturation temperature, one or more of the outer surfaces may be extended in area by fins, as in Figure 124b. Still larger heat flux values may be accommodated by using a cold plate for a condenser surface, Figure 124c. Finally, an external heat exchanger condenser may be provided as in Figure 124d, and may be sized to handle any required heat load. External condenser systems may be provided with vapor flow augmenting devices (fans), and with flow and/or temperature regulating controls.

If a liquid is used as the ultimate sink in a condenser, it will generally be pumped through tubes. Condens r tubes may be arranged horizontally or vertically, with headers at both ends for distribution, or may consist of coiled tubing. Horizontal tubes are usually preferable from a heat transfer standpoint (section 11.1.3). Straight run tubes provide for simple, relatively inexpensive internal cleaning, with removable headers.

If air, or some other gas, is used as the ultimate sink, the condenser usually is formed as vertical plates. Spaces between plates are alternately used as condensing surfaces and air passages. The air side of the condenser plates may be finned to increase the air forced convection surface area.

Computations of condenser fluid temperature rise, pressure drop, and required heat transfer rate and flow rate may be made utilizing the principles of chapter 9, for gas flow or chapter 10 for liquid flow.

The "evaporator" of a direct vaporization cooling system is the actual heat source, which may be a directly immersed component, or an immersed sealed module. Evaporators should be arranged to promote the free flow of liquid and vapor bubbles, and to avoid vapor traps.

11.2.5 Spray systems. One of the drawbacks of immersion in liquid cooling systems, whether they be direct liquid cooling or direct vaporization cooling, is the weight of the fluid required to fill (or partially fill) the housing. This is a particularly critical factor in airborne and spacecraft systems. One method of reducing the fluid weight is to utilize a spray system. The spray nozzles are positioned so as to maintain a continuous film of coolant over the surfaces of the heat sources. If a boiling process is used, there will be a net generation of coolant vapor, which must either be expanded or condensed and returned to the system via a pump or gravity. Similarly a recirculating pump must be provided to return the pump coolant to the system.

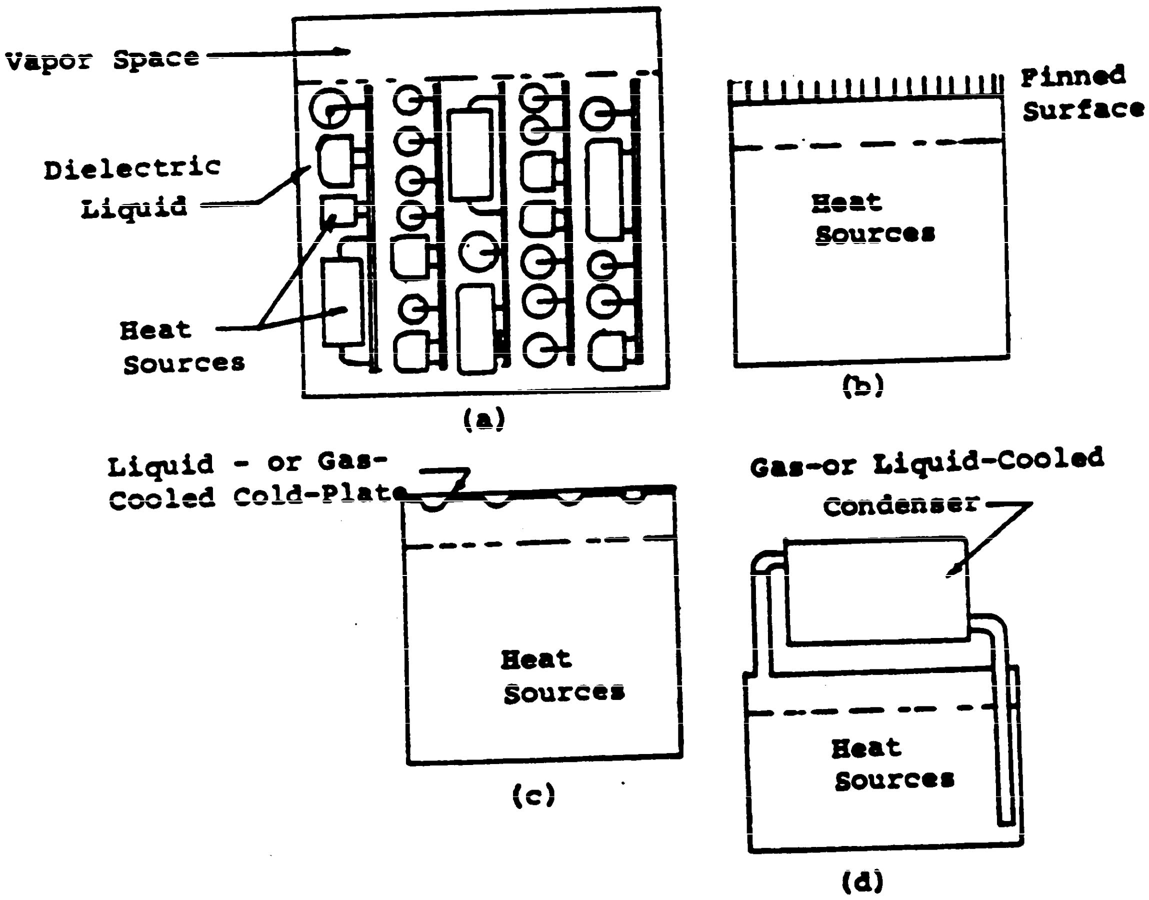


FIGURE 124. Condenser System Schematic

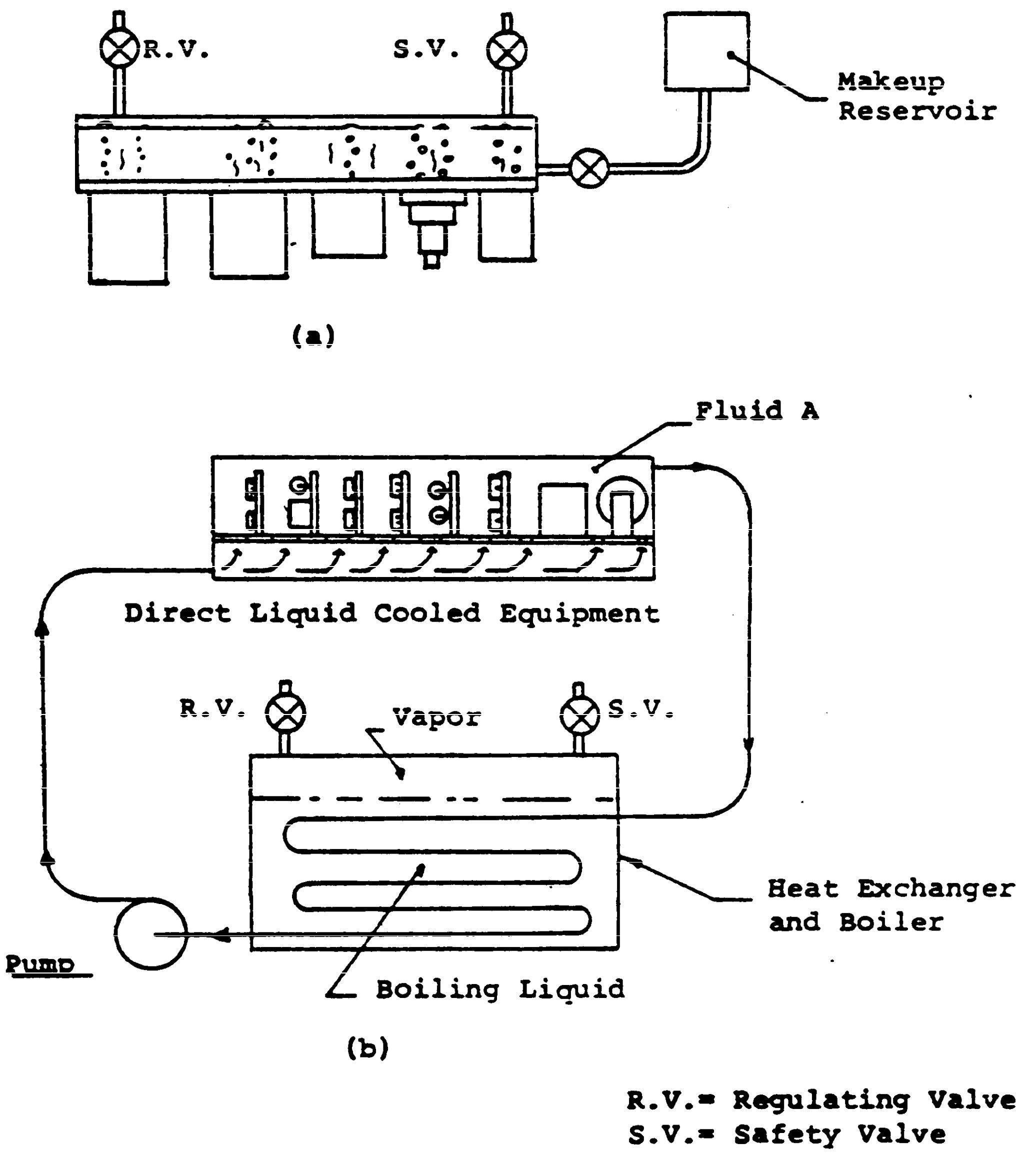


FIGURE 125. Indirect Vaporization Cooling System Schematics

These pumps must be capable of developing a high enough pressure to force the coolant through the spray nozzles. The savings in liquid weight by using a spray system must be weighed against the added weight and design complexity of the pumps and piping. Spray systems are affected by attitude changes with respect both to gravity return to the sump and also to the proper development of the spray pattern. Because the spray nozzles incorporate small diameter orifices, spray systems are also susceptible to malfunction due to dirt and impurities even in closed systems. Proper filters and maintenance are particularly necessary for these systems.

11.2.6 Capillary "wicking" of parts. Another method of reducing the weight and quantity of fluid required in a direct vaporization cooling system is part "wicking." The parts (or the heat producing surfaces of the parts) are covered with a wick of woven material; the other end or edges of the wick material is immersed in a sump or reservoir containing the coolant. The coolant flows from the sump through the wick by capillary pumping on to the surfaces of the parts where boiling heat transfer occurs. In effect, the parts become the evaporator of a large heat pipe formed by the equipment package.

Parts wicking is an excellent cooling technique and several advanced avionic systems are utilizing this method. Chapter 12 of this handbook discusses heat pipes, and the details of wicks, capillary pumping, etc. are covered in that chapter. It is important to note that the height or lift obtained by capillary pumping is strongly influenced by gravitational forces and care must be exercised in the application of this technique to equipments subject to acceleration and manuevering forces. Also the open wick system may not have any recovery of the liquid.

11.3 Indirect vaporization cooling system.

11.3.1 General. Indirect vaporization cooling systems are analogous to indirect liquid cooling systems. In indirect vaporization, the heat generating components are not directly immersed in the boiling liquid, but an intermediate heat path is provided. This intermediate heat path may be provided by conduction paths, as in Figure 125a, or by a direct liquid cooling loop (non-boiling), as in Figure 125b. Either system utilizes the high heat transfer coefficients (low resistance paths) associated with boiling heat transfer. The "cold plate" system of Figure 125a might be used in applications where direct immersion of components in the liquid would be unacceptable because of effects on electrical performance, as in high frequency circuits, or because of the requirement for rapid accessability for maintenance purposes or module interchangeability. The cascaded, or series, system of Figure 125b is frequently used where the ultimate boiling liquid coolant is incompatible with the electronics, but where low thermal resistances are imperative. For example, water is an excellent boiling heat transfer fluid because of its high heat of vaporization and specific heat. Its electrical properties, however, prevent it from being used in direct contact with electronic circuitry. The intermediate loop of fluid A (Figure 125b) provides a low thermal resistance path from the heat sources to the heat exchanger, where fluid A rejects its heat to the boiling water. The vapor evolved in the water boiling may either be vented as shown (section 11.4 - Expendable vaporization cooled systems) or condensed in another heat exchanger and recycled.

In any process, vaporization cooling (either direct or indirect) should be recognized as one possible method of heat transfer, providing very low thermal resistances which may be combined with any or all other methods (natural, forced convection, liquid cooling, etc.), to provide a total system design.

11.3.2 System evaporators, condensers, and boilers. The following general definitions apply as referenced to indirect vaporization cooling of electronic systems:

Evaporator: An evaporator is that assembly consisting of heat sources (components), boiling fluid, intermediate transport loops (if any), and the associated housing and controls. The vapor generated in an evaporator is normally condensed and the liquid recycled.

Condenser: The condenser is a heat exchanger which extracts the heat of vaporization from the vapor evolved in the boiling process, thereby liquifying it, and rejects this heat to another fluid (either gas or liquid). Condensors do not normally appreciably subcool the return liquid, although such a design is possible.

Boiler: A boiler is defined as a pool of boiling liquid whose vapor is expended, rather than being condensed and recycled. Examples of boilers are the relatively common water boiler systems used in aircraft and the use of cryogenic liquid fuels as heat sinks (these fuels are eventually expended after doing work in a propulsion or power generating system).

11.3.3 Pumps and compressors. The comments relative to pumps (chapter 10, section 10.6.3) and fans (chapter 9, section 9.2) are valid for vaporization system components, with a few additional precautions.

The selection of a pump for the vaporization fluid must be matched to the liquid generating capacity of the system to insure that the pump does not run dry. Intermittent pump operation may be used, with the pump switched on and off in response to a liquid level sensor in the condenser or in the return leg. Special pumps designed to be capable of running dry are available, but at increased cost and usually with a shorter life.

Since vaporization cooling systems operate at or near the liquid/vapor saturation point, slight local reduction in pressure can cause vapor formation. Cavitation at the inlet and along pump working surfaces is a strong possibility because of local pressure variations. The designer should pay particular attention in pump design and/or selection for vaporization cooled systems to prevent cavitation.

Pumps for the circulation of condensor fluid, if required, may be selected in accordance with chapter 10, section 10.6.3.

Compressors or fans are sometimes used for forced circulation of the vapor phase of vaporization cooled systems. The general design/selection criteria of chapter 9, section 9.2 are applicable for vaporization cooling systems, except that the designer should recognize that the vapor will be "wet," that is in a saturated condition, and droplets of liquid are likely to be present in the vapor stream. Any computations of flow rate or pressure drop must use the properties of the vapor being transported. Fans to circulate air through air condensers may be designed in accordance with information given in chapter 9.

In regulated vaporization cooling systems, designed to operate at a specific internal pressure, seals for pumps, fans, or their drives must be reliable at the system design operating temperature and pressure. Completely sealed units are generally preferable for vaporization cooling systems.

11.3.4 Refrigeration. In refrigeration, mechanical work is performed on a fluid system, producing a thermodynamic cycle. During part of this cycle the fluid is above the temperature of an ultimate sink, and thus rejects heat. During another portion of the cycle, the fluid is below the temperature of the heat sources, and accepts heat from them. By virtue of the mechanical work done on the fulid, the system can therefore reject heat to an ultimate sink at a higher temperature than that at which it accepts heat from the heat sources. Thus, a refrigeration system exhibits a negative thermal resistance.

A refrigeration cycle is indicated schematically in Figure 126. Beginning at point (1), the fluid is usually in the gaseous state. (Some systems may utilize a "wet compression" stage where the fluid at the compressor inlet is a mixture of vapor and liquid droplets, or mist). The compressor raises the pressure and temperature of the fluid. The compression must be sufficient to raise the temperature of the vapor above that of the ultimate sink, T_S. The condenser is a heat exchanger which cools the vapor by use of the ultimate sink coolant. This coolant may be air, water, fuel, radiation panels, or any other ultimate sink. Cooling of the vapor extracts any superheat, plus the latent heat of condensation, plus some slight subcooling, resulting in a subcooled liquid at point (3). The expansion valve is merely a device to separate regions of high and low pressure. Expansion through the valve results in a drop in temperature to some level below T_H, that of the heat sources. The evaporator is a heat exchanger which accepts heat from the heat sources, and raises the temperature of the fluid to point (1) where the cycle originated.

A variety of terms are used to rate refrigeration units. One commonly used term is that of "tons" of refrigeration. One ton is the rate of heat absorbed in melting 2000 pounds of ice at 32°F in 24 hours, and is equal to 12,000 BTU/hr., or 3516 watts. (Coefficient Of Performance) (COP) is a measure of the efficiency of the refrigeration process, and is defined as (Refrigeration)/(Work), where both terms are expressed in the same units.

Refrigeration systems must be considered whenever the temperature of the heat sources must be maintained below that of the ultimate sink, or when the temperature differential is so small as to make other methods impractical. The only alternatives are some of the special cooling techniques discussed in chapter 12, such as thermoelectric coolers and vortex tubes, which also can provide negative thermal resistances. Refrigeration is usually preferable for all but the smallest heat dissipations and/or concentrations.

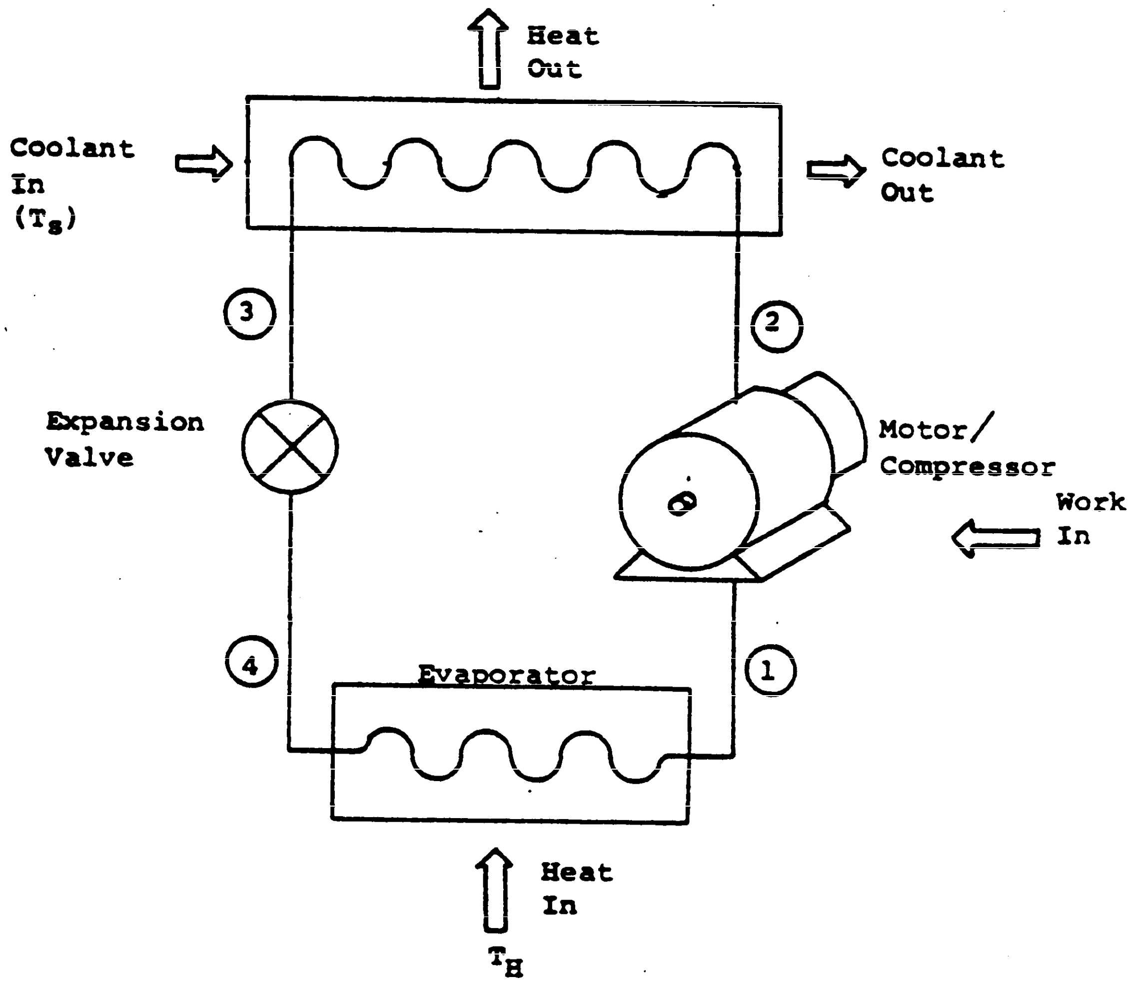


FIGURE 126. Typical Refrigeration System Schematic

11.3.5 Aircraft vapor cycle cooling systems. Vapor cooling systems in aircraft use a common form of refrigeration cycle. In this type of system, heat is transferred from the equipment to a vaporization fluid by any method compatible with the total heat dissipation and allowable temperature rise. Examples include direct conduction (cold plate), forced air cooling or direct liquid cooling. The system shown schematically in Figure 127 depicts a direct liquid cooling system with a liquid-to-liquid heat exchanger. The vaporization coolant (assume freon or equivalent) is boiled within the heat exchanger. The latent heat of vaporization of the coolant provides the cooling for the direct liquid loop coolant. The compressor circulates the vapor and raises its pressure and temperature. The condenser uses a ram air heat exchanger to extract heat from the high temperature vapor and convert the vapor to liquid. The liquid is expanded through an expansion valve. As a result of the pressure drop through the expansion valve, some of the liquid vaporizes. In the process, the heat of vaporization is extracted from the liquid, further reducing its temperature. The low temperature liquid and relatively small amount of vapor then enters the evaporator to repeat the cycle.

Two optional features are included in Figure 127: 1) An intercooler (heat exchanger) extracts heat from the liquid vaporization coolant leaving the condenser before it enters the expansion valve, resulting in a lower temperature. This heat is transferred to the coolant leaving the evaporator before it enters the compressor. This not only further optimizes the heat transfer system, but also vaporizes any liquid droplets in the vapor leaving the evaporator, assuring a "dry" vapor entering the compressor. 2) For high speed fluight, with high ram air temperature, an expendable water/alcohol mixture is injected into the ram air before it passes through the condenser, to reduce the ram air temperature by evaporation of the liquid.

Selection of aircraft cooling systems is a complex process, wherein weight penalties must be assigned to alternative systems for heat transfer apparatus and equivalent fuel weight for power for pumps, compressor, fans, etc. The vapor cycle system must compete with a variety of other systems, such as the simple air cycle, bootstrap air cycle, and regenerative systems. Details of these systems are given in chapter 9.

11.4 Expendable Vaporization Cooling Systems. A system which discharges the vapor evolved during boiling, rather than liquifying it in a condenser and recycling it, is termed an expendable vaporization cooling system. A decision to use such a system must include a tradeoff study of the weight, volume, cost, and effect on reliability of a condenser and any associated controls, pumps, and, piping, against that of the expended coolant. Several factors might influence a designer to the choice of such a system.

If the supply of coolant is abundant and relatively inexpensive, expendable vaporization may be the most economical choice. Thus, fixed ground stations, shipboard, and to a lesser extent, mobile ground station, might use water as an expendable coolant provided 100°C ultimate heat sink temperature is acceptable. Good maintenance is especially critical on such systems to prevent fouling of the heat transfer surfaces by impurities and mineral and salt deposits left by the boiling water. System design should include a generous safety factor to allow for fouling between scheduled cleaning operations.

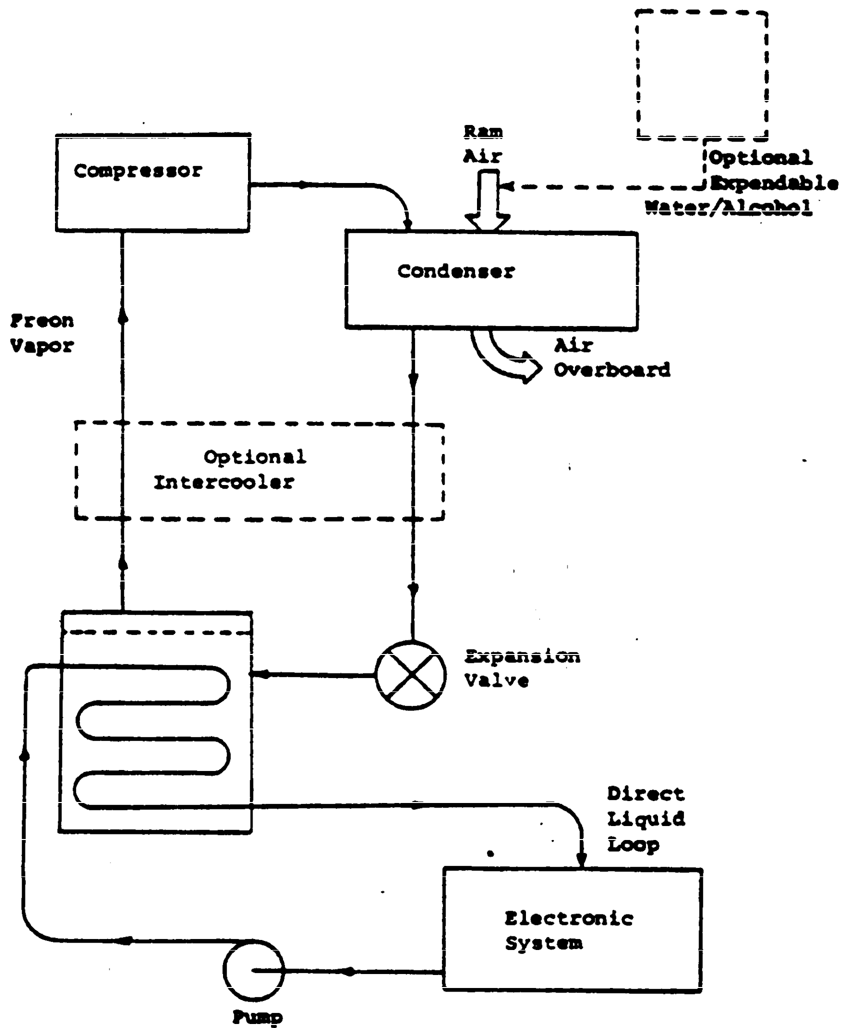


FIGURE 127. Aircraft Vapor Cycle Cooling System Schematic

If the heat dissipation exceeds the normal cooling system capacity for a short time, an expendable vaporization cooling system may be used as a "topping" device to function only during the thermal peaks. This condition might occur as a result of a short term dissipation rate operation of the electronic system, or as a result of a short term change in the thermal environment. For example, an attack aircraft usually operates in a relatively constant thermal environment for well over 90% of its mission, but during the short period of supersonic operation during the strike, aircraft skin temperature will rise dramatically, and ram air cooling becomes inoperative due to the high air stagnation temperature rise. Thus, cooling methods which are adequate for most of the mission become inoperative during the strike, when system operability becomes critical. To enlarge the cooling system to handle this peak is inefficient at best, and often impossible. A water boiler is an effective and common method of providing a short term ultimate heat sink. Water is vaporized in the boiler, and the vapor is expended. Because of the high heat of vaporization of water, (970 BTU/lb., or 284 watt-hrs./lb. at 100°C 1 atmospheric pressure) large amounts of heat may be disposed of by this method. The water boiler may act directly on the electronic system but direct or indirect vaporization cooling, or it may be used in an air cycle system to cool the ram air to acceptable temperatures for use in cooling the electronics.

In another application, the coolant is expendable only in the sense that it is not condensed and recycled. Where cryogenic fuels are used for propulsion or power generation (usually on aircraft), these fuels must be vaporized and superheated before use in the reaction chambers. Some of the heat of vaporization of the fuel may be supplied from electonic equipment, providing an ultimate heat sink for the electronic system. Since the boiling point of liquified cryogenic gases is far below the normal operating range of most electronics, the system involves indirect vaporization cooling, with an intermediate liquid system to transport the heat from the electronics system to the vaporizing fuel.

The quantity of coolant expended may be calculated by:

$$m = 3.413 \frac{q}{h_{f_g}} \qquad \text{(11-20) (D.E.)}$$

Where:

m = coolant expenditure rate, lbs./hr.
q = heat transfer rate, watts
h_{f_g} = coolant heat of vaporization, BTU/lb.

The temperature of the bulk of the fluid (assuming pool boiling) will be the saturation temperature of the liquid at whatever pressure is maintained.

11.5 Vaporization Cooling Systems, Design Considerations

11.5.1 Piping, controls, and valves. A vaporization cooling system is by definition two-phased; the coolant exists as a liquid, a vapor, and a wet mixture in various parts of the system. The fluid handling portion of the design must consider all states of the coolant, and that the state at any given point may change with time.

In the simple system illustrated schematically in Figure 128a and b, the housing performs the function of both evaporator and condenser, and no external piping or ducting is required for the vaporization fluid. (External ducting may be needed if forced air cooling is used on the exterior housing surface, but this is a fluid handling system for the cooling air, not the vaporization fluid. See chapter 9 for forced air system design.)

In Figure 128c and d, a condenser is built into the housing, and an external fluid (liquid or gas) is used as an ultimate sink. Again, no piping or ducting is used for the vaporization fluid, since it is cycled continuously within the housing. The piping or ducting for the condensor fluid is covered in chapter 9 for gas or 10for liquid.

If an external condenser is utilized, the vapor must be ducted from the evaporator to the condenser. The methods presented in chapter 9 for sizing ducts and determining pressure drops and duct heat losses may be used, provided the physical parameters of the coolant vapor are used. Those formulas of chapter 9 which have been simplified by including relatively constant air properties, are not applicable. The vapor flow rate (lbs./min.) is equal to the total heat dissipation (watts) divided by the heat of vaporization of the coolant (watt hrs./lb.). This will result in a conservative number, since there will be a heat loss from the evaporator (housing) to its thermal environment. If this heat loss is significant, it may be calculated or estimated, and the result subtracted from the total heat dissipation to determine vapor flow rate. This will avoid overly conservative design, and result in a better optimization of duct size and weight.

Ductwork should be designed to handle both vapor and wet vapor-liquid mixtures. Pockets which might act as liquid traps should be avoided, and the ductwork should be arranged so as to allow any liquid accumulation to run to the condenser, or back to the evaporator, as applicable. Liquid can form in the ductwork for a number of reasons. If the vapor temperature is above that of the surrounding environment, there will be a heat loss from the ducts, and condensation of the vapor may occur on the inner walls, particularly if the ducts are large or long. In extreme cases, duct insulation may be required. In regulated systems (Figure 128e, g, and h), a throttling process occurs across the regulating valve (R.V.). This may result in a wet mixture downstream of the valve. Finally, at some time between startup of the electronic system and attainment of steady state conditions with boiling, the system will operate with condensation within the ducts.

In non-expendable regulated systems (Figure 128e, g, and h), a regulating valve (R.V.) allows the flow of vapor sufficient to maintain the pressure (and therefore, the saturated liquid temperature) at a predetermined value within the evaporator electronic housing. This bled vapor is then liquified in an external condenser and recycled to the bulk liquid. Since there is a pressure drop across the regulating valve, and flow pressure drops in the condenser and associated lines, a pump must be used to return the liquid to the evaporator. The pump/piping system is sized so that the heat developed by the pump at the established flow rate equals

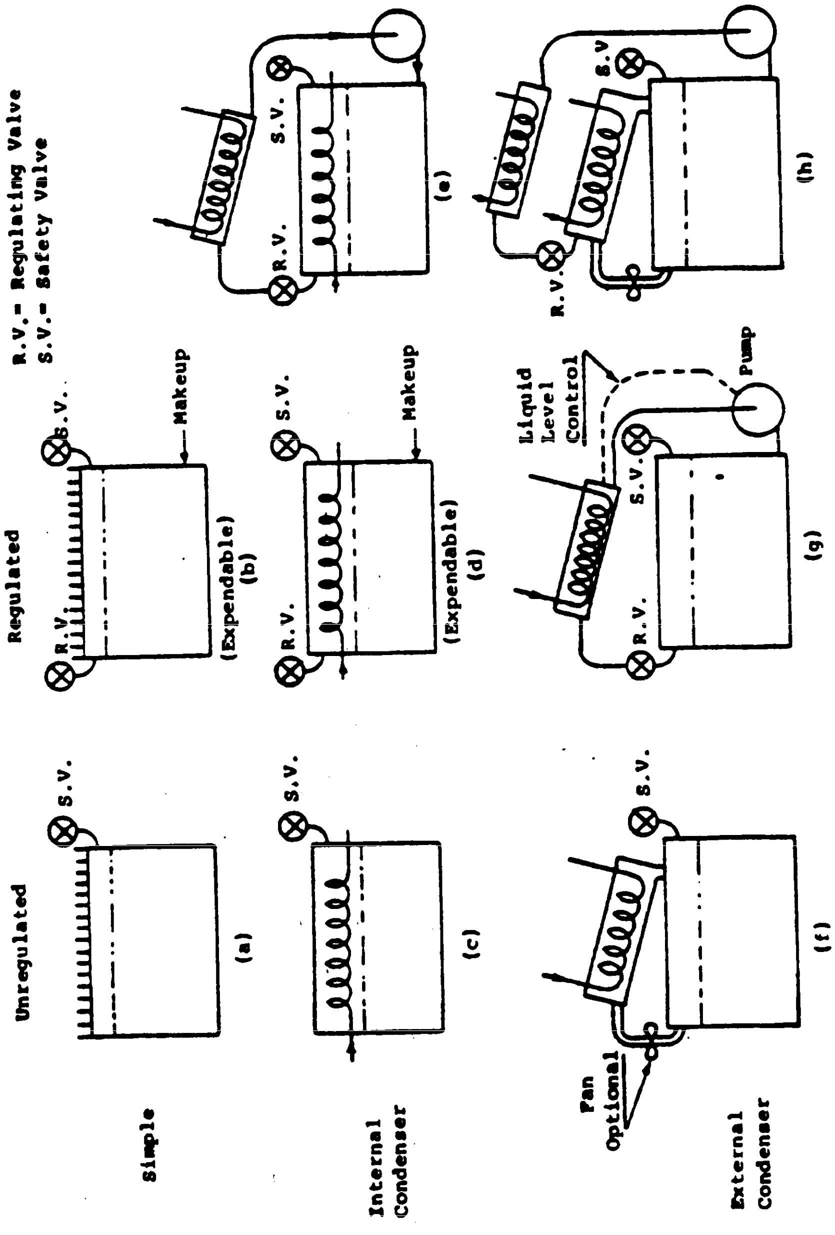


FIGURE 128. Vaporization System Schematics

or exceeds the friction head developed at that flow rate in the piping, valves, condenser, and electronics. Normally, however, the pressure drop across the regulating valve is more significant than the frictional load in the remainder of the system. Often the pump operation is intermittent. A pump with excess capacity is selected, and is energized by a liquid level sensor in the condenser or the return piping. When the pump has delivered enough liquid to lower the liquid level to a preset value, the pump is de-energized.

Welded, brazed or soldered connections are preferred in pipe lines, except where maintenance requires screw type connectors. Pipe and duct material must be selected to avoid corrosion effects, either directly from the coolant, or as the result of galvanic corrosion caused by the use of different materials in contact. Corrosion resistant materials, such as stainless steel, Monel or equivalent, and treated aluminum are recommended for valves and piping. Corrosion effects are discussed in more detail in section 11.5.2 following.

If in-line valves are required, gate valves are preferred to globe valves because of their inherently lower pressure drop. The number of in-line valves should be kept to a minimum; their normal purpose is to permit servicing of an individual chassis without the need to drain the entire system. Drain valves should be provided at the lowest point of each leg of the system. Valves or pet-cocks should be installed at the uppermost point of each leg to facilitate drainage and to vent trapped air during system filling. Inlet and outlet connections of the quick disconnect, self-sealing type facilitate the removal of an individual unit without loss of coolant. Thus, draining of the unit and venting and refilling after re-connection are unnecessary during normal maintenance and repair procedures.

Filters are used to extract particulate matter from coolants, to protect pump bearings and valve seats, and to prevent accumulation of flow restricting matter in bends, elbows, and fine passages. All systems using expendable coolants with makeup provisions should contain filters. Even sealed recirculating coolant systems are normally provided with filters since all common coolants are subject to some form of sludge contamination over very long periods of time. Filters are generally of the replaceable cartridge type in liquid systems, and should be located for ease of maintenance.

Unregulated vaporization cooling systems, such as those shown in Figure 128a, c, and f, have no inherent controls. The operating conditions are determined by a proper initial design balance between the evaporator and the condenser, based on the worst case conditions of anticipated heat dissipation and thermal environment. Consider, for example, the simple unregulated system of Figure 128a, initially turned off and in thermal equilibrium with the environment. When the electronic system is energized, heat will be transferred into the bulk liquid coolant, initially by direct immersion liquid cooling. Assuming that the heat dissipation exceeds the housing heat rejection rate, the system will rise in temperature, utilizing the thermal capacity of the electronics, coolant, and housing to absorb the excess dissipation. When the bulk liquid reaches the boiling point temperature at the internal housing pressure, the vapor evolved in the boiling process will condense on the upper cover and exposed side wall surfaces of the housing, establishing a highly effective heat transfer path. Fluid bulk temperature and internal pressure will continue to rise unitl thermal equilibrium is attained. This will occur when the difference between the saturated vapor temperature and the condenser (upper housing) temperature is sufficient to create a heat transfer rate in the condenser equal to the total heat dissipation. If the heat dissipation rate should subsequently increase, there will be a more rapid evolution of vapor. This will result in an increased pressure, and the bulk liquid/vapor saturation temperature will rise to that corresponding to the increased pressure. The increase in vapor temperature will provide an increased temperature differential for the condenser, and its heat transfer rate will increase to the new dissipation level. Thus, the bulk liquid/vapor saturation temperature and the internal pressure will assume whatever value is required to maintain thermal equilibrium under steady state conditions, and they will therefore vary with the total heat dissipation. Operating temperatures at a given heat dissipation are set by the initial heat balance. In Figures 128c and f, the external coolant flow rate through the condenser may be vaired in response to bulk liquid temperature variations, so as to enable some degree of control of the system operating temperature. Such control systems, however, have a slow response due to the relatively large thermal time lag of the condenser system. This coupled with the added complexity of the sensor and external coolant flow control valve, usually decides against such a control system.

Regulated systems employ a regulating valve (R.V. in Figure 128) which bleeds off sufficient vapor to maintain the internal pressure at a preset value. Since the pressure and temperature are precisely related for a saturated boiling liquid, the bulk fluid temperature is therefore established. The boiling process equations then define the surface temperature rise over the bulk fluid saturation temperature.

In Figures 128b and d, the vapor bled off by the regulating valve is expended. This system is frequently used for expendable vaporization cooled systems (section 11.4), but for direct vaporization cooling of electronics, makeup coolant must be supplied.

In Figures 128e, g, and h, the bled off vapor is liquified in an external condenser and returned to the system, but at the expense of adding ductwork, condenser, piping, pumps, and controls. In Figure 128g, all bled vapor is cycled throughout the condenser, so that the ductwork, condenser, piping, and pump must be sized to accommodate the full fluid flow. In Figures 128c and h, the main condenser (either internal or external) handles most of the heat dissipation, while the regulating valve, secondary condenser, and pump handle only a fraction of the heat dissipation, sufficient for control purposes. These components may therefore be sized smaller, at the cost of providing two condensers.

Each system schematically depicted in Figure 128 includes a safety valve (S.V.). Such a valve should always be included in a boiling heat transfer system to prevent rupture or explosion in event of a malfunction. In addition to the relief of internal pressure, these valves may also act to de-energize the electronics system, or to give an audible or visual alarm, or both.

Other controls not shown in Figure 128 include (1) liquid level sensors in the evaporator, which may act to introduce makeup coolant in the event of fluid loss or to activate an alarm, (2) filter pressure drop alarm systems, to warn of clogged liquid line filters, and (3) external coolant flow control devices.

Any system utilizing makeup coolant should provide filters in the intake lines, and any recycling regulated system should include a liquid line/filter to protect pump bearings and valves.

11.5.2 Corrosion effects. Corrosion effects as applied to liquid cooling systems were discussed at length in chapter 10, section 10.6.6. The same general design rules apply to vaporization cooling sytems, with the additional factor that the coolant exists in both liquid and vapor phase. Corrosion effects can be eliminated or minimized by adhering to these design guides:

1. Use corrosion-resistant materials or protective coatings to prevent atmospheric corrosion.

2. Where necessary, insulate exterior surfaces to prevent condensation.

3. Do not use dissimilar metals in contact in the presence of an electrolyte. If dissimilar metals are required, provide an insulator at their junction to break the galvanic circuit.

4. If possible, avoid direct coolant contact with electronics at a high potential. Ground all electronic equipment; do not utilize coolant system piping for a ground return.

5. In the selection of a coolant, carefully evaluate the chemical reaction of the coolant with all parts which it will contact, in either the liquid or vapor phase, over the operating temperature range. Choose a coolant which is chemically compatible. Include consideration of long term effects.

11.5.3 Maintenance aspects. Vaporization cooling systems are generally more prone to fouling than liquid cooled systems due to accumulation of scale deposits and contaminants. The initial effects of fouling at the evaporator may increase the efficiency of the boiling process due to roughening of the surface, but prolonged buildup will result in a significant thermal resistance. Sealed recirculating systems are less liable to fouling, provided adequate precautions are taken to exclude contaminants at the initial filling. Any system liable to fouling should include provisions for cleaning as part of the normal maintenance procedure.

Other maintenance provisions relative to liquid cooling systems (chapter 10, section 10.6.7) are equally applicable to vaporization cooling systems.

11.5.4 Safety. Vaporization cooling systems usually operate at a pressure differential relative to the environment. Design of the unit must insure that all housings, seals, piping, joints, etc. are capable of operating at this pressure differential without rupture or leakage. Dependent on the system design the internal pressure relative to the environment may be either positive or negative. A given system may be subject to both positive and negative internal pressures under different conditions; e.g. - positive during normal operation and negative when the electronics is shut off.

Extremely high positive internal pressures are possible in the event of a malfunction of either the electronic system, resulting in extra-ordinarily high heat dissipation, or of the cooling system, resulting in a failure to reject the dissipated heat to the ultimate sink. All vaporization cooling systems should incorporate a safety valve, set to release at a pressure low enough to insure that no rupture occurs of any cooling system part. The safety valve must be located so that the hot gasses cannot come in contact with operating or maintenance personnel.

A warning of differential internal pressure should be posted conspicuously on the exterior surface to alert personnel if the system should be opened during maintenance.

External surfaces of a vaporization cooling system likely to be touched by operating personnel should be kept low enough to prevent injury (usually less than 50°C (122°F)).

11.6 Design methods for predicting thermal resistances using vaporization cooling; application notes for groundbased, shipboard, aircraft, and spacecraft.

11.6.1 General. Vaporization cooling systems are capable of providing very low thermal resistance paths from a heat source to the ultimate sink. Thermal resistances on the order of 0.1°C/(watt/sq. in.) are practical. Non-expendable systems are relatively complex and expensive, and expendable systems involve replacement of the coolant and/or a weight penalty associated with the expendable coolant.

These systems should be considered (a) where unit heat dissipation or concentration is extremely high, or (b) where the total heat dissipation of an electronic system is too high for simpler cooling methods to dispose of, without incurring excessive temperature differentials, or (c) where short time, high dissipation cycles exist.

11.6.2 Procedure. The general selection and design of a vaporization cooled system should proceed along the following steps:

(1) Determine the maximum unit heat dissipation in watts per square inch and unit heat concentration in watts per cubic inch.

(2) Refer to chapter 7 to determine a logical choice or choices of cooling methods.

(3) Assuming that step (2) leads to vaporization cooling as a potential cooling system, the next step is to choose a coolant. Determine those coolants which are compatible with the application. Fluid compatability includes several areas:

(a) The fluid boiling point (at the system pressure) must be compatible with the temperature required for reliable operation of electronics.

(b) The fluid must not be detrimental to performance of the electrical system, and the electrical system must not degrade the fluid.

(c) The fluid must be compatible chemically with all materials used within the electronic system with which the fluid will come in contact.

(d) The fluid pressure must not exceed the capabilities of the system structure, including all joints and seals.

If no suitable coolant can be found (for example, because of an extremely sensitive high frequency circuit), an indirect vaporization cooling system is indicated. An indirect vaporization cooling system may also be required by other system restrictions, as for example, the need for high accessibility for rapid module replacement. This may occur in systems where interchangeable modules are used to set system parameters, such as frequency or power output. If the use of an indirect vaporization system is indicated, apply the fluid compatibility checks to that part of the system containing the fluid, and ensure low thermal resistance paths from the heat sources to the indirect vaporization cooling system.

Steps 3 and 4 (following) should include compatibility tests between the fluid and the system materials, unless previous useage can demonstrate that no problem exists. Be sure to include all materials in compatibility checks, such as elastomer seals, potting compounds, circuit board materials, solders, and plastics.

(4) Assuming several candidate fluids have been derived in steps 3, determine the peak heat flux for each, by use of equation 11-12 or 11-13. Apply a generous safety factor to allow for local "hot spots" and long term system degradation (50% derating available). Reject those fluids whose resulting peak heat flux is less than the maximum unit heat dissipation determined in step 1. If several candidate fluids remain, final selection must be made based on a tradeoff study including cost, availability (logistics), total coolant weight, and any other parameters pertinent to the particular application.

(5) Formulate an electro-thermal analog of the system, and determine the local heat transfer coefficients. Use equation 11-11 for the evaporator, and the appropriate equation from 11-14 through 11-19 for the condenser. Include any intermediate loops or contact resistances. Tests to correlate evporator heat transfer coefficients by use of equation 11-3 are highly recommended.

(6) Using the electro-thermal analog, refine the design to vary the thermal resistances as required to achieve acceptable maximum component temperatures.

(7) The particular application will determine the ultimate sink available. This ultimate sink may be a liquid (e.g. - seawater or liquid fuel), or gas (e.g. - air), or an expendable coolant. The condenser design will depend in large measure on the ultimate sink available.

(8) Complete the design of any intermediate thermal paths (liquid loops or contact resistances), pumps, piping, controls, etc.

The foregoing is a general design procedure, and will be modified for any particular application. The electro-thermal analog is a particularly useful tool for optimizing vaporization cooling systems, due to the complexity of these systems and the number of parameters involved. Larger systems generally require computer simulation for optimization. Verification testing is imperative for all but the simplest systems.

11.6.3 Application notes-groundbased systems. Groundbased systems may be either fixed site or mobile. Fixed site installations employing vaporization cooling techniques might include high power transmitters for communication or navigation, or power generating equipment. Liquid or vaporization cooling for these installations is generally integrated into a central cooling system, and makes use of available water or large refrigeration units as an ultimate sink. Mobile installations are usually standardized trailers, and employ refrigeration units as an ultimate sink for the cooling systems.

Large fixed site installations often supply treated water (filtered, de-mineralized, and de-aerated) for cooling. This water may be used as an expendable coolant or as a condenser fluid. In either case, the electronic system designer must coordinate his heat disposal requirements with the primary installation, in terms of pounds per hour of expendable coolant, or watts and allowable water temperature rise for condenser installations. The use of untreated water for vaporization cooling systems should be avoided. If used as an expendable coolant, the buildup of mineral deposits and scale will soon result in a high thermal resistance, greatly reducing the evaporator heat transfer rate. If used as a condenser fluid, the scale buildup within the condenser passages will likewise reduce its heat transfer capacity.

Mobile groundbased systems generally include self contained refrigeration units to provide both air conditioning for the occupants and cooling for the electronic systems. These may be separate systems (preferred) or integrated into a single unit. In either case, the electronic system designer must integrate the cooling requirements with the capacity of the trailer cooling system. Expendable coolant vaporization cooling systems must include a coolant reservoir to supply makeup coolant.

11.6.4 Application notes-shipboard systems. Shipboard vaporization cooling systems use the ships fresh water cooling supply as a condenser fluid, or employ a vapor-to-air heat exchanger as a condenser. Of the two, the fresh water condenser is preferable, both because water is much more efficient than air as a heat transfer fluid, and because the water provides a closed system. A forced air condenser must use either external or internal air. External air creates problems in ductwork and in variations in air temperature, water content, and entrained dirt. The use of internal air adds to the sensible heat load of the compartment air conditioning system. Indiscriminate use of compartment air for cooling has in the past created numerous situations of intolerable or uncomfortable compartment temperatures. Adding electronics system heat loads to the compartment air conditioning load should be avoided.

Because of the many additives, anti-freeze, inhibitors, etc., present in the ships fresh water cooling supply, and because this supply is not unlimited, it should not be used as an expendable coolant in shipboard installations.

11.6.5 Application notes-aircraft. Vaporization cooling systems for electronics in aircraft are usually either vapor cycle systems (section 11.3.5) or expendable coolant systems, commonly called "water boilers" (section 11.4). Selection of the optimum cooling system for aircraft electronics depends (among other factors) upon the aircraft flight envelope and mission profile. Alternate systems are evaluated in terms of the penalty of each on aircraft performance due to additional weight, drag, and power consumption. Reference 63 discusses two method of comparing alternate systems in terms of performance penalties (the Breguet method and the Stepwise Integration Method). Because of the wide range of variables involved, computer programs are frequently used to choose the optimum cooling system.

Two particular effects should be noted relative to aircraft installation of vaporization cooling systems: gravity variations, and attitude effects. Since both the boiling process and the condenser process are dependent on the value of the local gravitational constant, g, (equations 11-3, and 11-14 through 11-19), variations in g caused by aircraft maneuvering will cause temporary variations in heat transfer rates. Normally, these transient variations must be sustained for significant time periods, however, the effect of the change in the direction and/or intensity of the gravitational force field must be evaluated and included in the design to prevent system malfunction.

In addition to variations in the gravitational field, aircraft maneuvering will also cause attitude variations. (Rigidly speaking, an attitude change is a change of gravitational field direction. The distinction is merely one of convenience). Since a vaporization cooling system is a two-phase system, attitude changes will cause displacement of the vapor phase within the housing. The system must be so designed that no possible operating position will leave a heat dissipating component exposed with no liquid in contact with it. In order to maintain a constant heat transfer rate spray systems must ensure that an adequate spray pattern is maintained in all attitudes. All recirculating systems should investigate the effect of gravity field and attitude changes on pumps, to ensure that no damage occurs either thermally or to the pump by losing the pump head (running dry). This is especially trueof spray systems because of the limited coolant supply within the system.

11.6.6 Application note-spacecraft. There are only two ultimate sinks for spacecraft: (1) radiation into space from the exterior surfaces, or (2) expendable coolants. The weight penalty associated with expendable coolants generally precludes their consideration, except for extreme cases of short duration, high intensity heat transfer, as during a re-entry phase. There are no known examples of expendable vaporization cooling for electronics for spacecraft, other than the portable astronaut carried electronics for the Apollo Moon walks.

Non-expendable vaporization cooling systems except for heat pipes are ineffective on spacecraft in a zero-g environment. It is possible that a vaporization cooling system could be developed with forced circulation and a liquid-vapor separator. Such a system would be quite complex. Where an artificial gravity is supplied, as for example, by centrifugal forces, vaporization cooling could be employed.

11.7 Design examples.

11.7.1 General. The design and application of any vaporization cooling system includes complex relationships involving many variables. A comprehensive coverage of any particular design example would require so much space as to preclude its inclusion in this handbook. The examples following therefore present the final design of the systems described. It should be recognized that a great deal of analysis and testing preceded the evolution of any of these designs.

11.7.2 Example #1 - Airborne transmitter. A set of ten high power transmitters was originally designed and produced for use in a relatively low speed, patrol-type aircraft. The transmitters were cooled by means of direct liquid cooling, with the coolant rejecting its heat load to the slipstream air by means of a liquid-to-air heat exchanger.

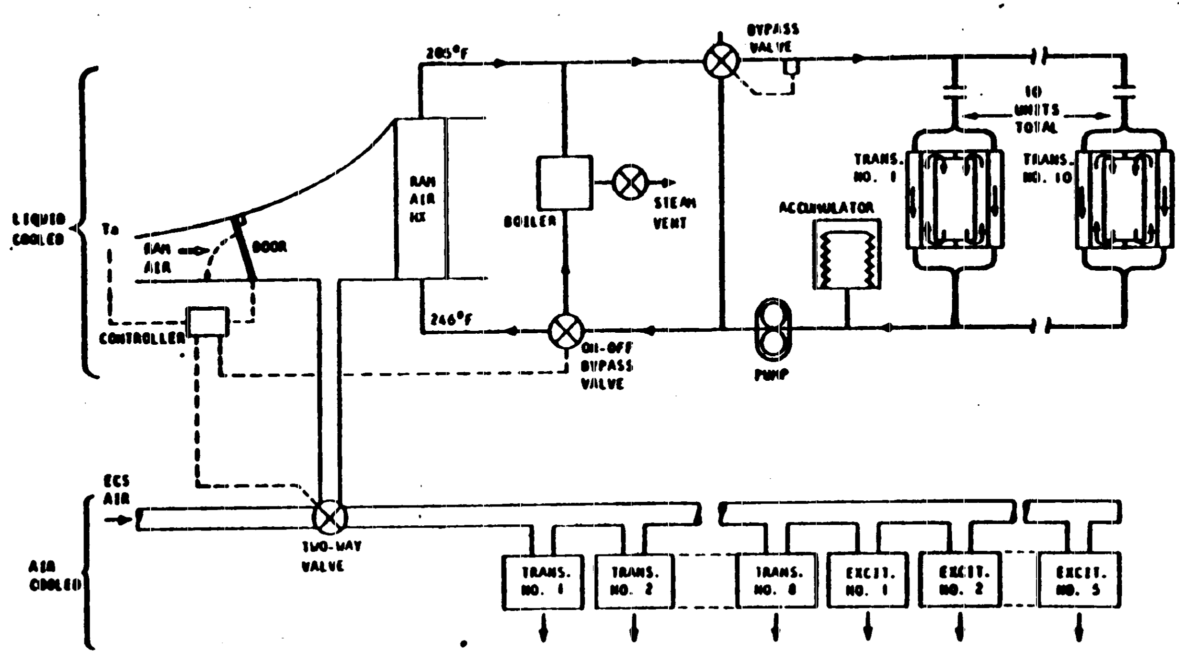
For reasons of economy (to avoid a costly redesign), it was desired to incorporate these same transmitters into a high speed attack aircraft, with a minimum of physical change to the electronics package.

The proposed system is shown schematically in Figure 129. (This system is the result of an extensive study of several alternate cooling methods.) The individual pumps associated with the transmitter enclosures are used to circulate the fluid through the enclosures and through the individual liquid-to-liquid heat exchangers integral to the enclosure. A single large pump is used to circulate a fluid through the individual integral heat exchangers and the large ram air heat exchanger, where the heat is dumped to the ram air. The liquid loop heat load for this system is as follows:

Electronic heat load	=	100.4 kw
Pump load (8 transmitters) 8 x 1.0 kw	=	8.0 kw
Pump load (2 transmitters) 2 x 0.8 kw	=	1.6 kw
Pump load (main circulating pump)	=	2.5
		112.5 kw

The main pump circulates 40 gpm of Coolanol-35 or equivalent, through a single ram air heat exchanger at a maximum inlet temperature of 262°F. A liquid return temperature to the electronic enclosures of 100°F minimum to 221°F maximum is maintained by a temperature actuated bypass mixing valve which mixes the cold fluid from the heat exchanger discharge with the hot liquid from the pump discharge to maintain this range of outlet liquid temperatures. This valve is used here primarily to protect the system against low temperature operation. The flow is distributed to each of the enclosures (i.e., 4 gpm to each) in a parallel flow arrangement controlled by a series of orifices in the coolant lines going to the individual enclosures.

The ram air heat exchanger is sized to reject the total heat load of 105.6 kw with an inlet air temperature of 160°F. The air exits the ram air heat exchanger and is dumped into the weapons bay compartment where it is eventually dumped overboard. The minimum airflow required at an air inlet temperature of 160°F is about 594 lb./min. to limit the temperature of the air exhausting from the ram air heat exchanger to 203°F (95°C). This temperature of 95°C is the maximum ambient temperature permissible with class 2X electronics. (Per MIL-E-5400)



NOTE:
1. RAM AIR DOOR OPEN Ta <160°F; NO ECS AIR
2. RAM AIR DOOR CLOSED Ta >160°F; ECS AIR IS USED, WATER BOILER IS USED
3. EXIT AIR TEMPERATURE FROM RAM AIR HEAT EXCHANGER ≤205°F
4. BYPASS VALVE PROVIDES MINIMUM 100°F COOLANT TO ENCLOSURE HEAT EXCHANGER

FIGURE 129. System Schematic

During operation on ram air, a portion of the air is ducted to the air cooled electronics where it picks up about 6.5 kw of heat (4 kw from the transmitter and 2.5 kw from the five exciters). At inlet air temperatures in excess of 160°F, the ram-air door is closed and cooling air is supplied to the air cooled electronics from the aircraft's air cycle refrigeration system. In addition, the liquid coolant is diverted through the water boiler where the system heat is rejected by boiling off water. Table XXXI presents a summary of the design point performance under both ram-air and boiling modes of operation. Figure 133 shows the operating regions of this system superimposed on the aircraft flight envelope.

Water Boiler

The water boiler selected for the system uses the core from the water boiler currently used to cool bleed air under some flight conditions. A picture of this heat exchanger is shown in Figure 131. The boiler has the capacity to reject the total system heat load of 105 kw with a maximum coolant outlet temperature of 221°F. The water boiler is a tubular heat exchanger that is placed in the bottom of a water tank. The Coolanol-35, or equivalent, is pumped through the tube side of the heat exchangers in a six-pass flow configuration. The water is boiled on the outside of the tubes and the resulting steam vented overboard. The size of the tank will depend on the amount of time desired to operate the system in the boiling or partial boiling mode. If additional water capacity is desired, supplemental water tanks can be placed remotely from the boiler assembly, and the water fed to the boiler. The water boiler assembly weighs 12 lbs. and occupies a volume of about 500 cu. in. The design point for the boilers was taken as 20,000 ft. This results in a design boiling point of 181°F, which provides the desired temperature gradient between the Coolanol, or equivalent, and the water. At altitudes below 20,000 ft. the boiler will be capable of less than full power dissipation.

11.7.3 Example #2 - Radar system, airborne. The high operating voltage and power dissipation associated with airborne radar systems represent a significant challenge in the development of a high performance radar package that requires minimum space and weight.

With transmitter operating voltages in excess of 15,000 volts and power dissipation involving several kilowatts, it is not practical to allow transmitter components to operate in air. Electrical breakdown of the air would occur unless large spacings were provided between the high voltage points and ground. The limited heat removal capability of air would also require greater space in the transmitter package.

The use of transformer oil as a coolant provides some space and weight savings, but because of its low permissible heat flux, which requires a prohibitive amount of space, the component temperature remains high.

The selected method of packaging places a complete airborne radar transmitter inside a flexible rubber liner filled with a fluorochemical dielectric coolant. The flexible rubber liner compensates for changes in the volume of the liquid due to expansion and contraction over the wide environmental temperature range.

TABLE XXXI. System Design Point Performance*

Ram Air	
Altitude	48,000 ft
Day	Hot (Mil Std 210A)
Ram air temperature	160°F
Ram airflow	
Heat exchanger	594 lb/min
Air cooled electronics	88 lb/min
Coolanol-35* flow rate	40 gpm
Coolanol-35* delivery temperature	221°F
Boiling Mode	
Altitude	≥20,000 ft
Airflow (ECS air at 77°F)	24.5 lb/min
Coolanol-35* flow rate	40 gpm
Coolanol-35* delivery temperature	221°F
Water consumption	6 lb/min
*Heat Loads	
Liquid loop = 105.6 kw	
Air cooled = 6.5 kw	

* or equivalent

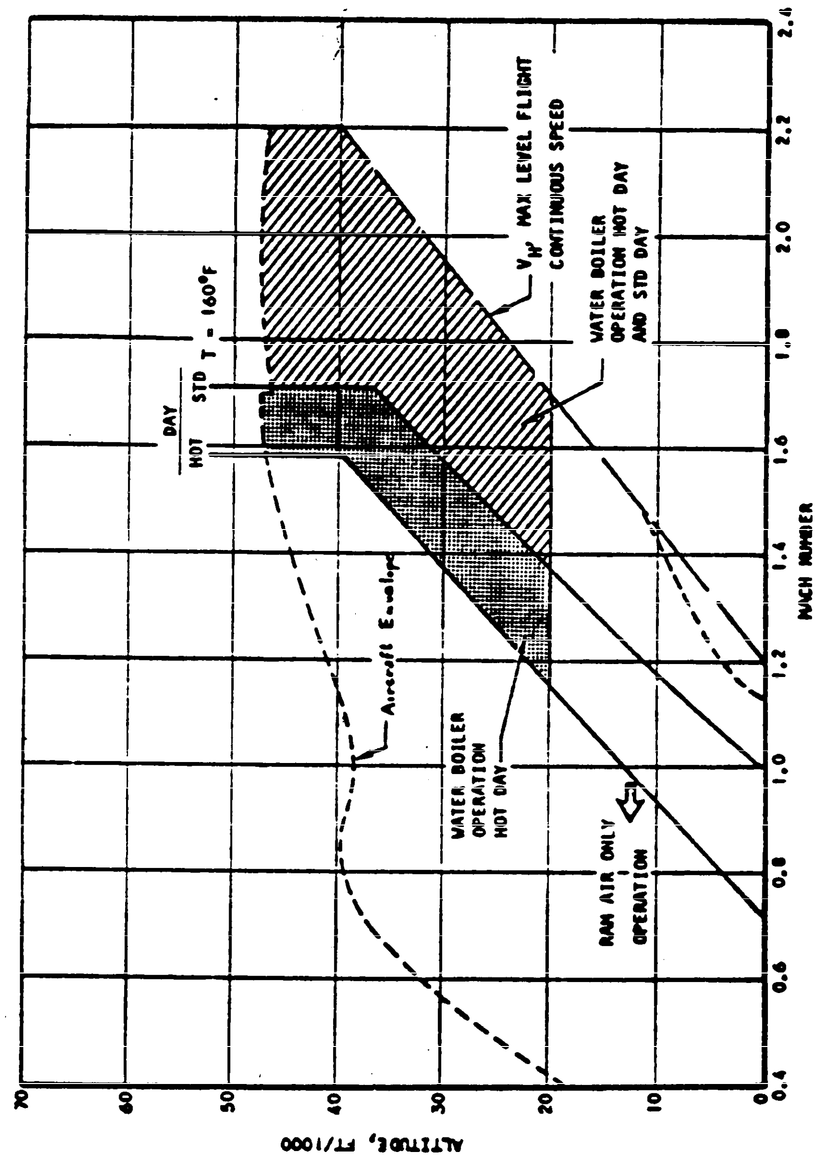


FIGURE 130. Aircraft Operating Envelope

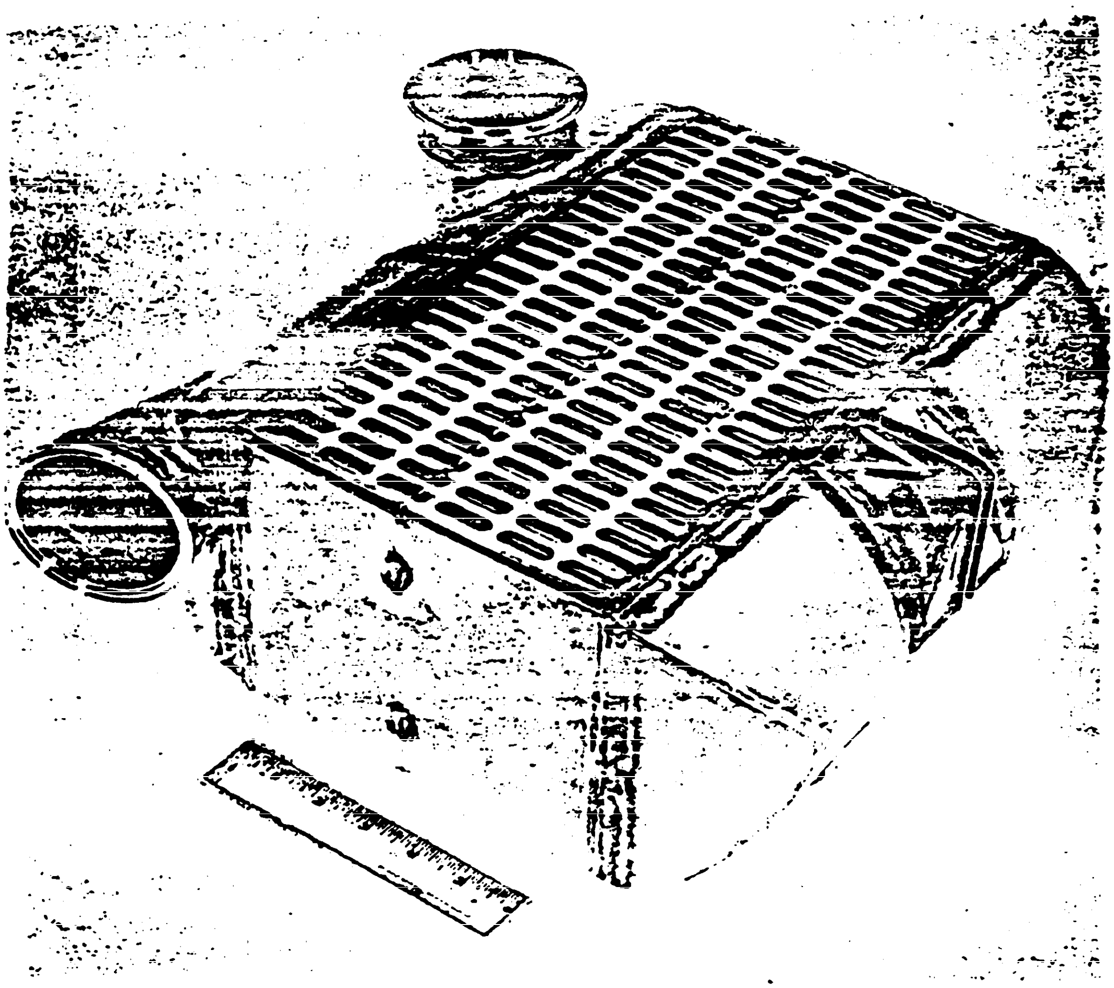


FIGURE 131. Water Boiler

The nucleate boiling region, up to and including the critical point.

This package is one third the size of an air cooled transmitter and results in a 40% size reduction when replacing an oil cooled system.

The key to the new packaging method is a Fluorochemical liquid whose dielectric strength of 350 volts/mil provides ample protection between high voltage points and ground potential.

The packaging method incorporates a vaporization cooling technique that takes advantage of the large amount of heat transferred when a liquid boils.

The low boiling characteristics of this inert Fluorochemical liquid produce very high heat transfer coefficients, normally 1-2 watts/in.^{2}/degree C. Power dissipating components such as resistors, transformers, chokes, and electron tubes are immersed in the liquid. Heat developed on component part surfaces causes local boiling to occur. The vapor formed rises as bubbles and passes through the liquid to a vapor space above. The heated vapor is cooled by a cold plate and condenses as liquid droplets.

As a coolant, this Fluorochemical liquid has increased the life and improved the performance of electronic components. The liquid's boiling point permits the components to be operated at lower temperatures than if cooled by oil or air.

Because of its completely fluorinated structure, the liquid is non-flammable and exceedingly non-reactive chemically. The absence of any chlorine or hydrogen in its molecular makeup results in low solvent properties. This, combined with a high degree of chemical inertness, makes the fluid compatible with materials of construction.

Recently, a two-year old transmitter that had accumulated 400 operating hours in the field was opened and examined.

Upon examination, there was no deterioration of the plastics, elastomers or metals in contact with the fluid during the two year period. In addition, the fluid maintained its high dielectric strength.

11.7.4 Example #3 - Radar system, ground. Temperature stability recently posed a problem in design of a radar system component being developed.

The basis of this problem was whether a cooling system could be found that would guarantee steady frequency performance of a klystron used as a pump for a parametric amplifier installed in airport surveillance radar units.

A heat transfer system was designed, which featured a captive atmosphere where heat removal and temperature stabilization are accomplished through boiling and condensation. (See Figure 132)

A boiling inert fluid was selected that stabilized the temperature, and thus the frequency of the klystron tube and cavities operating under varying ambient temperatures.

The cooling system proved efficient, inexpensive, uncomplicated, and needed a minimum of maintenance. The fluorocarbon cooling agent selected had the unique ability to maintain a constant temperature despite severe changes in the ambient temperature external to the system.

Previous techniques of temperature stabilization involved expensive electronic or mechanical heat removal systems, many of which were complicated, needed power for operation, and in some cases, required a thermostat.

The boiler is self-contained, independent of outside controls, and operates on a continuous fluid-vapor fluid cycle, which prevents dissipation and waste of cooling agent.

The boiler was designed to insure the lowest possible change in klystron frequency or power output, despite temperature fluctuations, in a parametric amplifier system.

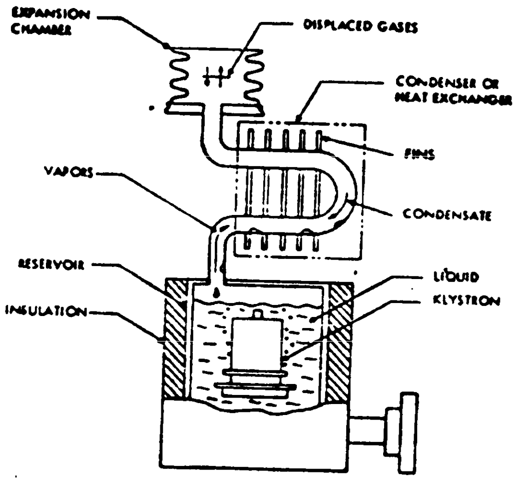


FIGURE 132. System Schematic

Tests showed the klystron (producing 60 watts at a frequency of 12.57 ghz) changed less than 3.0 mhz during ambient temperature changes ranging from -12°C to 58°C when temperature stabilized. Without the boiler, the frequency changed about 0.77 mhz per degree C change in ambient temperature, or about 53 mhz over the same temperature range.

The basic feature of the cooling system is that the boiling point of the fluorochemical fluid is keyed to the peak operating temperature of the klystron. When the klystron heats the inert fluid to its boiling point, the temperature of that liquid will remain at the boiling point despite additional energy from the klystron.

In the above closed-cycle system, fluid vaporized by the heat of the klystron is channeled through cooling tubes to a condenser and then back to ~~the~~ fluid reservoir where the cycle is repeated. This method required a liquid ~~that would not~~ break down under repeated boiling, was chemically inert with all parts of the system, and had a boiling point near the temperature required by the klystron (99°C to 107°C). Properties of this fluid include thermal stability, low surface tension, inertness to physical and chemical change, low freezing point, an electric strength in excess of 35 kv with a dielectric constant of 1.8, and a dissipation factor of less than 0.0005. Another advantage is that the heat transfer capacity of its vapor is nearly that of the liquid.

Boiling creates vapor pressure inside the sealed system which tends to increase the boiling point of the fluid, and in turn raise the temperature of the klystron. As a solution to this problem, a flexible elastomer bellows was designed that expands gently during vaporization and maintains a reasonably constant internal pressure.

The expansion chamber accepts the displaced gases in the system when vapors are generated in the fluid reservoir and must be capable of holding all displaced gases at the highest ambient temperature that the system will encounter.

The klystron may be immersed directly in the coolant or it may be jacketed. Direct immersion gives better temperature control and eliminates hot spots but the necessity of passing leads and tuning couplings through the wall of the reservoir (which must be resealed each time the klystron is removed) tends to increase fabrication and assembly costs.

The alternative is to insert the klystron in a housing, which allows the klystron to remain dry and eliminates the reservoir, fittings, seals, and windows. Removal and replacement of the tube is simple and costs are reduced. However, this method has the disadvantage of permitting the klystron to operate at a higher temperature with less temperature stability.

The condenser can vary in construction, but it must be capable of removing the required quantity of heat under a full range of ambient temperatures. It can be designed to pass off heat by natural air convection, forced air convection, or liquid convection. The condenser must also accommodate the expansion chamber.

Hot vapor is cooled by the condenser and the condensate is drained back into the reservoir through one or more tubes.

The stable output from the klystron is used to pump an S-band parametric amplifier designed to improve receiver sensitivity. Tests showed improved sensitivity from 5 to 7 db while making possible an operating system noise figure of 3 db.

11.7.5 Example #4 - Airborne transformer. Modern military aircraft employ electronics systems and weapons systems requiring large amounts of primary power. Projections indicate that primary power demands will increase by orders of magnitude in future aircraft. Programs are in progress to develop systems capable of generating, regulating, and distributing this power, consistent with the lightweight design inherent to airborne equipment.

On necessary component of any such system is a power transformer. Note that with a typical transformer system output of 5 Megawatts and a transformer efficiency of 90%, the resultant heat load is 500 kw. Dissipation of this magnitude presents two major problems to thermal designers: (1) that of heat disposal into the ultimate sink, and (2) preventing excessive temperature rise within the heat generating elements. The second of these problems is further aggravated by the requirements of small size and weight for airborne applications, thereby increasing the unit heat dissipation and concentration.

An analysis indicated that the transformer winding internal temperature rise could be maintained within acceptable limits if cooling could be achieved at the rate of 50 watts/sq. in. of conductor surface area. Reference to chapter 7 will show that this cooling rate can be achieved by boiling heat transfer. A transformer designed for boiling heat transfer raises several significant problems, however, Windings must be spaced to allow liquid to reach all conductor cooling surfaces and to allow escape of the generated vapor. Pockets or obstructions which might cause vapor lock must be eliminated. This results in a "free standing" coil, with minimum mechanical support. On the other hand, tests have demonstrated current carrying capabilities of over 100,000 amperes per square inch in copper with a conductor temperature rise of less than 20°C over the bulk liquid. This allows the use of relatively fine conductors, which alleviates the individual conductor support problem, since fine conductors can be tightly wound into single layer solenoids which are individually cooled. The great reduction in conductor size for any given current results in a smaller winding, a smaller core window, and thus a much smaller core. Size and weight reduction of about 5 times better than the best of the state of the art has been achieved.

12. SPECIAL COOLING TECHNIQUES

12.1 General. This chapter discusses special and unusual cooling techniques including heat pipes, thermoelectric cooling, Hilsch vortex tubes, refrigeration, catalytic chemical reactions, cryogenic cooling, and sublimation. Several of these techniques involve refrigeration and provide negative thermal resistances. Some techniques offer significant advantages in cooling electronic equipment in commonly encountered thermal situations while the application of other techniques can only be justified under very special conditions. It is recommended for each contemplated application of these special cooling techniques, that a tradeoff study be performed to determine the merits of applying the special technique compared to the more conventional techniques discussed in chapters 8 and 11.

12.2 Heat pipes.

12.2.1 Characteristics, advantages, and disadvantages. The heat pipe is a thermal conductor of very high conductivity. It is essentially a closed evacuated chamberlined with a capillary structure or wick. See Figure 133. Heat is transported by evaporation of a suitable volatile fluid, which is condensed at the cold end and returned by capillary force to the hot end. The vapor passes through the cavity. Heat pipes can be constructed in practically an endless number of configurations, but always consist of three zones or sections, namely; the evaporator, condenser, and the adiabatic section connecting the other two (Reference 75). In some designs the adiabatic section may be very short.

The heat pipe has antecedents in the Condor Cooler and other devices using evaporative cooling coupled with condensation at a heat sink interface. The basic principle was first advanced by R.S. Gaugler in 1942, but was not applied at that time. (U.S. Patent No. 2,350,348 issued in 1944.) G.M. Grover independently conceived the idea, coined the name, and developed practical models in 1963. (U.S. Patent No. 3,229,759 issued in 1966.)

The heat pipe has seven important properties potentially useful in electronic equipment cooling systems (References 73, 74, 76). First, it has many times the heat transfer capacity of the best heat conducting materials on a weight and size basis. Second, it maintains an essentially uniform temperature equal to the boiling temperature of the fluid over the surface area at the evaporator, thus automatically providing a uniform heat source temperature. Third, the areas and configurations of contact with heat source and sink are independent and can be designed separately to suit various applications. Fourth, it can transport heat over considerable distances (several feet) with a very small temperature drop. Fifth, it may be constructed entirely of electrically insulating material, and so can be coupled directly to high voltage equipment. Sixth, it requires no power for pumping condensate, since capillary pumping receives its power from the heat itself. Seventh, it operates satisfactorily in a zero gravity environment.

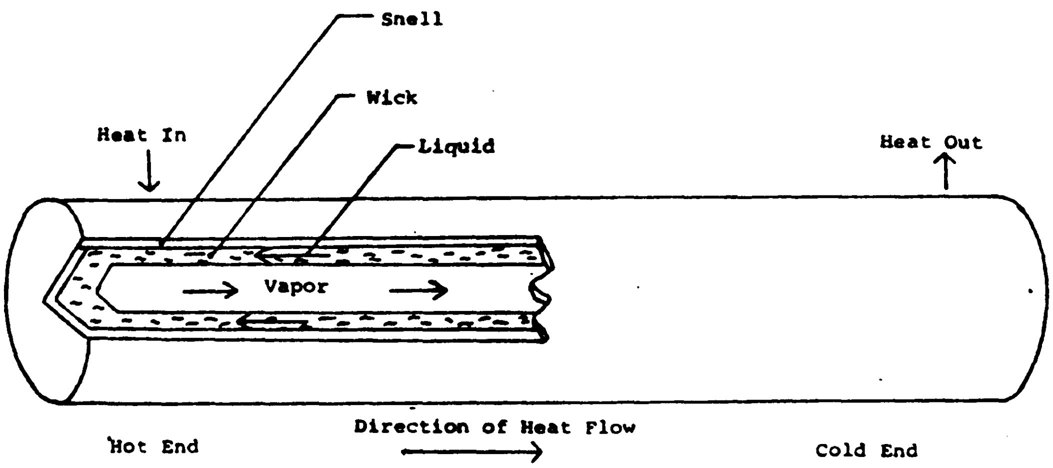


FIGURE 133. Basic Heat Pipe

As with all devices there are compensating disadvantages. First, a heat pipe is sensitive to gravitational force. The thermal resistance at a given heat flux decreases when the cold end is higher than the hot end, and vice versa. In applications to high performance aircraft this is an important consideration since hot end temperature could vary widely during maneuvers. A multiple design of heat pipes could be used to compensate for this defect. Second, freezing of fluid during inoperative periods at low environmental temperature is possible in some cases and requires careful startup procedure. Third, the heat pipe cooling system has thermal capacitances which cause transient temperature variations when the heat flux is changed. This phenomenon can cause difficulties particularly during startup. Fourth, since the fluid vapor pressure is a function of temperature, it is possible for excessive pressure to develop in a heat pipe envelope, particularly during transient conditions. This can result in catastrophic failure (explosion). Fifth, the hot end temperature of a heat pipe is normally a function of the heat flux. For electronic equipment cooling it is desirable to maintain a constant temperature with varying heat flux. Heat pipe temperature control methods are being developed, control methods are discussed at length. Heat pipe control is a wide open field, with many possibilities and few practical answers.

In a well designed heat pipe the temperature, and therefore the pressure of the vapor is nearly constant with length. There must be some slight gradient however, to cause vapor flow. The liquid pressure in the wick has a larger and a relatively constant gradient. The ideal and actual pressures are shown in Figure 134. Vapor temperature is practically constant over the entire evaporator section, drops slightly through the adiabatic section, and then drops rapidly and linearly through the condenser section. Liquid temperature in the wick is approximately linear, flattening some in the evaporator. $T_V - T_L$ is nearly constant in the evaporator, and varies linearly in the condenser.

Temperature on the outer surface of a heat pipe evaporator is higher than that of the liquid in the wick due to the thermal resistance of the wick and pipe wall.

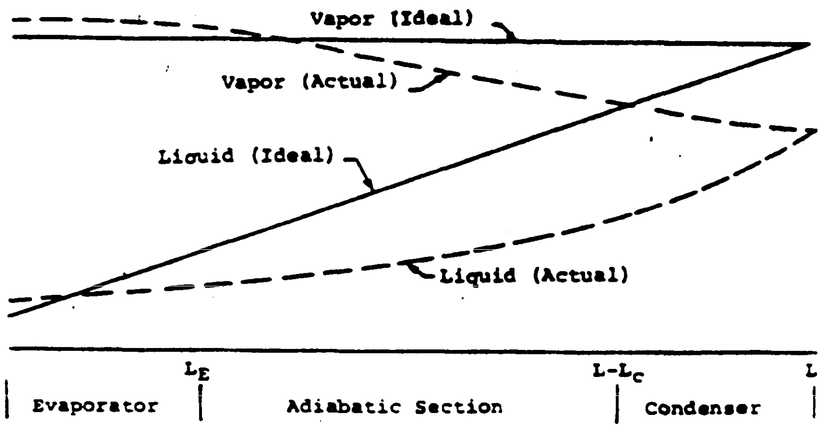


FIGURE 134. Pressure Gradients in a Heat Pipe

The nearly uniform vapor temperature varies approximately with the heat flux. Figure 135 shows the relative heat flux vs. T_V calculated for a small heat pipe. Figure 136 shows published experimental data for wick temperature for two heat pipes. (Reference 81)

12.2.2 Present state of the art and applications. Heat pipe R&D has reached the stage where standard production models are commercially available, chiefly in the form of cylindrical tubes with various heat transport ratings. Straight, bent, and flexible configurations, and a few flat plate forms, are on the market. Many electronic cooling applications, however, require special configurations which are not off the shelf items. Consequently, these applications require specific, detailed design based on carefully developed comprehensive specifications.

Since heat pipe fabrication for high reliability demands highly specialized design talent, manufacturing facilities and know-how, it is advisable to employ the services of specialists in the field to design and fabricate heat pipes for cooling electronic equipment.

A numb er of new companies have become established specializing in heat pipe design and manufacture. Also a number of well known large companies maintain divisions dealing with this specialty.

Heat pipes fall into three general classes. First, high temperature high performance applications operating at 500 to 2000°C. Second, applications at moderate temperature of 0 to 250°C. And third, low temperature and cryogenic applications. The first class has received the earliest and probably the most intensive development, chiefly by AEC, Sandia Laboratories, Los Alamos, etc. Metals ranging from sodium (500-900°C) to silver (1500-2500°C) are used as the heat transfer fluid and applications are chiefly in nuclear reactors. The second class is of greatest interest to electronic equipment designers. Many fluids are applicable, including water, various alcohols, benzene, Coolanol*, Dowtherm*, various freons*, and sulphur.

Methanol is usable from -50 to +130°C; water from 50 to 200°C; Dow Corning DC 200* from -50 to +150°C; Dowtherm* from 20 to 300°C. The two latter are dielectric liquids. The third class has received very little attention. The low freons, ammonia, sulfur dioxide, nitrogen, and helium are possible fluids. (Reference 79) Electronic engineers involved with infrared and laser equipment, in particular, should be aware of the possibilities of cryogenic range heat pipes.

It is very important to realize that, while a heat pipe is intrinsically an excellent thermal conductor, its effectiveness in a cooling application is strictly limited by the thermal resistances at its interfaces with the heat source and sink. This fact is demonstrated in the theoretical discussion, especially in paragraph 12.2.8 wherein an equivalent thermal circuit is developed.

Traveling wave tubes and other electronic devices incorporating high voltage electron beam collectors are difficult to cool and heat pipes using eielectric fluid and shell have been applied successfully. (Reference 92, 93) The characteristics of several typical heat pipes follow.

An experimental flat heat pipe for cooling two cold plates each 5 x 10 inches has been developed and tested. Thermal resistances of 0.01°C/watt with coolant air and 0.0045°C/watt with coolant water were achieved.

A cylindrical production type heat pipe 3/8 inch in diameter, with a 5 inch heater length, a 4 inch condenser length, and a 7 inch adiabatic transfer distance, transmitted 58.4 watts with temperatures of 133°C at the hot end and 89°C at the condenser section. The thermal resistance over the 7 inch length is 0.753°C/watt, compared to 6.5°C/watt for a solid copper rod of the same size.

In each of the above instances the time constant was observed to be long, since 20 to 40 minutes were required to reach constant temperature distribution with a steady heat load. The heat pipe itself has a time constant measurable in seconds (Reference 83) but the thermal capacitances of the source and sink cause the overall time response to be much slower. Overall time constants of 30 to 40 minutes are not uncommon. (Reference 81, 96)

Heat pipes have been successfully applied as integral parts of high voltage power supplies, power diode switches, infrared detectors, extra-vehicular life support systems, nuclear power generators, and space vehicles. High temperature heat pipes several feet in length are also in use in nuclear reactor installations.

Several thousand wick fed evaporators have been installed in F-104 aircraft as part of the air cycle cooling system. While not strictly heat pipes, these devices have several characteristics in common, particularly capillary liquid transport, and versatile geometry which provides a low spreading thermal resistance.

Small power supplies using radioactive isotopes have become practical because of the ability of the heat pipe to concentrate heat from the lower power level source. Conversely, the heat from concentrated sources is spread over large radiating surfaces in satellite cooling systems. Many applications have also been made in spacecraft where the effects of gravity are absent. (Reference 98)

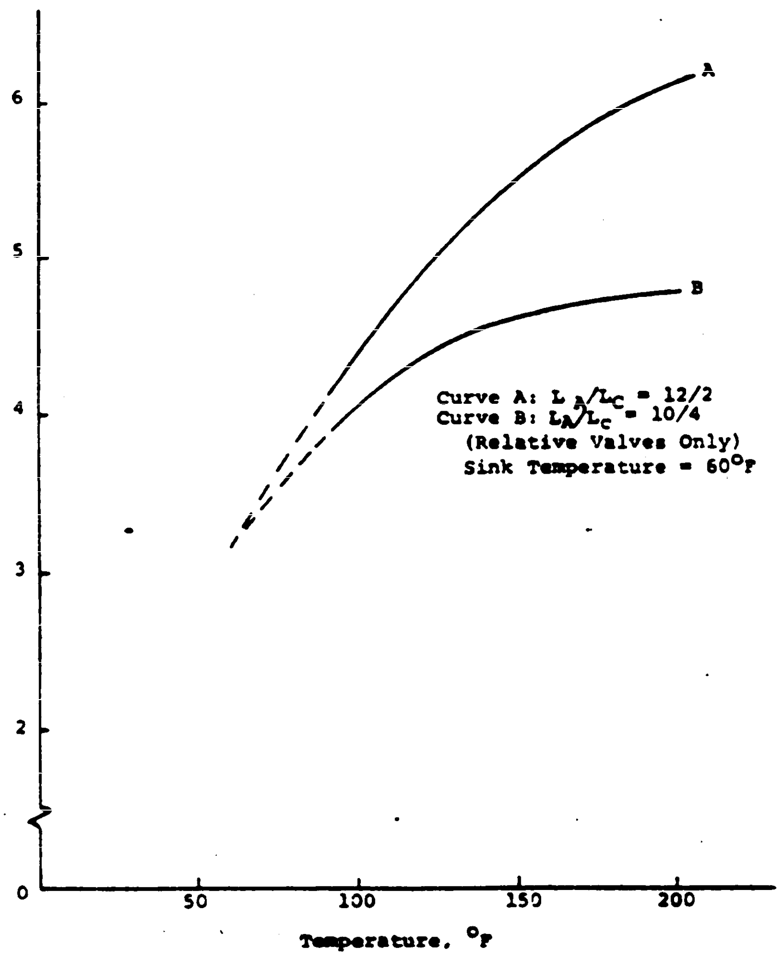


FIGURE 135. Heat Flux vs. Evaporator Temperature

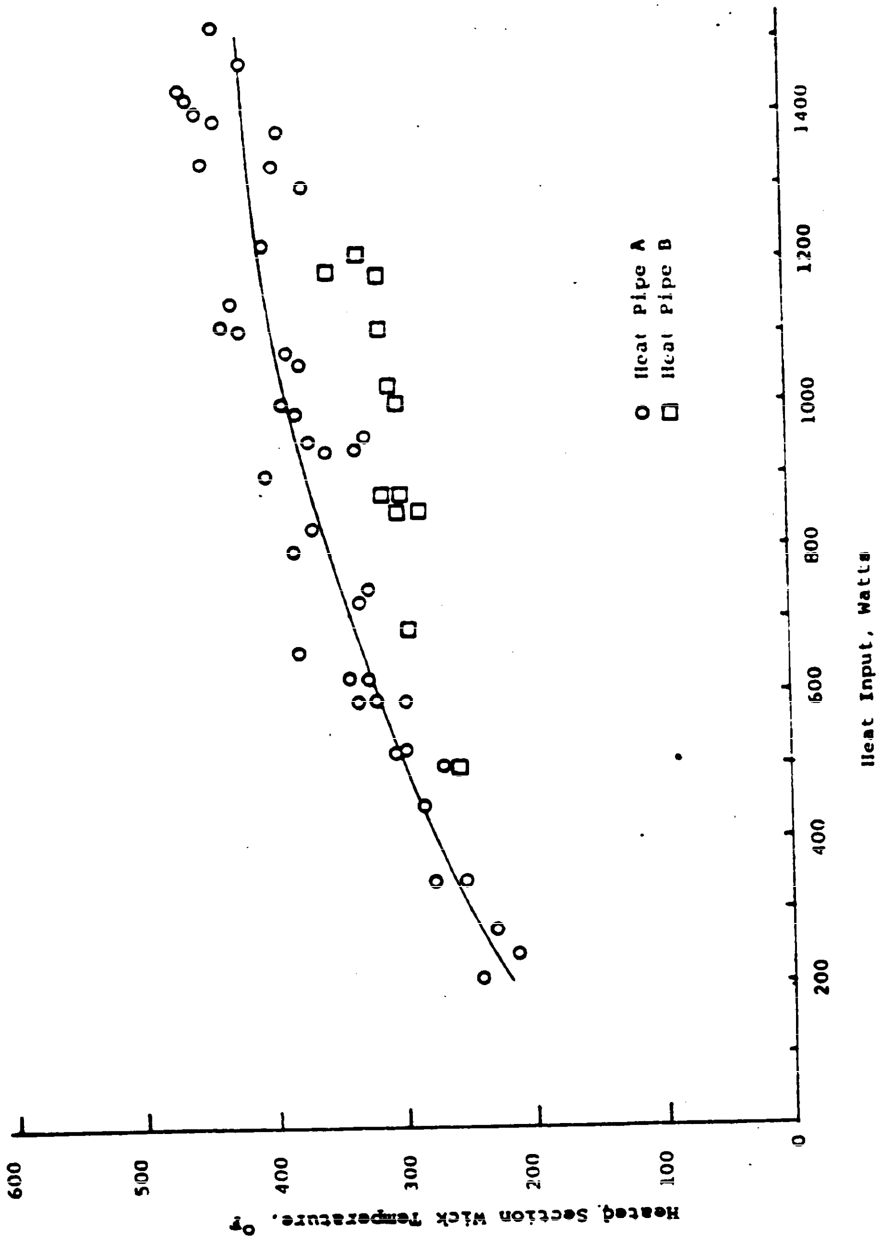


FIGURE 136. Wick Temperatures vs. Heat Input (Ref. 81)

A variation of the heat pipe where condensate is returned to the evaporator section in a thin film by centrifugal force instead of by capillary force has been developed for cooling rotating machinery. (Reference 95) Another wick substitute consisting of electrodynamic force acting on a dielectric working fluid is the subject of a patent disclosure dated July, 1972. (Reference 94)

12.2.3 Basic theory. Heat pipe design and fabrication is a highly specialized field, partaking sometimes even of a "black art," and obscured by proprietary secrets. The following paragraphs are intended to give the electronic design engineer some insight into the behavior of heat pipes in order to assess their applicability and assist him in preparing feasible specifications.

Heat pipes can be constructed in an almost limitless number of configurations. For theoretical development the most convenient model is a cylindrical tube closed at both ends, open through the center, and lined with a layer of porous material known as the wick, as shown in Figure 137.

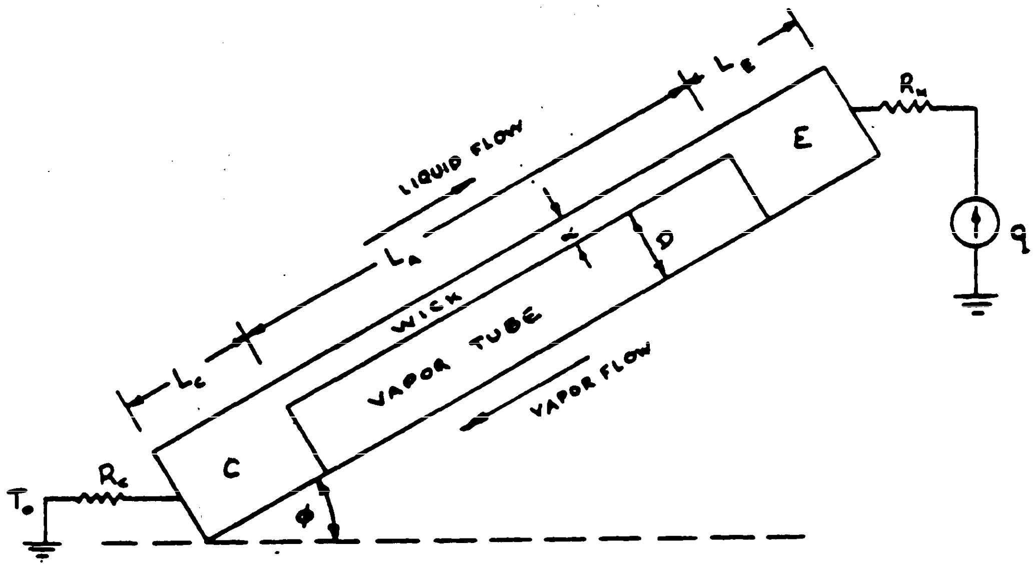


FIGURE 137. Heat Pipe Schematic

The nomenclature is as follows:

c = condenser section
E = evaporator section
A = vapor flow region, of diameter D

The passage of diameter d represents the wick, or capillary liquid flow path.

d = equivalent diameter of wick
D = equivalent diameter of vapor flow path
L_C = length of adiabatic section
L_A = length of adiabatic section
L_E = length of evaporator section

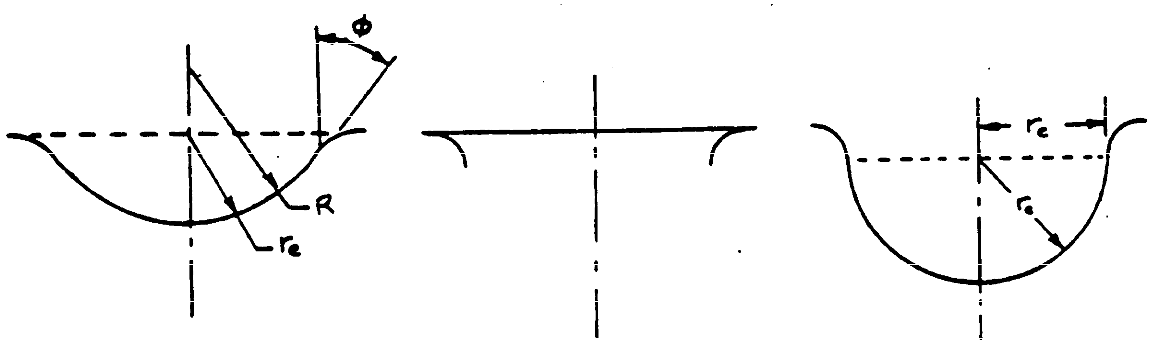
T_o = temperature of sink
T_s = temperature of heat source
R_C = interface thermal resistance, sink to heat pipe
R_H = interface thermal resistance, source to heat pipe

The basic equation for the circulation of fluid in the heat pipe is (Reference 5, 16):

$$\Delta P_c \geq \Delta P_g + \Delta P_L + \Delta P_V \tag{12-1}$$

Where:
ΔP_c = pressure difference due to capillarity
ΔP_g = pressure difference due to gravity
ΔP_L = pressure difference due to liquid flow
ΔP_V = pressure difference due to vapor flow

12.2.3.1 Capillary pumping pressure. The actual performance of a heat pipe wick can be computed from physical dimensions only for a few simple geometries (Reference 80). For most practical wicks, the performance must be determined experimentally. For purposes of analysis, the wick is considered to be a porous path of cross section A_w, a fraction of which is permeable by liquid, terminating at each end in sections perforated by circular pores of radius r_e. The liquid surface at each pore forms a meniscus of radius R, as shown in Figure 138.



$r_e = \kappa \cos \phi$ $\qquad$ $R = \infty;\ \phi = \frac{\pi}{2}$ $\qquad$ $R = r_e;\ \phi = 0$

FIGURE 138. Range of Circular Meniscus Curvature Variations

When fluid flow occurs due to evaporation and condensation, the meniscus radii at the evaporator and condenser end will be different.

Surface tension develops a pressure difference across the liquid-vapor interface in a capillary tube, according to the LaPlace and Young equation (Reference 79).

$$\Delta p = \frac{2\,\sigma \cos\theta}{r_e} \tag{12-2}$$

Where:
σ = surface tension, lbm/sec.^{2}
r_e = hydraulic radius of the tube, in.

Since $\cos\theta = r_e/R$ where R is the radius of curvature of the meniscus.

$$\Delta p = \frac{2\sigma}{R} \quad . \tag{12-3}$$

The total pumping pressure difference in a heat pipe is:

$$\Delta p_c = 2\sigma \left(\frac{1}{R_E} - \frac{1}{R_c} \right) \tag{12-4}$$

where R_E and R_C are the radii of curvature at evaporator and condenser ends of the wick, respectively.

The maximum and minimum values of θ are 0° and 90°, so the maximum value of capillary pressure is:

$$P_c \max = \frac{2\sigma}{r_e} \tag{12-5}$$

This occurs at maximum evaporation rate when liquid is completely filling the condenser end pores and being converted to vapor at the evaporator end.

The capillary pressure is strictly a surface effect, and has nothing to do with conditions in the bulk vapor and liquid, except so far as surface tension is affected by temperature. However, the temperature throughout the heat pipe wick is nearly constant during normal operation. The pores on the wick surface are not circular but in general have irregular shapes. Their geometry is almost impossible to define except for packed spheres or for grooved wicks. Equation 12-5 is therefore idealized. Later it will be shown how p_c can be defined in terms of quantities that can be measured for any fluid and wick structure.

12.2.3.2 Gravity effect. The pressure drop due to gravity acting on the liquid in the wick is given in equation 12-6.

$$p_g = \rho_L g L \cos \phi \tag{12-6}$$

Where:
ρ_L = liquid density, lbm/cu. in.
g = gravitational acceleration, in./sec.^{2}
L = effective length of saturated wick, in. approximately,

$$L = L_A + (L_c + L_E)/2.$$

ϕ = angle between heat pipe axis and gravitational field, deg. (Figure 137).

12.2.3.3 Liquid pressure drop. The pressure required to force liquid through the wick is calculated by Darcy's Law, assuming laminar viscous flow through the effective flow area of the wick, using the flow friction factor 64/Re (Reference 78):

$$\Delta p_L = \frac{32 \mu_l L_m}{\rho D_{e_f}^{\,2} A_{e_f}} \tag{12-7}$$

Where:

m = mass flow rate, lbm/sec.
μ = liquid viscosity, lbm/in.-sec.
D_{e_f} = effective diameter, in.
A_{e_f} = effective wick area, sq. in.

Equation 12-7 corresponds to that given by equation 12-5:

$$\Delta p_L = \frac{\mu_l L_m}{\rho_L K A_w} \tag{12-8}$$

Where:

K = wick permeability, sq. in.
A_w = total wick area, sq. in.

Since for most wick structures K, D_{ef}, and A_{ef} cannot be calculated, the usual procedure is to measure K for the type of wick and fluid and the expected temperature to be used. This is done as indicated in Figure 139. The flow rate m through a wick of known L and A_w is measured with a constant static head H, and K is calculated by equation 12-10.

$$\Delta p_l = \rho_L g H = \frac{\mu_L L_m}{\rho_l K A_w} \tag{12-9}$$

$$K = \frac{\mu_l L_m}{\rho_l^{\,2} g H A_w} \tag{12-10}$$

Values of K are given for several wick fluid combinations in pertinent technical literature. (References 77, 78, 88, 91, and 97)

12.2.3.4 Approximate performance calculation. In a heat pipe with a wick area much smaller than the area of the vapor passage, the pressure drop due to vapor flow is small. Neglecting this, and substituting equations 12-5, 12-6, and 12-9 in 12-1, the maximum possible mass flow rate is derived:

$$m_{max} = \left(\frac{\rho_L \sigma}{\mu_L}\right)\left(\frac{KA_w}{L}\right)\left(\frac{2}{r_e} - \frac{\rho_L g L}{\sigma} \cos \phi\right) \tag{12-11}$$

the maximum heat rate is:

$$q_{max} = m_{max}\,\lambda = \left(\frac{\rho_L \sigma\, \lambda}{\mu_L}\right)\left(\frac{KA_w}{L}\right)\left(\frac{2}{r_e} - \frac{\rho_L g L}{\sigma} \cos \phi\right) \tag{12-12}$$

Where λ = heat of vaporization, watt-sec./lbm

With the dimensions indicated, q_{max} is in watts. This development considers only the wick limited performance. Other limitations on heat pipe performance are discussed in paragraph 12.2.6.

The first bracketed term depends on fluid properties evaluated at the wick temperature which can be considered constant at the average of the fluid temperature at the evaporator and condenser ends. It is known as the liquid transport factor and is a criterion of the relative effectiveness of heat pipe fluids.

$$N_f = \frac{\rho_L \sigma \lambda}{\mu_L} \tag{12-13}$$

Values of N_f for several fluids and temperatures are given in References 77, 78, 79, and 88.

The pore radius cannot be determined except for wire mesh, perforated screens, and other precisely made wick surfaces. For most types of wicks it is an average of many randomly varying values. The pore radius can be measured indirectly by determining the greatest height, L_m to which the liquid rises in the wick. (References 84 and 91)

$$\rho L\ g\ L_m = \frac{2\sigma}{r_e} \tag{12-14}$$

Substituting in equation 12-12,

$$q_{max} = N_f \left(\frac{KA_w}{L}\right)\left(\frac{\rho_L g}{\sigma}\right)\left(E_M - L \cos \phi\right) \tag{12-15}$$

The equations for maximum heat transmission developed here involve several approximations. More precise equations are available for design work. (References 78, 79, and 81) Following are some of the refinements.

For some wick structures a distinction should be made between the porosity for bulk liquid flow and the pore size at the evaporator surface.

Integration shows that the effective length should be used as

$$(L_E + 2\,L_A + L_C)/2.$$

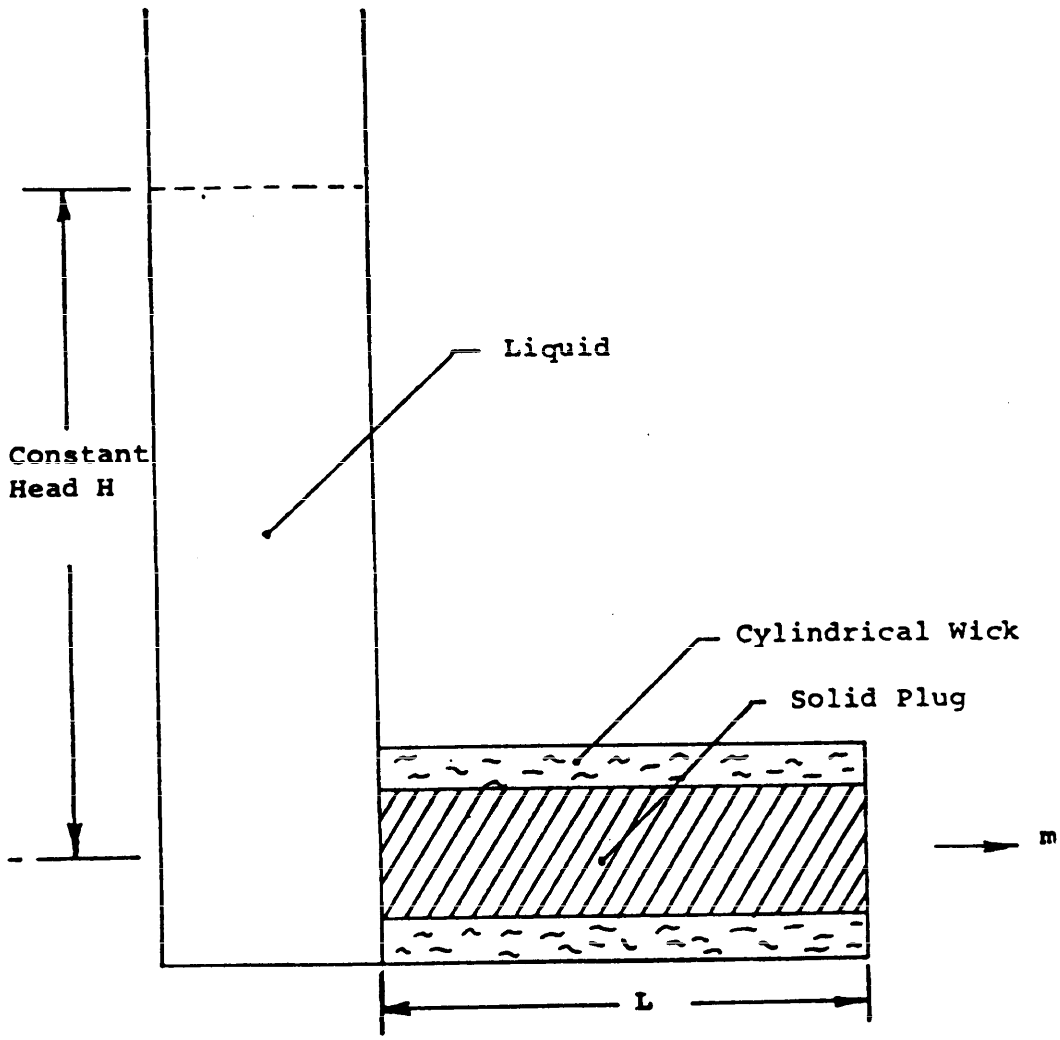


FIGURE 139. Measurement of Wick Permeability

Vapor pressure drop should be considered in determining the size of the vapor passage. It should be made small with respect to the liquid pressure drop in the wick. If the vapor velocity approaches Mach 1, heat pipe operating ceases.

12.2.3.5 Vapor pressure drop. Writing equation 12-7 in differential form for the vapor flow:

$$\frac{d\,p_v}{d\,x} = \frac{32\,\mu_v m}{\rho_v D^2\,A_v} \qquad (12\text{-}16)$$

Where:

D = hydraulic diameter of vapor passage

A_v = area of vapor passage

Subscript v indicates vapor parameter and integrating over the adiabatic length, assuming constant temperature:

$$\Delta p_v = \frac{32\,\mu_v m}{\rho_v D^2\,A_v} = \frac{32\,\mu_v\,L_A g}{\rho_v D^2\,A_v\,\lambda} \qquad (12\text{-}17)$$

12.2.4 Fluids, wicks, and wick parameters. Various structures are used for wicks. The simplest type is fiberglass tubing or rolled cloth. Rolled sheets of metal screening, usually 100 or 200 mesh, are rolled in varying degrees of tightness. Packed beds of metal grains mixed with a binder, painted on tube surfaces, and sintered, are very good. These structures may be made with a surface layer of small pores (for high capillarity) over a thicker layer of large pore size (for low flow resistance). Pore sizes, porosity, and the standard deviation of pore size, can be precisely controlled. Felted, sintered matts of metal fibers are available with any desired average pore size. Wicks are also formed by cutting straight, shallow grooves on the inside surface of the tube and covering with one layer of metal screen.

Ideally, the wick should provide an open passage for liquid flow, but should have a surface with minute pores on the vapor side to provide high pumping pressure. Arteries of rolled, fine mesh screen and internally grooved tubes covered with fine mesh screen are among the methods of accomplishing this. Structures of sintered metal beads of varying size are commercially produced in which the porosity is controllable, providing good longitudinal permeability and small pore size transversely.

In order to achieve high reliability and long life it is very important to thoroughly degas the heatpipe structure and pump to a high vacuum before charging with fluid, to prevent slow absorption of noncondensible gas during operation. Chemical and heat treatment of the wick has been found beneficial in some cases.

The amount of fluid is usually determined by measuring the quantity required to saturate the wick and adding five to ten percent excess. Many techniques for filling have been described in the literature, and elaborate apparatus has been used. The desired amount of fluid is evaporated into the prepared tube assembly under high vacuum after which the filler tube is sealed off permanently.

The important wick parameters are permeability, capillary pumping pressure, and liquid volume required for saturation. These are all functions of temperature. Permeability and capillarity are also very dependent on the surface chemistry of wick and fluid, and on the physical nature of the wick and envelope surfaces. Surface chemistry is an obscure and imperfectly developed science. It is impossible to predict accurately by calculation the performance of a particular wick and fluid combination. Empirical methods must therefore be used. The parameters should be measured with the actual wick structure and fluid to be used, and at the expected operating temperature.

As described in paragraph 12.2.3, K is determined by weighing the fluid flow during a given time under a constant static head, and assuming D'Arcy type flow. H_c is determined by direct observation, sometimes using an acid or alkaline tracer and narrow litmus paper strips. There is a considerable difference of opinion among researchers regarding the precision of these measurements, and a variety or actual techniques have been tried. Sometimes a vertical tube covered by a screen mesh is immersed in the liquid and moved up or down until the liquid is seen to reach the mesh. The screen must be heated to the desired operating temperature for accurate determination. Saturation liquid quantity can be measured by weighing wet and dry wicks, or by calculation from measured porosity and wick volume. (References 84, 85, and 86)

The liquid transport factor is a useful index of relative effectiveness of fluids for use in heat pipes. High values of N_f do not assure good performance, however, since the fluid wick combination is the determining factor.

A great deal of data has been published and the performance of prototype heat pipes reported. Application of the available data for estimating heat pipe performance is explained in paragraph 12.2.12, <u>Design and design examples</u>.

Chemical as well as physical compatibility of wick material and fluid is also important. It has been found that certain fluids and wick materials slowly react and evolve noncondensible gas which inhibits heat pipe action. Even when the combination shows high pumping power and permeability when new, the performance decreases seriously after weeks or months of service. Water and aluminum screen has been found unsatisfactory in this respect, whereas water and copper or bronze screen is a long lived combination.

Other considerations in the choice of fluid are the chemical stability, vapor pressure at operating temperature, freezing point, and compatibility with heat pipe materials. Toxicity is a minor factor since the tube is hermetically sealed. Liquid transport factor should be as large as feasible consistent with operating temperature and pressure compatibility, and good wicking performance in the wick design chosen.

12.2.5 Wick geometrics. The wick can have almost any desired configuration provided that it forms a continuous path for liquid flow from condenser to evaporator, and a separate vapor passage continuous throughout the pipe length, so arranged so that vapor enters and leaves it only through the wick surface. This is to prevent bulk flow of liquid into the vapor space, where it can not evaporate and thus inhibit heat transfer.

Figure 140 shows some structures and configurations that have been used. Figures a through g apply to conventional cylindrical heat pipes, but the design ideas can be applied to other forms as well. Figure g illustrates one commercial method of sealing the wick. Special tools are available for making this seal. Several of the designs, particularly the grooved tube, are used in high temperature, high performance heat pipes.

Figure 140 shows a heat pipe integral with an electronic package. The sintered grain wick of Figure 140 is a good choice for this design as it can be produced with small pores at the surface and high permeability.

Figure 140 shows a flat package filled with honeycomb wick structure. Deformation under pressure and vacuum is minimized. Vapor and fluid can move in all directions, so that heat source and sink areas may be chosen anywhere on the surface.

Heat pipes have been configured to fit integrally over black body and crystal chambers, cooking vessels, nuclear heated thermopiles, and many other forms. Wick materials are available which can be fabricated to complicated shapes, or which can be formed in place over almost any surface.

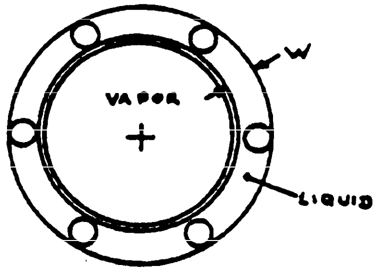
12.2.6 Heat flux limitations. During normal steady state operation of a heat pipe, the profile of temperature is as shown in Figure 141. For a conservatively designed heat pipe at rated load the drop in the adiabatic section (T_E-T_{C1}) will be from 1 to 3°C, and the overall drop (T_E-T_{C2}) 5 to 10°C. The entrance and exit thermal resistances are due chiefly to interfaces and are not determined by the heat pipe itself.

For steady state operation the wick must remain saturated with liquid throughout its entire length and the vapor should have free passage at a small pressure differential. If the flow of either liquid or vapor is impeded, pressure and temperatures will rise. If either flow is completely shut off at any point in the cycle, heat pipe action ceases.

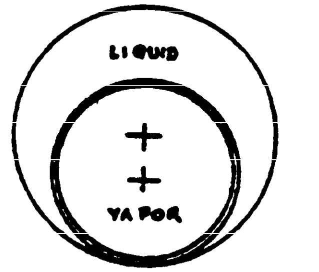
The fluid in excess of that which saturates the wick condenses and collects at the condenser end so that it masks part of the condenser end wick and thus reduces the maximum heat flux or increases the vapor temperature.

Condensation occurs within the wick structure by a simple heat conduction process. Normally the condensate saturates the wick and is carried away by capillary pumping, at the same rate that it is evaporated at the other end. So long as vapor can flow freely to the cold end, condensation does not limit heat pipe action. However, to maintain the required evaporation rate at a desirably low temperature differential, the thermal conductivity at the condenser end should be as low as allowable by good startup performance. (Reference to paragraph 12.2.11, Start up.)

Evaporation is a much more complex phenomenon, which can limit the peak heat flux. The evaporator must never become flooded with liquid, which would inhibit evaporation. The configuration must be such as to prevent flooding even when the pipe is inoperative. Evaporation is a surface phenomenon occurring at the liquid meniscus surface in the wick pores.



a. Concentric Annulus Rolled Screen with Space Wires

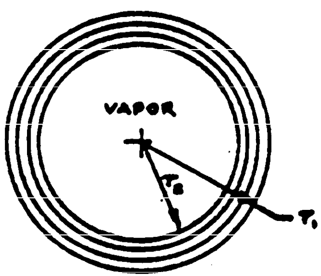


b. Crescent Annulus with Rolled Screen

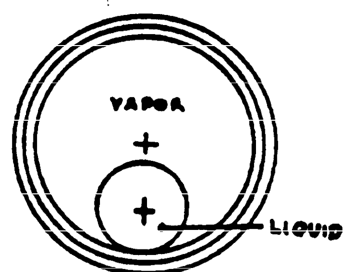
$$\Delta P_l = \frac{12}{\pi D_w^3} \psi$$

$$\Delta P_l = \frac{4.8}{\pi D_w^3} \psi$$

D = Equivalent dia of crescent
W = Avg. Distance, Tube-to-Screen



c. Rolled Screen or Fiberglass Contact with Tube



d. Arterial Type Rolled Screen Wick and Artery

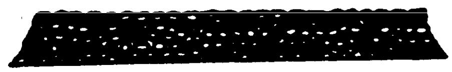
$$\Delta P_l = \frac{K_1}{\pi (r_1^2 - r_2^2) K} \psi \quad (8 < K < 20)$$

K_1 = d constant

$$\Delta P_l = \frac{8}{\pi a^4} \psi$$

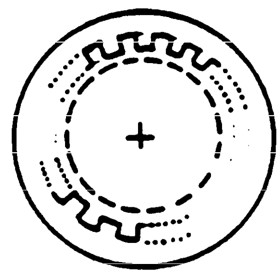
a = Equivalent Radius of Vapor Passage

FIGURE 140. Wick Geometries
(Sheet 1 of 3 sheets)

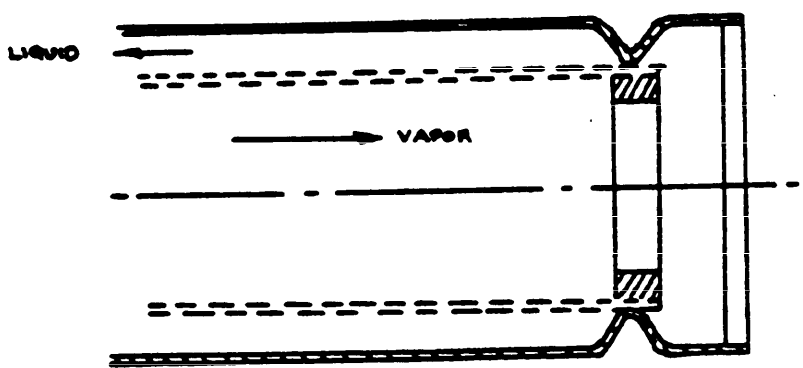


e. Composite-Packed Spheres or
Irregular Crains

$$\psi = \frac{\text{[illegible]}}{\rho_l} \times L$$

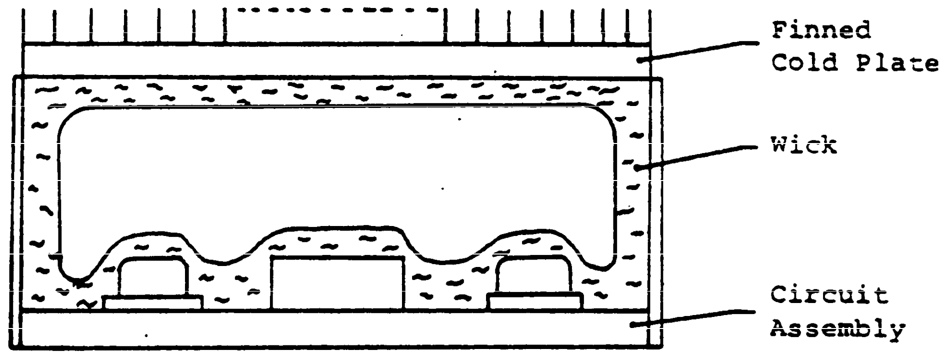


f. Grooved Tube with Screen

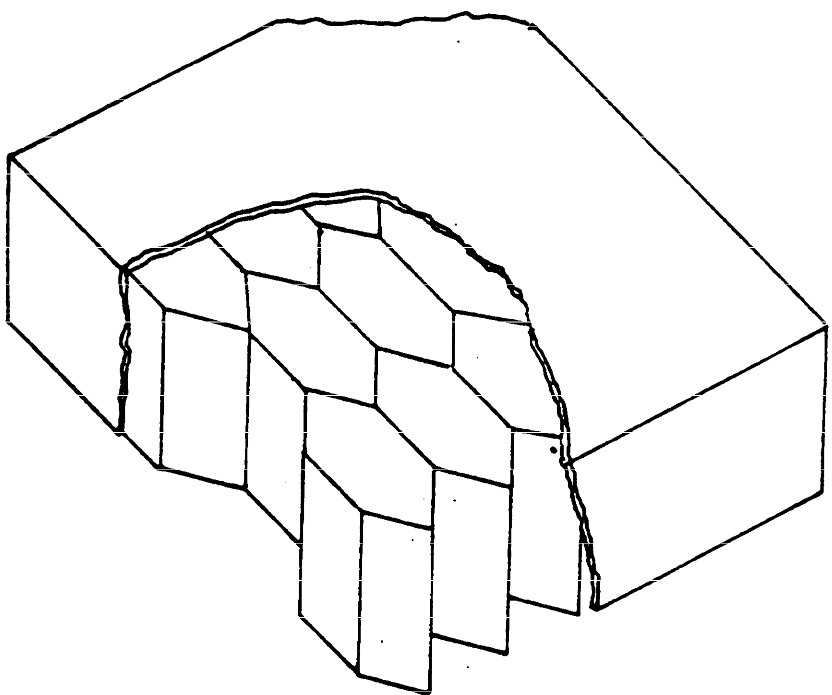


g. Method of Sealing Screen Wick by
Swaging Tube

FIGURE 140. Wick Geometries.
(Sheet 2 of 3 sheets)



h. Electronics Application



i. Sintered Screen Honeycomb

FIGURE 140. Wick Geometries
(Sheet 3 of 3 sheets)

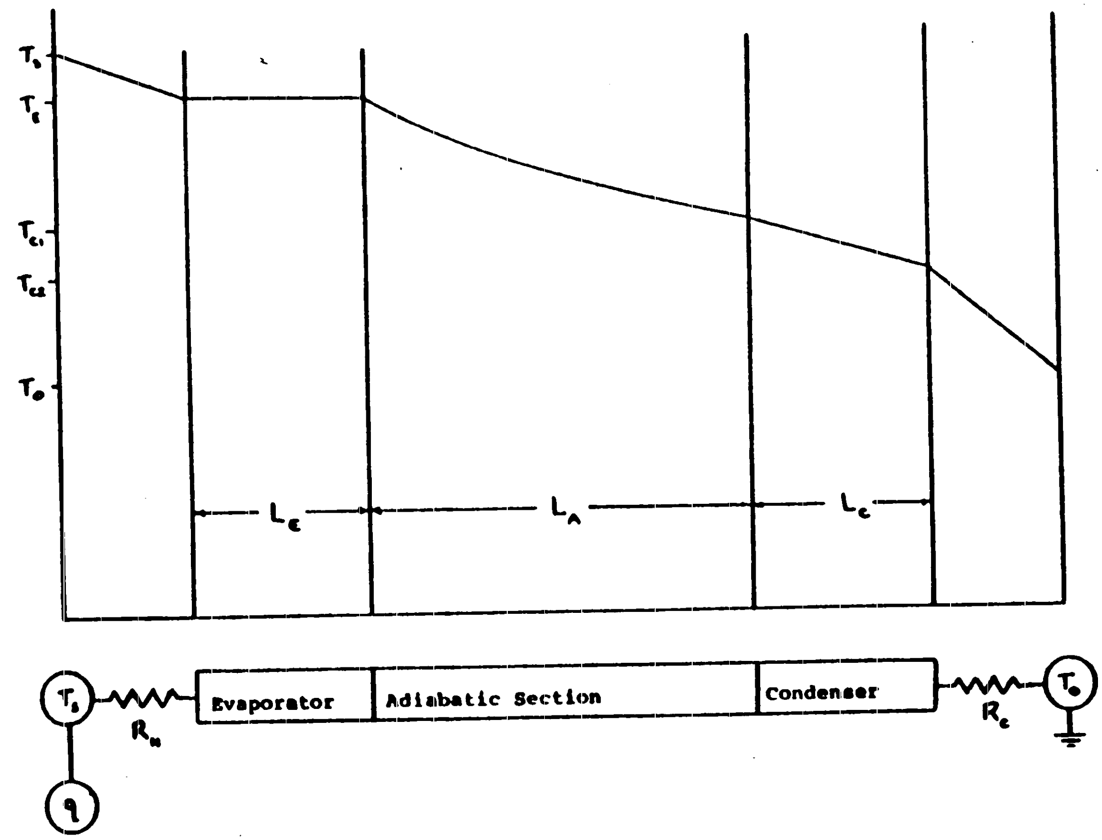


FIGURE 141. Temperature Gradient in Heat Pipe

At high heat flux nucleate boiling develops. If the bubbles which form inside the wick are too large to pass through the pores vapor lock results and heat pipe action may cease. Evaporation rate should be kept below the level of the nucleate boiling regime for conservative design.

Wick drying occurs if the liquid flow rate at any point is less than the evaporative rate. This may be due to any of several causes. When the wick dries, heat pipe action ceases and the effective thermal resistance becomes that of the shell and wick which is much larger than the normal effective value. The heat pipe then may fail due to overpressure if the heat flux is maintained. Normally after shut down due to wick drying, the wick will gradually absorb liquid. Resaturation may take several minutes, depending on the length of the wick and on the fluid properties. The various causes of wick drying are as follows:

12.2.6.1 Gravity. If the evaporator end is raised above the condenser end by an amount greater than the capillary height, liquid flow will cease. This height varies from 5 to 20 inches for different wicks and fluids. If this occurs momentarily from change in orientation as in a manuevering air-craft or rolling ship, heat pipe action may not be interrupted but severe transient temperature variations may occur. This limitation does not exist in zero gravity environment.

12.2.6.2 Sonic. The maximum heat flux is limited by vapor velocity. When this velocity reaches the speed of sound at the prevailing pressure and temperature of the vapor, a standing wave develops and mass flow at the condenser becomes zero.

$$\text{When } V = V_o = \sqrt{\gamma p/\rho} \tag{12-18}$$

The maximum transport rate of vapor is:

$$m_{max} = A\rho V_o = A\sqrt{\gamma\rho p} \tag{12-19}$$

where γ = ratio of c_p/c_v

For vapor Mach No. less than one, the heat flux is given by:

$$q = A_v \lambda \sqrt{\gamma\rho p_o} \quad \frac{M\sqrt{1 + M^2 (\gamma - 1)/2}}{1 + M^2 \ \gamma/2} \tag{12-20}$$

Where the stagnation values are:

$$T_o = T_v \left(1 + M^2 (\gamma - 1)/2\right) \tag{12-21}$$

$$p_o = p_v \left(1 + M^2 \ \gamma/2\right) \tag{12-22}$$

$$\rho_o = \rho_v \left(\frac{1 + M^2 \ \gamma/2}{1 + M^2 (\gamma - 1)/2}\right) \tag{12-23}$$

The sonic limit is usually not reached with water and other fluids suitable for operation at moderate temperatures. It is much more important in high temperature heat pipes using liquid metals. However, it is important to provide sufficient vapor passage area in order to avoid high vapor velocity in all heat pipe designs.

12.2.6.3 Entrainment. If the vapor velocity is in the high subsonic range, droplets of liquid may be stripped from the wick surface and carried to the condenser end, thus reducing liquid flow to the evaporator. If severe enough, this causes evaporator end dryout. It is reported that this effect is audible. The heat flux limit is proportional to the square root of ρ_V/λ. The vapor velocity increases with increasing heat flux. Also, if the vapor flow is throttled a shock wave may develop, causing sonic as well as entrainment failure.

12.2.6.4 Boiling. If the evaporation rate is high enough to produce vapor bubbles too large to pass through the pores of the wick, film boiling and vapor lock occur, and the evaporator temperature rises to a high value, thus drying the wick. In general, evaporation should occur in a heat pipe at a rate below the nucleate boiling threshold. However, nucleate boiling starts at a higher heat rate and temperature difference on surfaces with small pores than it does on conventional smooth surfaces of standard evaporators. The boiling limitation is not definite in precise quantitative terms.

12.2.6.5 Noncondensible gas. If the heat pipe fluid becomes contaminated by a gas having a critical temperature much higher than that of the condenser, such as nitrogen, hydrogen, etc., this gas masks the condenser surface. Heat pipe deterioration and failure results from evolution of such gas due to chemical action.

If a fixed or controllable mass of noncondensible gas is present, the peak heat flux is merely reduced. Operation is feasible in this case, as described in paragraph 12.2.9, Regulation and control.

12.2.6.6 Runaway startup. When a step function heat flux is applied to a heat pipe, time varying temperatures develop due to the thermal capacitances. If the step is sufficiently large the evaporator temperature rises high enough to dry the wick. This effect is particularly sensitive to the value of output thermal resistance, and too low a value causes temperature instability. Refer to paragraph 12.2.10, Startup, for details of this effect.

Figure 142 shows in general the variation in heat flux with vapor temperature due to four of the limiting factors. The ordinates and abscissas of these curves vary with pipe dimensions and fluid properties. The boiling limit curve has the same shape as that shown in heat transfer textbooks (References 6 and 9). The straight part at low temperature represents surface evaporation at the liquid meniscus in the pores. As temperature rises nucleate boiling causes bubble formation with the danger of vapor lock. After the nucleate boiling peak the vapor temperature rises rapidly to a possible dangerous level.

The sonic limit curve terminates when vapor velocity reaches Mach 1. Theoretically, the curve continues to a peak at about Mach 2 and then decreases, as shown.

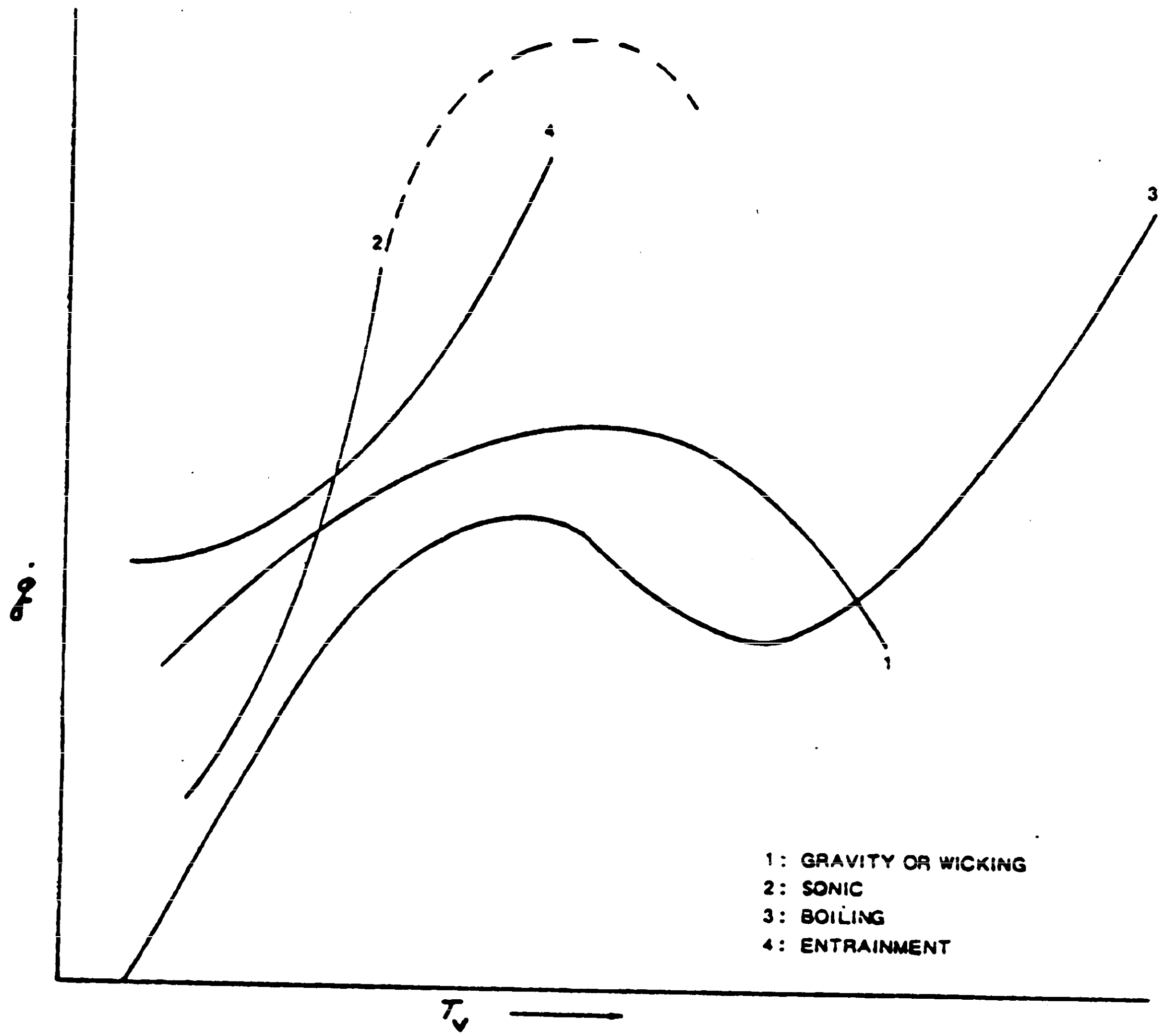


FIGURE 142. General Shape of Heat Pipe Limitations As A Function of Vapor Temperature

12.2.7 Reliability and safety. If properly constructed, processed, and operated, a heat pipe should have a long life. Gradual degradation and eventual failure results from corrosion and from evolution of noncondensible gases. Heat pipes must be thoroughly out-gassed before filling and sealing. Compatibility of fluid and pipe materials is very important for high reliability. Heat pipes have operated over 20,000 hours.

The operating temperature of a heat pipe is roughly a linear function of the heat flux. The vapor pressure is approximately linear with temperature. The tube wall must be designed structurally to withstand the highest desired heat flux, without deformation or rupture. An overload can cause catastrophic failure.

Glass tube walls are sealed off after filling by the usual glass-blower techniques. Metal tube walls are usually pinched off with a special tool which produces a cold weld. Over-pressure generally opens this pinch-off seal without causing damage to the environment, but the tube might behave like a rocket in severe, sudden overloads. Also, the liberation of hot mercury or sodium vapor might well have unpleasant consequences. However, these fluids would not normally be used for electronic cooling applications.

Heat pipes in the form of flat plates, kettles, etc., must be stayed internally to prevent bulging. The wall should be as thin as possible to keep thermal resistance low. Flat surfaces used in cooling electronic equipment must remain flat to maintain good thermal contact. The honeycomb wick structure illustrated in Figure 140 (sheet 3 of 3 sheets) resists pressure stresses well.

A heat pipe will not "blow up" if it is properly designed for the application involved. Catastrophic failure can occur only if the heat pipe is operated at an excessively high temperature. This condition can occur under two situation (a) if the heat input to the pipe is significantly greater than the quantity of heat the pipe is designed to transfer, or (b) if the heat sink or heat flow path, to which the output (condenser) end of the pipe is transferring heat, rises to an abnormally high temperature through failure of the heat sink. In the case of electronic equipment, the heat pipe should be designed to handle the maximum heat the electronics can possibly dissipate under the worst operating conditions.

12.2.8 Equivalent thermal circuit for heat pipe. A heat pipe can be adequately represented by the network of resistances and capacitances shown in Figure 143. The thermal resistances represent heat flow paths transversely through tube wall and wick, and axially through tube, wick, and fluid. A tee circuit is used for the transverse wick and pipe wall, and a Pi circuit for the adiabatic section. As a practical matter the capacitances labelled C_v are negligible. They are shown in Figure 143 because the adiabatic section actually does have a small thermal capacity.

To an electrical engineer, this circuit suggests either a low pass filter or a cable, according to whether he is thinking in terms of frequency or time analysis.

Most of these parameters can be estimated from a knowledge of dimensions and material characteristics. In general, these vary with temperature, but this can be neglected in an approximate solution. The parameter R_t is much larger than R_v, which is very small and, unfortunately, a function of the vapor temperature in the pipe. Although it is customary to consider vapor temperature constant throughout pipe length, there must obviously be some small gradient to provide driving pressure differential to the vapor. The heat pipe resistance may be approximated by integrating equation 12-24, the perfect gas approximation for the relation between temperature and pressure (Reference 79).

$$\frac{dT_v}{T_v} = \frac{1}{\rho_v \lambda} \, dp_v \tag{12-24}$$

Integrating;

$$\ln \frac{T_v(x)}{T_v(0)} = \frac{p_v(x) - p_v(0)}{\rho_v \lambda} \tag{12-25}$$

Substituting $p_v(x) - p_v(0)$ from equation 12-17:

$$\ln \frac{T_v(x)}{T_v(0)} = \frac{32^{\mu}v \; Lag}{\rho_v \; D^2 A_v \lambda} \times \frac{1}{\rho_v \lambda} = \frac{32^{\mu}v \; Lag}{\rho_v^{\;2} \; D^2 A_v \lambda^2} \tag{12-26}$$

Equation 12-26 describes the temperature through the adiabatic sections shown in Figure 141. By calculating the various temperature drops in the circuit of Figure 143, equation 12-27 for R_v is derived.

$$R_v = e^{cq} \left(T_o/q + R_E + R_o \right) - \left(T_o/q + R_E \right), \text{ where} \tag{12-27}$$

$c = 32\mu_v L/\rho_v^{\,2} D_v^{\,2} A_v \lambda_v^{\,2}$ depends on dimensions and vapor parameters.

A plot of this function starts at $R_v = R_o$ when $q = 0$ and increases with q, linearly at first and exponentially for large values of q. Over the linear part:

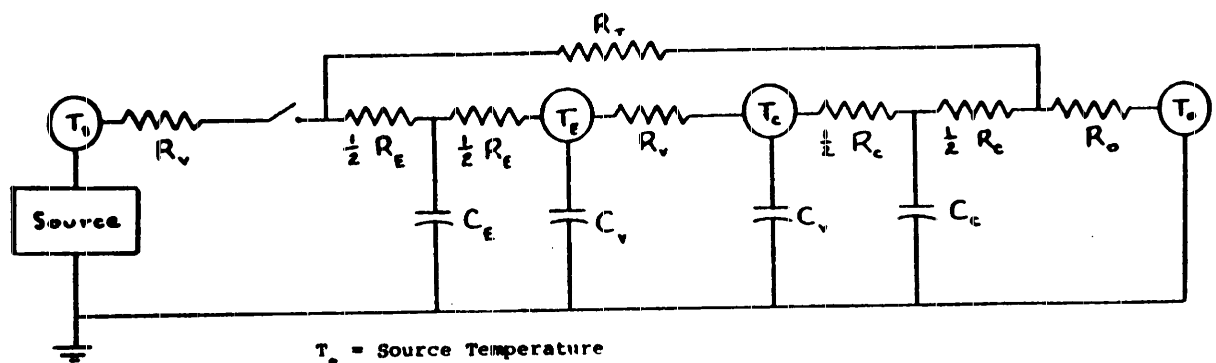
$$R_v = R_o + cT_o + \left(c \; (R_E + R_o) + c^2 T_o/2 \right) q \tag{12-28}$$

Heat pipes used in electronic equipment cooling systems generally operate in this linear range.

The heat source which is a time function in the general case may be either a heat flow rate (equivalent current source) or a temperature (equivalent voltage source). The source generally has a thermal resistance and capacity, as shown in Figure 144.

Heat pipes as elements of a cooling system can be represented by this equivalent thermal circuit, and the standard methods of circuit analysis and synthesis can be used for both steady state and transient problems.

Figure 145 shows a simplified thermal equivalent circuit for use in transient problems when a heat source is suddenly connected to a heat pipe. R_V is the sum of the wall and wick resistances and the equivalent resistance representing the vapor temperature drop which is R_V of Figure 143. C_H is the thermal capacitance of the interface structure.



T_s = Source Temperature
T_e = Evaporator Temperature
T_c = Condenser Temperature
T_O = Sink Temperature
R_s = Source-Heat Pipe Interface Resistance
R_O = Sink-Heat Pipe Interface Resistance
R_t = Longitudinal Shell and Wick Resistance
R_eC_e = T Circuit for Evaporator Wick
R_cC_c = T Circuit for Condenser Wick
R_vC_v = Pi Circuit for Vapor

FIGURE 143. Equivalent Thermal Circuit for Heat Pipe

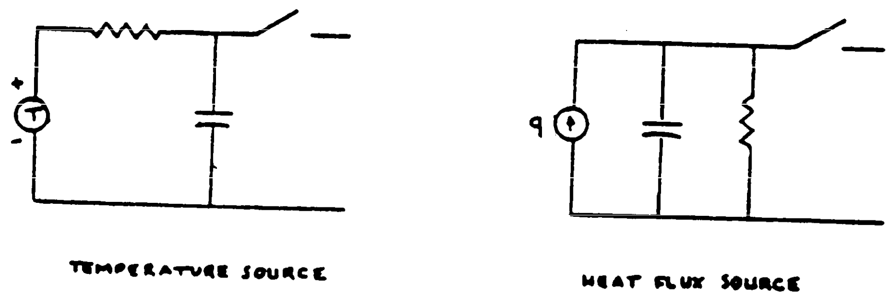


FIGURE 144. Finite Driving Circuits

The matrix equation of this network is given by equation 12-29:

$$\begin{bmatrix} \left(\frac{1}{R_V} + C_H s\right) & \left(-\frac{1}{R_V}\right) & 0 \\ \left(\frac{1}{R_V}\right) & -\left(\frac{1}{R_V} + \frac{1}{R_C} + C_V s\right) & \left(-\frac{1}{R_C}\right) \\ 0 & \left(-\frac{1}{R_C}\right) & \left(\frac{1}{R_C} + \frac{1}{R_O} + C_O s\right) \end{bmatrix} \begin{bmatrix} \theta_E \\ \theta_V \\ \theta_C \end{bmatrix} = \begin{bmatrix} q \\ 0 \\ 0 \end{bmatrix}$$

where $\theta_x = T_x - T_o$

(12-29)

The characteristic equation has three roots, whereas that of Figure 143 has four. Since the coefficients are all positive, all real roots are negative. There is the possibility of conjugate complex roots with positive coefficients, however, indicating instability. The Routh criterion can be used to check for unstable behavior.

The transient préformance of a heat pipe is important in startup, and in any type of control, either proportional or on-off. Heat pipe failure can occur if the parameter values permit instability.

The model shown in Figure 145 is suitable for use in analogue computer and complex computing board methods of systems analysis.

12.2.9 Regulation and control. Regulation implies a built in automatic capability of maintaining heat source temperature constant under variation of heat dissipation or sink temperature.. Control implies a capability to vary temperatures by external means.

12.2.9.1 Regulation. Automatic regulation is successfully accomplished by several manufacturers through the use of an inert noncondensible gas such as helium (Reference 99). Referring to Figure 146 a fixed mass of gas is contained in a reservoir inside the evaporator section, and fed through a small tube to the condenser section. As the temperature rises this gas expands, flows into the vapor space and pushes the heat transfer vapor back, thereby reducing the condenser area. The gas-vapor interface remains stable and moves back and forth as the temperature changes, the pressures of the vapor and the gas being substantially equal.

The total volume of inert gas is:

$$v = v_R + A \ (L_c - x) \tag{12-30}$$

Where:

v_R = gas reservoir volume

A = vapor space area

Maximum and minimum values are:

$$\begin{aligned} v_{max} &= v_R + AL_c \\ v_{min} &= v_R \\ v_{max} - v_{min} &= AL_c \\ v_{max} - v &= A\ x \end{aligned} \tag{12-31}$$

Assuming uniform temperatures of vapor, T_V, and tube shell T_W, over the condenser area, the heat flux is:

$$q = \pi D\ x\ h\ (T_v - T_w) \tag{12-32}$$

From equation 12-30 and 12-31:

$$\frac{x}{L_c} = \frac{v_{max} - v}{v_{max} - v_{min}} \tag{12-33}$$

Substituting for x in equation 12-32,

$$q = \pi D\ h(T_v - T_w)\ L_c\ \frac{v_{max} - v}{v_{max} - v_{min}} \tag{12-34}$$

Since there is a fixed quantity of inert gas present, its volume can be related to temperature by an appropriate equation of state if the pressure temperature relation is known. The vapor pressure of a fluid in contact with its liquid phase can be represented over a certain temperature range by equation 12-35:

$$p = K\ T\alpha \tag{12-35}$$

Where α is a large number, 8 to 10.

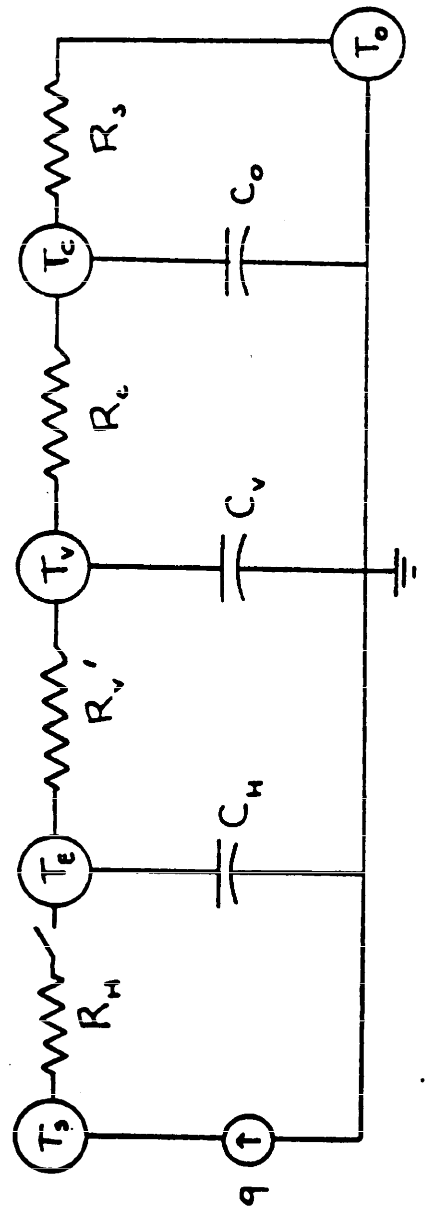


FIGURE 145. Simplified Thermal Circuit

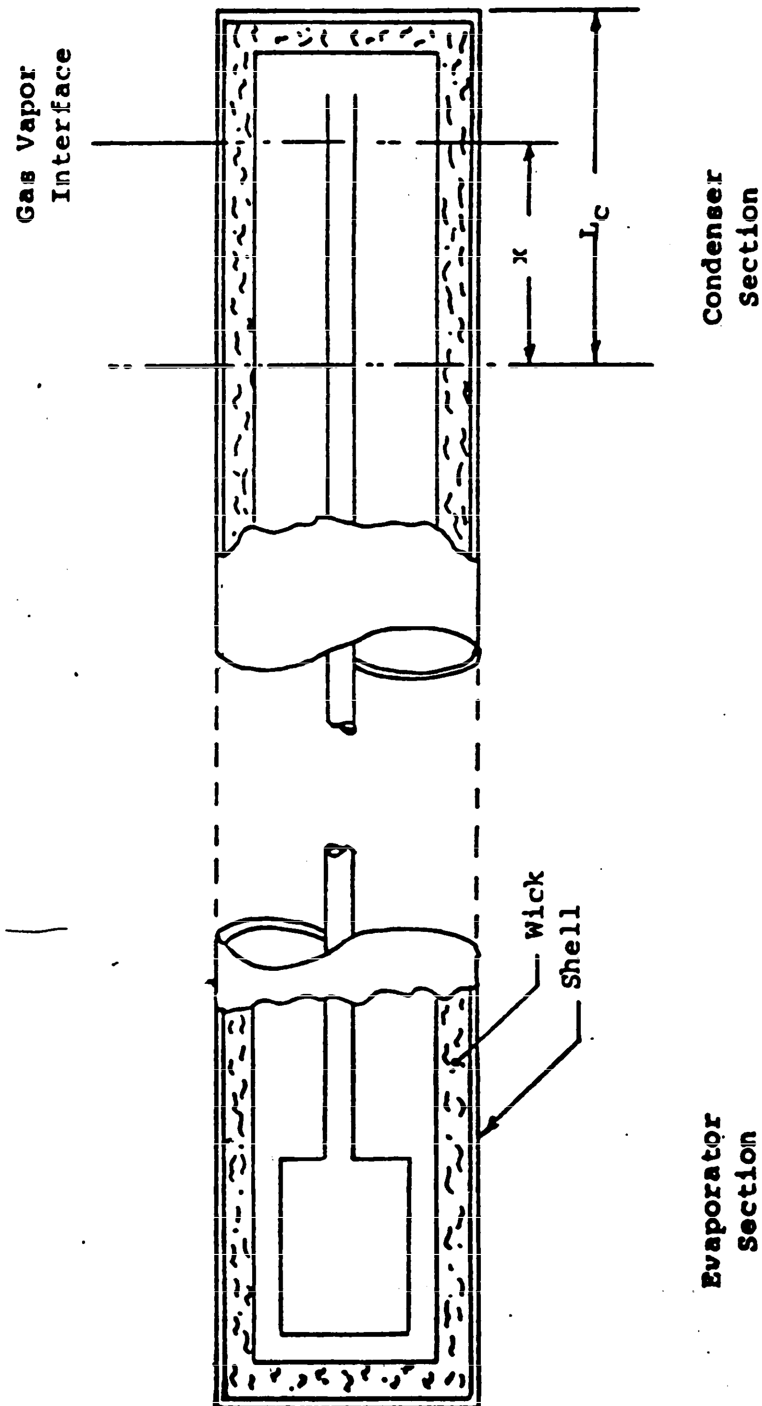


FIGURE 146. Inert Gas Regulation

As an example, for water vapor between 0 and 200 degrees C,

$$p = \left(\frac{T}{273}\right)^{11.4} \tag{12-36}$$

Where:
p = atmosphere
T = degrees Kelvin

The ideal gas law can be used as a first approximation for the inert gas volume. Substituting for the pressure in equation 12-37:

$$v = \frac{N\,R\,T}{p} = \frac{N\,R}{K\,T}\ (\alpha - 1) \tag{12-37}$$

Substituting in equation 12-34 and solving for the temperature difference;

$$T_v - T_w = \left(\frac{q}{\pi D\ L_c h}\right)\left(\frac{v_{max} - v_{min}}{v_{max} - \dfrac{N\ R}{K\ T_v^{\ \alpha - 1}}}\right) \tag{12-38}$$

Equation 12-38 can be written as a polynomial in T_v and differentiated with respect to q:

$$\frac{dT_v}{dq} = \frac{v_{max} - v_{min}}{\pi DL_c\ h\ \left(v_{max} + \alpha NR/KT_v^{\ \alpha + 1} + (\alpha - 1)\ T_w NR/KT_v^{\ \alpha}\right)} \tag{12-39}$$

Since α is a large number, the high powers of T_v cause the last two terms of the denominator to be very small and the rate of change of T_v with q is approximately:

$$\frac{dT_v}{dq} = (v_{max} - v_{min})/\pi v_{max}\ DL_c h \tag{12-40}$$

This compares with the derivative obtained from equation 12-32 when no inert gas is used.

$$\frac{dT_v}{dq} = 1/\pi DL_c h \tag{12-41}$$

The variation in T_v with q is reduced in the ratio of $(v_{max} - v_{min})/v_{max}$ by the presence of the inert gas.

It is claimed that, by proper choice of gas and dimensions, vapor temperature can be held constant within a few percent. However, referring to Figure 143 the interface resistances at the heat pipe terminals introduce two more temperature drops. Thus, if T_v remains constant with changing q and T_w, T_s does not remain constant. Also T_w is larger than T_0. The success of automatic regulation depends on having very small interface resistances.

12.2.9.2 Control. There are at least nine possibilities for varying the performance of a heat pipe by external means, viz:

1) variable capillary pumping pressure by varying effective pore size.
2) Varying evaporator surface area
3) varying condenser surface area
4) varying input thermal resistance
5) varying output thermal resistance
6) throttling the condensate flow
7) throttling the vapor flow
8) varying the angle of tilt
9) vibration and insonification

Method 1 has been suggested by several researchers but the literature does not show any reports of successful prototypes. At least one recent patent covers a method of continuously varying the effective pore size.

Method 2 could be done mechanically by sliding a masking surface over the evaporator area. However, a vacuum sealed actuating mechanism is required. There appear to be no reports of successful application.

Method 3 can be accomplished by the use of noncondensible gas described in paragraph 12.2.9.1, by using a bellows type gas reservoir the size of which would be varied. Only a static vacuum seal would be required for the gas feed tube. This method appears to be feasible.

Methods 4 and 5 are similar. It appears undesirable to tamper with the heat input end, since the thermal resistance between the heat source and the evaporator end of a heat pipe should be as low as possible in all types of application.

The output thermal resistance appears to offer a useful control means. It could be done by changing contact pressure or contact surface, or by introducing a varying amount of conductive gas or liquid. This is done with helium in experimental high temperature heat pipes, to prevent runaway startup. Very low output resistance can cause startup failure (see paragraph 12.2.10).

Output resistance can be changed by a mechanism outside of the evacuated tube, so vacuum seals on the actuating rods are not required. Bellows and magnetic actuators can be designed.

Methods 6 and 7 are similar. A recent patent application for a dual tube heat pipe includes means for compressing the wick to throttle liquid flow. Method 7 has been applied to flat or "cold plate" types of heat pipes, where a flexible vapor flow tube is pinched mechanically to shut off the flow. The structure is a dual tube affair like that described in the patent mentioned above.

As used, it is an on-off scheme, but could evidently be made proportional.

Method 8 might have a limited application. By changing the tilt angle the flow of liquid could be controlled within certain limits, but the configuration of the equipment to which the heat pipe is applied would severely limit its use.

Method 9 depends on the fact that evaporative heat flux is increased by vibration of the surfaces, particularly at ultrasonic frequencies. (References 64-70). One suggestion is to make the wick of porous magnetostrictive material. The evaporator heat flux could be increased by exciting such a transducer. The power supply would be an undesirable necessity, so that the scheme is probably feasible only for large capacity heat pipes.

Proportional or continuously variable control is preferable to on-off or relay control, since all relay type control schemes produce transients in the controlled variable. The usual methods of feedback control analysis apply. Thermal time constants can be long due to the heat capacities present, although the heat pipe itself has a short time constant. It is reported that high temperature sodium heat pipes attain operating temperatures in less than one minute, and water heat pipes in about one second. The startup time is a function of the design parameters, which determine the values of the R's and C's in the equivalent circuit.

A controllable heat pipe should be designed to start up successfully with a step function input equal to the maximum heat flux which it is expected to carry. Evaporator surface area and vapor passage dimensions may thus be oversized for steady state operation. Thus, the normal operating point of a controllable heat pipe must be well below the maximum capabilities of the heat pipe itself exclusive of the control system.

12.2.9.3 Control system performance. The performance of any control system can be explained best by following the sequence of events following an impressed change in variable. Consider a heat pipe operating steady state at constant heat input. Refer to the equivalent circuit shown in Figure 143.

If sink temperature increases at constant heat flux, the wick and wall temperatures at the condenser will rise rapidly. Since the vapor temperature does not change at first, the condensation rate decreases and liquid flow to the wick decreases. Eventually, the source temperature will rise increasing the vapor temperature, and, if the initial disturbance was not too severe, a new steady state condition will be reached with all temperatures higher than the initial values.

Now if the evaporator pore radius is decreased, the maximum pumping pressure increases, thus increasing the maximum heat transfer capability. Changing the condenser pore size has no effect on the maximum heat flux, and for lower heat fluxes it has relatively little effect. This is the principle of the first method of control described above.

If a throttling control scheme were used, either the liquid or vapor passage would be opened, the pressure drop in either passage would decrease, thus causing a larger pumping pressure which is equivalent to reducing pore size.

If a control scheme were used to reduce the input resistance, the wick and wall temperatures would not rise, therefore the source temperature would not be disturbed.

If a stepwise increase occurs in heat flux, evaporator temperature will increase rapidly, the evaporation rate, vapor temperature, and pressure will increase, and a new steady state will be reached at higher temperatures everywhere except at the sink, assuming the disturbance is not too severe. An output resistance control would tend to keep condenser temperature difference constant for the higher vapor temperature and thus increase the condensation rate to balance the increased evaporation. A throttling control would increase the mass flow rate. A pore size control would increase the pumping action to supply the increased demand for liquid to be evaporated. As before, a new steady state would be reached eventually with the same source and sink temperatures at the higher heat flux.

Different control schemes will have different response speeds and different temperature-time responses will occur. Only by mathematical analysis or experiment can the optimum control method be selected.

12.2.10 Startup. Startup is a transient condition initiated by connecting a heat source to the heat pipe. In electronic equipment cooling applications, a heat flux is usually applied to the system which is initially at the sink temperature.

If, before applying the heat load, the entire system is at a uniform temperature so high that the vapor pressure in the pipe is above 1 tor, (1 mm of mercury at 0°C or about 0.019 psi) no startup difficulty occurs and steady state temperature is reached in a few seconds. This is usually the case with intermediate temperature heat pipes using water, alcohol, etc., which are liquid at normal environmental temperature.

Figure 147 illustrates the progression with time of the temperature distributions (lengthwise) of a heat pipe when a heat flux is suddenly applied. (Reference 87) Four situations are described. The numbers on the curves, 0, 1, 2, etc., indicate successive times.

When the initial "cold" condenser temperature is above the critical temperature at which continuous vapor mass flow occurs, no startup difficulty occurs. This domain is called uniform startup.

When condenser temperature is below the critical value, frontal startup occurs. The evaporator heats but the condenser remains cold for a long time (measured in seconds). Depending on the heat pipe dimensions, fluid properties, and the rate at which heat input is increased, this may result in burnout. Heat input follows the sonic limit curve since the vapor velocity becomes transonic, and a shock wave forms at the condenser end. Since the fluid is evaporated to a nearly saturated condition, there is a possibility of drop wise condensation in the vapor space as the wave front moves along the pipe and the pressure decreases.

With frontal startup, condensation is delayed and continuing evaporation depletes the wick, with the possibility of complete drying and burnout.

If noncondensible gas is present in the tube, the advancing vapor wave front compresses this gas. A definite vapor gas interface evidently persists, and moves so as to gradually increase the active condenser area. Startup proceeds smoothly and rapidly if the properties and the quantity of gas are properly chosen.

Startup difficulties occur with high temperature heat pipes in which the fluid is frozen at normal environmental temperatures. These are frequently heated uniformly by induction to melt the fluid, before applying the heat load.

The actual mechanism of startup is complicated and a meaningful quantitative theoretical treatment is practically impossible. (Reference 87) If parameter values for the equivalent circuit of Figure 143 are known, a solution for vapor temperature can be obtained for a given heat input time function. The sonic limit can then be calculated and the danger of burnout be evaluated. Oscillatory solutions for temperature indicate a dangerously high heat input level.

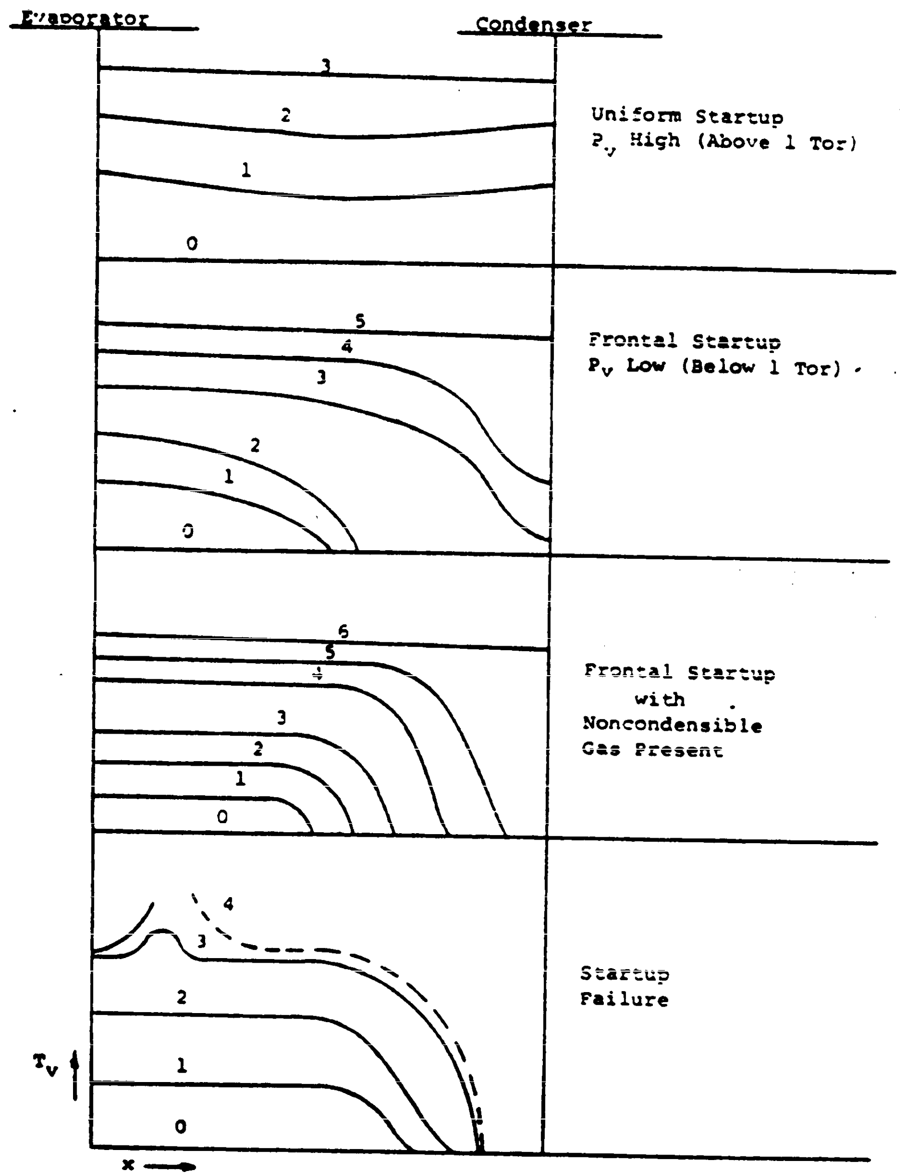


FIGURE 147. Vapor Temperature-Time Relation at Startup

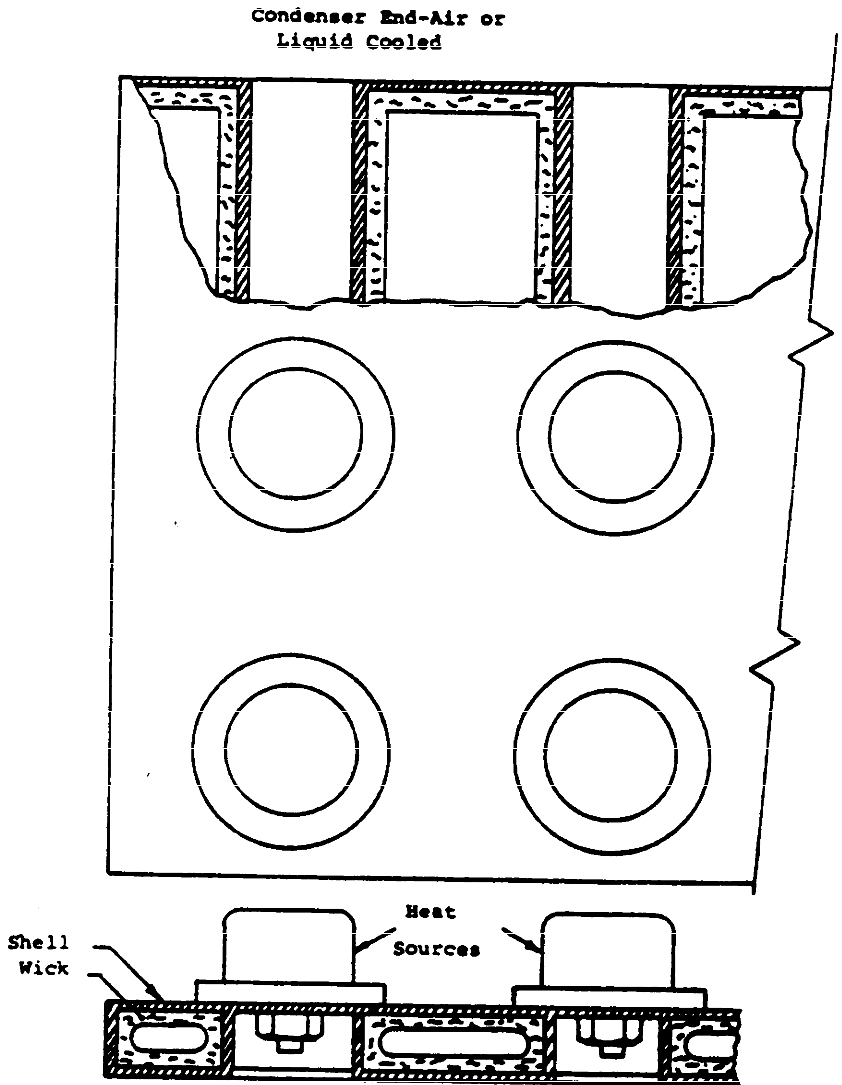


FIGURE 148. Heat Pipe Cooled Cold Plate

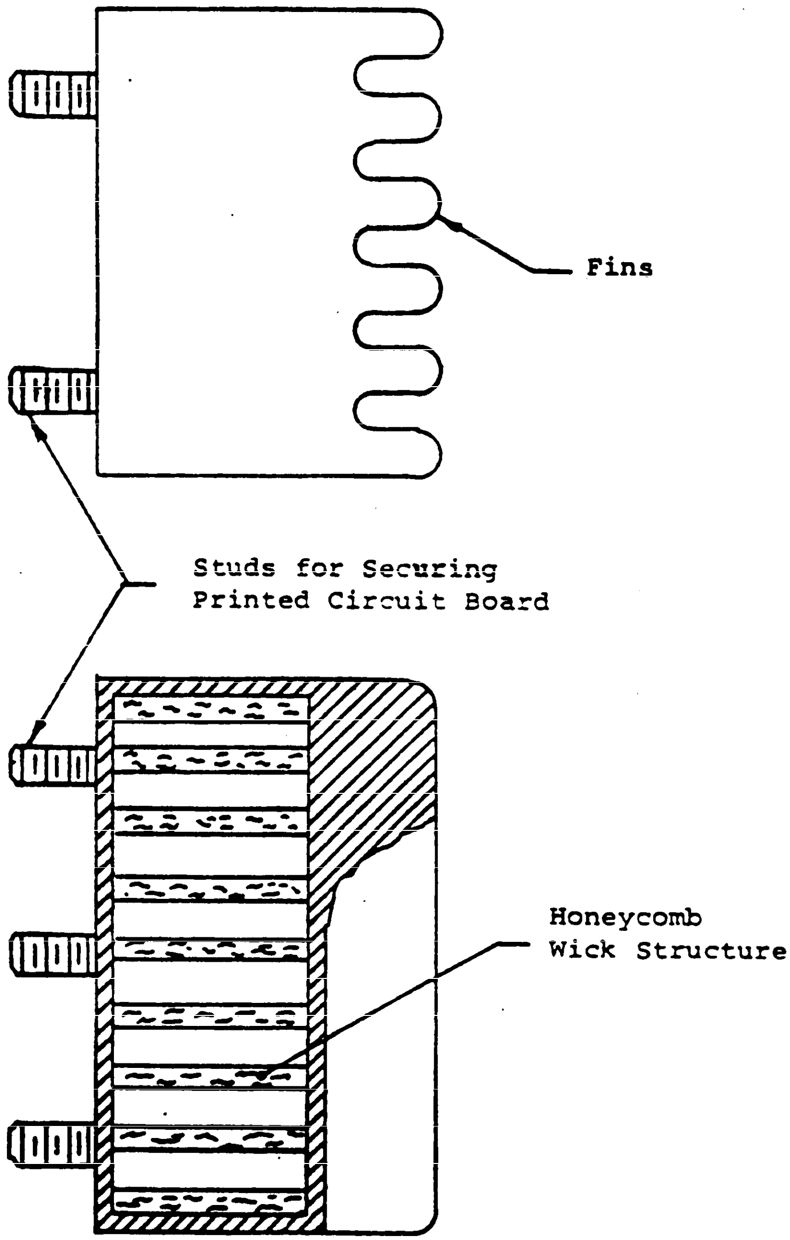


FIGURE 149. Heat Pipe Cold Plate for PCB

12.2.11 Applications as a cooling system element. Any properly designed heat pipe has a very low thermal resistance between the outside surfaces at the heat input and output. Referring to Figure 143 for steady state operation:

$$T_{in} - T_{out} = q(R_E + R_v + R_c) \tag{12-42}$$

This is the primary advantage of the heat pipe for cooling electronic equipment. A further advantage is the very uniform temperature over the evaporator surface. A heat pipe potentially has a spreading resistance superior to any solid thermal conductor, on a weight basis.

However, the heat pipe must be coupled to the heat source and sink through thermal resistances which must generally be of solid material. It is imperative to minimize these interface resistances.

So far as the input end is concerned, the ideal design is to apply the wick directly to the circuit elements which act as heat sources. This is generally impossible because the outgassing, sealing, and welding or brazing processes involved in heat pipe fabrication require temperatures high enough to damage or destroy most electronic parts. The heat pipe cold plate assembly must be so designed that electronic parts can be thermally bonded to it by low temperature techniques. Also, the assembly must have a low thermal resistance and high spreading ability.

The output end offers less difficulty since finned convective surfaces or liquid coolant passages can be made integral with the condenser end of the heat pipe.

A further difficulty occurs from the internal pressure. The heat pipe vapor is generally at a fairly low vacuum when cold and a high pressure at full load. The input and output structures must have thin sections for high conductivity and still must not deform excessively or rupture at operating pressure (Figures 148 and 149). Flat heat pipes must be internally stayed mechanically.

In devices such as klystrons and tubes having a high voltage anode a heat pipe can be constructed integral with the anode. Advantages are weight saving and electrical isolation from the coolant system.

It is generally advisable to use one large heat pipe rather than several small ones. In electronic cooling problems it is not usually necessary to convey the heat over distances of more than a few inches, so that one of the major heat pipe advantages is of little use. There are situations, however, (airborne equipment in pods and large liquid cooled cabinets for example) where heat pipe application could reduce weight, plumbing, and pumping requirements. The gravitational limitations must be considered. If the condenser is above the evaporator, the heat pipe transmitting ability is greatly increased.

Heat pipes used in cold plates should be of rectangular shape with a maximum area of the plate covered with wick. Round heat pipes fitted between or brazed to the plates require an increased weight of metal for the required low thermal resistance.

The use of heat pipes in electronic equipment introduces certain problems in maintenance and repair. A heat pipe is usually a rigid structure, although flexible ones are manufactured. There is no possibility of disconnecting a heat pipe by opening it. It must be separated as a unit, which requires bolted or press fit connections to any replaceable chassis, module, or part. This introduces additional thermal resistance. It appears that a thermally ideal module must be constructed as a replaceable module, comprising both the electronics and the condenser cooling device. Figures 148 and 149 illustrate methods of attaching replaceable power transistors and rectifiers, PCB's, etc. Bolted thermal joints are not ideal but if carefully made can be as low as 0.5°C watt/sq. in.

If heat pipes are applied to convey heat from several assemblies to a single heat sink, it is not good practice to use a branched heat pipe. Separate heat pipes leading to the heat exchanger should be used. Flexible heat pipes are useful with cabinet drawers. Branched and multiple heat pipes are commonly used in spacecraft, solar energy converters, thermo electric generators, etc., where the problems of disconnection and replacement do not exist. Check valves can be builut in to prevent loss of fluid if one pipe develops a leak.

Automatic temperature regulation by the use of noncondensible gas is an important advantage of heat pipes for cooling electronic equipment. It is claimed that MTBF can be increased by six or seven times if constant parts temperatures are maintained, even if that temperature is slightly above the recommended maximum. (Reference 100) If must be remembered, however, that this regulation method maintains constant vapor temperature. The interface temperature drops must be considered and they vary directly with heat flux.

The techniques of external control through feedback loops have not yet advanced beyond preliminary experiment and theory. Their reliability is completely unknown.

In summary, a heat pipe cooling system is only as good as the thermal coupling to the heat source. The tradeoffs between temperature uniformly, weight advantage and temperature regulation on one hand, and complexity, reliability, maintenance, and cost on the other, must be very carefully examined in a contemplated application.

12.2.12 Design examples.

Design Example 12-1: Approximate heat pipe capability. A heat pipe 1.0 in. I.D. with a fiberglass cloth wick 0.15 in. thick is 12 in. long. Evaporator and condenser section length are each 2 in. The fluid is water. The shell is designed for 100 psi working vapor pressure. Determine the maximum heat transfer capability.

Water vapor at 100 psi is at 325°F (163°C).

Values of permeability, liquid transport factor, and capillary heat are taken from Reference 77.

$$H_c = 10 \text{ in.}$$

$$N_l^c = 1.03 \times 10^{11} \text{ BTU/sq. ft.-hr. at } 325°\text{F}$$

$$= 1.03 \times 10^{11} \times 0.293/144$$

$$= 2.089 \times 10^{8} \text{ w/sq. in.}$$

$$K = 6.5 \times 10^{-12} \text{ sq. ft.}$$

$$= 9.36 \times 10^{-10} \text{ sq. in.}$$

at 325°F

$$\rho = 56 \text{ lbm/ft.}^3 = 0.0324 \text{ lbm/in.}^3$$

$$g = 32 \text{ ft./sec.}^2 = 384 \text{ in./sec.}^2$$

$$\sigma = 3.1 \times 10^{-3} \text{ lb./ft.} = 3.1 \times 10^{-3} \times 386.04/12$$

$$= 0.099727 \text{ lbm/sec.}^2$$

$$\rho g/\sigma = 0.866 \times 10^3/\text{in.}$$

$$A_w = \frac{\pi}{4}(1)^2 - \frac{\pi}{4}(1 - 2 \times 0.15)^2$$

$$= \frac{\pi}{4}(1 - 0.49) = 0.4006 \text{ sq. in.}$$

$$KA/L = (9.36 \times 10^{-10})(0.04006)/10$$

$$= 0.37492 \times 10^{-10} \text{ cu. in.}$$

$$q = (2.089 \times 10^8)(0.37492 \times 10^{-10})(0.866 \times 10^3)(10)$$

$= 68$ watts with the pipe horizontal <u>ANSWER</u>

$$q = (6.8)(10-12) \sin\phi$$

ANSWERS

Angle of tilt degrees	Heat transfer capabilities of heat pipe vs. attitude Condenser up	Condenser down
0	68 watts	68 watts
15	89 watts	47 watts
30	109 watts	27 watts
45	125 watts	10 watts

Design Example 12-2: Thermal resistance and capacitance for a heat pipe thermal circuit. A heat pipe designed for 40 watts is 3/4 inch in diameter and has a 2 inch long evaporator section and a 3 inch long condenser section . It is cooled by a radially finned heat sink of 3 inches O.D. which is brazed to the pipe. The pipe wall is 0.02 inch thick. The fiberglass wick is 0.06 inch thick. The heat pipe tube is stainless steel. The heat sink is aluminum and is 0.125 inch thick at the fin root. It has 25 fins 1 inch long. The heat input is a square aluminum block board for a press fit onto onto the heat pipe and weighing 0.33 lbs.

Estimate the R anc C values for an equivalent thermal circuit, as in Figure 145.

Input thermal resistance, R_H, assuming the block resistance is negligible.

The contact thermal resistance at the press fit is estimated at

$$R_H = 1/(0.75\pi)(2)(2000)(0.044)$$

$= 0.0024$°C/w Answer (1)

R_v = evaporator wall thermal resistance

$$R = \frac{0.02}{(0.75\pi)(2)(0.65)} = 0.0066°C/w$$

Evaporator wick thermal resistance

$$R = \frac{0.06}{(0.75\pi)(2)\ K} = \frac{0.01273}{K}$$

Since the saturated wick is a composite material, use for K the average of water and glass conductivities

$$K = \frac{0.0167 + 0.025}{2} = 0.02085$$

$$R = \frac{0.01273}{0.02085} = 0.61055$$

The thermal resistance representing evaporation is found from the slope of the boiling heat transfer curve (Reference 9) at low heat transfer values. This is approximately 5 x 10^{-6}°C/watt and is negligible.

Assuming 1° drop at 40 W in adiabatic section

$$R_v = \frac{1}{40} = 0.025°C/w$$

R_v = 0.0066 + 0.61055 + 0.025 = 0.642 <u>ANSWER (2)</u>

R = for condenser wick and wall is 2/3 that for evaporator.

R = (0.0066 + 0.61055)(2/3)
= 0.411°C/w

For the convective heat sink,

$$R = \frac{0.25}{(0.75\pi)(3)(5)} = 0.00707°C/w$$

R_c = 0.411 + .007 = 0.418°C/w <u>ANSWER (3)</u>

The convective fin surface area is (25)(1)(2)(3) = 150 sq. in.
Assume free convection heat transfer rate in air of 0.0035w/sq. in.-°C.

$$R_o = \frac{1}{(0.0035)(150)} = 1.9°C/w \quad \underline{\text{ANSWER (4)}}$$

Thermal capacitance = (mass)(specific heat), joules/°C.
For the input block C = (0.33)(.215)(1870)
= 132.67 joules/°C

For the wick, assuming it is all water, per inch length
(0.75π)(0.06)(60/1728)(1)(1870) = 9.179 joules/°C-in.
Evaporator wick 18.36 joules/°C
Condenser wick 27.54 joules/°C
C_H = 132.67 + 18 36 = 151 j/°C <u>ANSWER (5)</u>
C_V = 27.5 j/°C ANSWER (6)

For the finned heat sink

$$\begin{aligned} \text{Volume} &= (0.75\pi)(0.125)(3) = 0.883575 \text{ (body)} \\ &+ (25)(1)(3)(0.03) = \underline{\quad 2.25 \text{ (fins)}} \\ &\qquad\qquad 3.13 \text{ cu. in. (total)} \\ \text{mass} &= (3.13)(160/1728) \\ &= 0.2898 \text{ lbm.} \\ c_o &= (0.2898)(0.215)(1870) = 116.5 \text{ joules/°C } \underline{\text{ANSWER (7)}} \end{aligned}$$

Design Example 12-3: Transient performance. Substituting the parameters estimated in Design Example 12-2 into the approximate equivalent circuit of Figure 12-13, the determinant is:

$$\begin{bmatrix} (1.558 + 151s) & -1.558 & 0 \\ 1.558 & (2.55 + 27.5s) & -2.39 \\ 0 & 2.39 & -(2.918 + 116.5s) \end{bmatrix} = 0$$

The characteristic equation is:

$$4837665^3 + 62224s^2 + 1158s + 10.2 = 0$$

The Routh Criterion shows this to be stable. The roots of this equation are -0.101, -0.0225, and -0.0002. The system is close to instability. The output resistance R_0 would be reduced to 0.5 by forced air cooling, with greatly improved stability and faster response.

Design Example 12-4: Power transistor cooling. Four 2N4048 transistors are to be mounted on a rectangular finned aluminum plate. The average dissipation of each is 30 watts. Cooling air at 40°C is available. The maximum air velocity is limited to 8 ft./sec. No cooling air is permitted to touch the transistor cases. The maximum permitted junction temperature is 90°C. From manufacturer's data, R_{jc} = 0.5°C/W and the best insulated mounting has a resistance of 0.4°C/W. Evaluate comparatively a solid finned heat sink and a flat heat pipe application.

Figure 150 shows a feasible design for a forced air cooled mounting plate. This aluminum structure weighs 5 lbs. and measures 8 x 7 x 2.25 inches. The maximum temperature variation over the plate is designed to be 2°C, requiring at least 0.25 inch of stock at the fin root.

A flat heat pipe using a honeycomb wick design is proposed for comparison with the aluminum heat sink.

Design calculations for this flat heap pipe are as follows:

Choose sintered 200 mesh nickel screen formed into a honeycomb wick with the hexagonal cells 1 inch 0..D. Use water as the fluid combination, from Reference 77,

$$L_M = 9.2 \text{ in.}$$
$$K = 0.0834 \times 10^{-10} \text{ sq. ft.} = 12.07 \times 10^{-10} \text{ sq. in.}$$

The required temperature is:

$$90 - (30)(0.5 + 0.4) = 63°\text{C}$$

at this temperature for water,

$P_v = 4$ psi
$N_f = (10^{11})(0.293/144) = 2/035 \times 10^8$ w/sq. in.
$\rho = 62/1728 = 0.0359$ lbm/cu. in.
$\sigma = 4.4 \times 10^{-3}$ (386/12) = 0.1415 lbm/sec.^{2}
$g = (32.2)(12) = 386$ in./(sec. x sec.)

From equation 12-15, with L = 0.5 in.

$$q_{max} = (2.035 \times 10^8) \frac{(12.07 \times 10^{-10})(A_W)}{0.5} \frac{(0.0359)(386)}{0.1415} \quad (9.2)$$

$$= 442\ A_w$$

For 120 watts $A_W = 120/442 = 0.272$ sq. in.

The required wick area is 0.272 sq. in.

From Reference 77 the effective pore diameter is 50 microns = 197×10^{-5} in. The wire diameter in the mesh is 0.0015 in. The perimeter of each hexagonal cell is 3 in.

$A_w = n\,(3)(0.0015) = 0.0045\ n$

$n = \frac{0.272}{0.0045} = 60.5$ cells; say 64 cells

A flat heap pipe 8.5 x 8.5 in. can contain the 64 cells required, allowing space for transistor mounting bosses.

Figure 151 shows the proposed design. From calculations for the forced air cooled fin, h = 0.0135 watts/sq. in.-°C. The design provides 29 fins, each 1 x 8.5 in. Total heat transfer area is,

$(29)(2)(8.5) + (8.5)(8.5) = 565$ sq. in.
$q = 120 = (0.0135)(565)\Delta T$
$\Delta T = 15.7°C$

Estimated weight of the heat pipe cold plate is 3.0 pounds. It should be mounted either vertically or with the fins on the top surface to alleviate the adverse effect of gravity. The shell can be fabricated of 28 gauge sheet (0.0156 in.) since the sintered and brazed structure resists dishing from the low pressure.

The heat pipe application reduces weight from 5 to 3 pounds (40%), reduces volume from 140 to 108 cu. in. (23%) and provides a nearly constant temperature at all points on the mounting plate for a given heat load compared to the 2°C spread in the solid metal sink.

In this particular application the only advantages of the heat pipe are weight and space reduction. The solid heat sink of Figure 150 is otherwise a very satisfactory solution to the problem. To offset the advantages, the undoubtedly higher cost and the relatively unknown reliability of the heat pipe must be considered. Maintainability is about the same for each design.

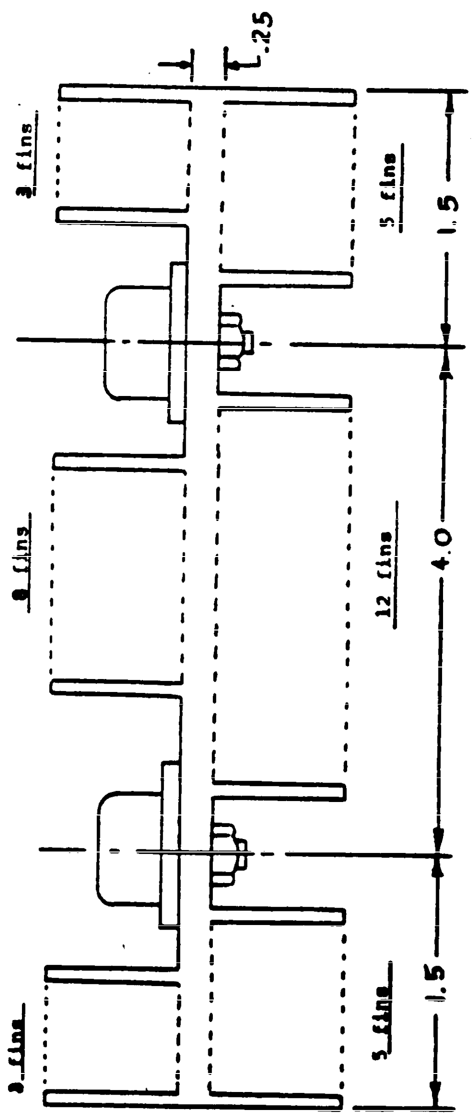


FIGURE 150. Finned Cold Plate for Design Example 12-4 Power Transistor Cooling

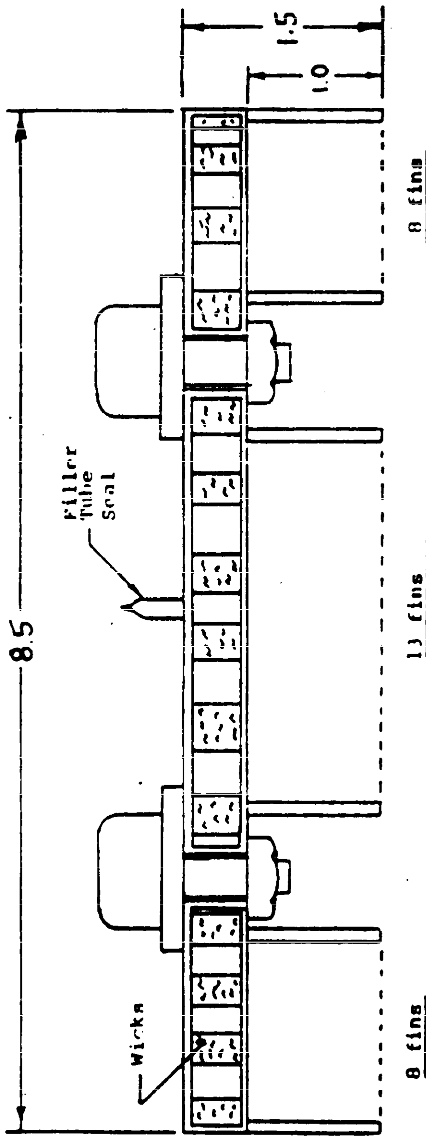


FIGURE 151. Heat Pipe Cold Plate for Design Example 12-4

Design Example 12-5: Regulation with noncondensible gas. A cylindrical heat pipe 1.0 in. I.D. using water for the heat transfer fluid transmits 50 watts with a vapor temperature of 110°C and a sink temperature of 30°C. The condenser section is 4.0 in. long.

Determine the gas reservoir volume required to hold vapor temperature variation to within 0.5°C/watt.

From equation 12-32

$$q = 50 = \pi(4)h(110\text{-}30)$$

$$h = 0.0497 \text{ w/sq. in.-°C}$$

From equation 12-41 for the unregulated heat pipe

$$dTv/dq = 1.6\text{°C/w}$$

$$V_{max} = V_R + \pi(1/2)^2(4) = 3.1416 + V_R$$

From equation 12-40

$$0.5 = \frac{3.1416}{(3.1416 + V_R)\ \pi\ (1)(4)(0.0497)}$$

$V_R = 6.87$ cu. in. ANSWER

12.3 Thermoelectric cooling.

12.3.1 General. Thermoelectric junctions or couples commonly provide a means of conversion of heat to electrical energy. This conversion is also reversible and thermoelectric junctions can be connected in series to form a pile which, then energized will act as a simple heat pump or refrigerator.

In brief, heat can be transferred between two points by converting thermal energy to electrical energy at the location to be cooled (cold junction), transporting the electrical energy (electron current flow) along a conductor, and reconverting the electrical energy to heat at the local heat sink (hot junction). See Figure 152. It is necessary to supply additional electrical energy to the system to accomplish this function. The external voltage is applied to oppose and overcome the voltage generated by the thermoelectric junctions so that the current is forced to flow "upstream." The additional energy is of a significant magnitude and the overall efficiency may be relatively low. The primary losses associated with thermoelectric cooling are those due to the conduction of heat from the hot junctions along the leads back into the cold junctions and the Joulean heating (I^2R losses) at the junctions.

Thermoelectric coolers provide refrigeration through the application of solid state technology to thermocouples. Consequently, there are no moving parts to wear out or be maintained and the reliability and life is the same as that of other solid state devices. Thermoelectric coolers, since they are refrigerators, can depress the temperature between a heat source and a heat sink so that the heat source is cooler than the sink. Thus, a "negative thermal" resistance can be obtained. Alternatively, thermoelectric coolers can be operated as positive thermal resistances (no refrigeration) with the heat source at a higher temperature than the heat sink. In this mode, the thermal efficiency (Coefficient of Performance; COP) is high and low values of thermal resistance can be achieved.

The major disadvantage of thermoelectric coolers is that the COP is low in the refrigeration mode. For example, a representative thermoelectric unit cooling a 10 watt heat source to 30°C below the heat sink temperature will require 50 watts of electrical power and the heat sink must now dissipate 60 watts instead of 10 watts. The COP is 10/50 or 0.2 compared to a COP of 3.5 for a Freon cycle refrigerator. Also, thermoelectric coolers are relatively heavy, cannot transport heat over distances greater than a few inches, and require special power supplies and heavy electrical conductors because of the very low electrical impedances involved.

Typically, the maximum temperature decrease obtainable under load with a single stage thermoelectric cooler is of the order of 60°C. However, several stages can be cascaded for greater temperature decreases, the usual limit being three stages. Cascaded thermoelectric coolers have provided minimum temperatures of approximately 150°K for small heat loads.

For example, 1000 watts input power was required to cool a 1/2 watt heat load to 150°K (Reference 109).

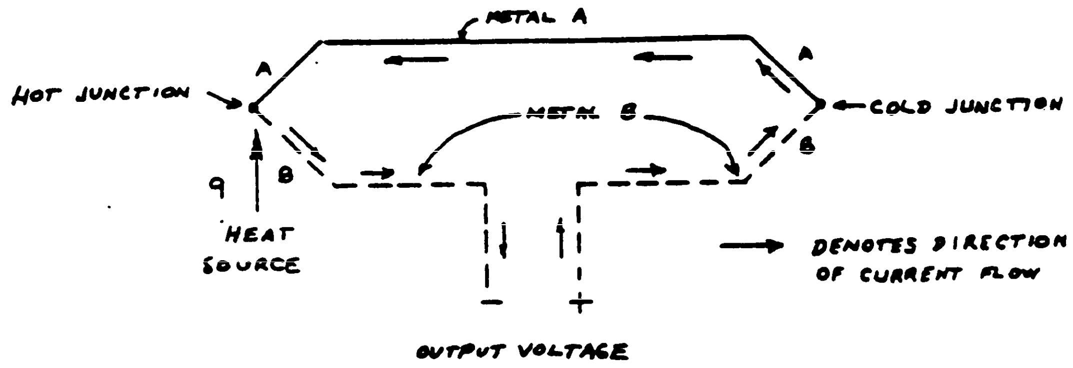
It is worth mentioning that the first thermoelectric cooler developed under DOD sponsorship was developed under Bu Ships sponsorship in 1953 (Reference 106). This unit used Bismuth-Telluride junctions.

12.3.2 Theory.

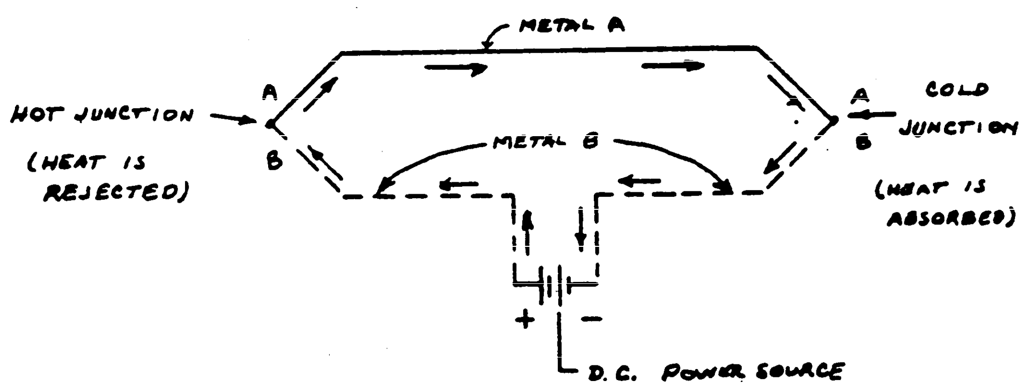
12.3.2.1 Fundamental thermal effects. The operation of a thermoelectric cooler is based on the following fundamental effects.

a. Seebeck effects. If wires of two different materials are joined at both ends and one junction heated while the other is cooled (a temperature difference between the junctions), a voltage will be produced across any opening in the circuit. This voltage increases with increasing temperature difference between the junctions: (This is the voltage obtained from thermoelectric generators and temperature measuring thermocouples).

$$V_{AB} = \alpha_{AB}(T_h - T_c) \qquad (12\text{-}43)$$



CONFIGURATION OF A SIMPLE THERMOELECTRIC GENERATOR



PELTIER COOLING ARRANGEMENT

FIGURE 152. Thermoelectric Junctions

Where:

V_{AB} = the output voltage

T_h = temperature of the hot junction

T_c = temperature of the cold junction

α = the Seebeck coefficient usually expressed in microvolts/°C

Since α varies with temperature,

$$dV_{AB} = \alpha AB(T)dT \qquad (12\text{-}44)$$

b. Peltier effect. If power is supplied to oppose and overcome the Seebeck voltage, then heat is absorbed at the cold junction and generated at the hot junction. The amount of heat transferred or pumped is proportional to the current flow.

$$q = I\ \Psi_{AB} \qquad (12\text{-}45)$$

Where:

I = the current

Ψ = the Peltier coefficient

Since Ψ varies with temperature

$$dq = \Psi_{AB}(T)dI \qquad (12\text{-}46)$$

c. Thompson effect. If within a length of conductor, dx, of material A, there is a temperature gradient $\partial T/\partial x$, with a current flow dI, a quantity of heat dq will be generated.

$$dq = \tau_A(T)dI\ \frac{\partial T}{\partial x}\ dx \qquad (12\text{-}47)$$

τ_A = the Thompson coefficient for material A

12.3.2.2 Heat pump theory. The construction of a typical single couple of a thermoelectric cooler is shown in Figure 153 (Reference 107). A n-type (doped) and a p-type (doped) slug of thermoelectric material are bonded to conductive metal straps. If multiple couples are used they are usually connected thermally in parallel and electrically in series to increase the electrical impedance.

To minimize heat leakage from the hot side to the cold side, a low thermal conductivity, k, is desirable. To minimize Joulean heating (I^2R losses) a low electrical resistivity, ρ, is desirable. Most materials do not match this requirement for a wide deviation from the Weidemann-Franz relationship. (Equation 8-16, pg. 54). That is, good electrical

For the finned heat sink

$$\text{Volume} = (0.75\pi)(0.125)(3) = 0.883575 \text{ (body)}$$
$$+ (25)(1)(3)(0.03) = \underline{2.25 \text{ (fins)}}$$
$$3.13 \text{ cu. in. (total)}$$

$$\text{mass} = (3.13)(160/1728)$$
$$= 0.2898 \text{ lbm.}$$
$$c_o = (0.2898)(0.215)(1870) = 116.5 \text{ joules/°C}$$ ANSWER (7)

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$$\begin{bmatrix} (1.558 + 151s) & -1.558 & 0 \\ 1.558 & (2.55 + 27.5s) & -2.39 \\ 0 & 2.39 & -(2.918 + 116.5s) \end{bmatrix} = 0$$

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compressor inlet is a mixture of vapor and liquid droplets, or mist.

The absorption refrigeration system utilizes a relatively complicated layout of devices in place of the compressor of a vapor cycle system. Figure 158 is a schematic of the basic elements of an absorption refrigeration system. Ammonia is the refrigerant used. The condenser, expansion valve, and evaporator are similar in function to their equivalents in the vapor cycle system. The condenser removes heat from the refrigerant by use of the ultimate sink coolant, extracting any vapor superheat plus the latent heat of condensation. Some slight liquid subcooling (cooling below the vaporization temperature) might also occur in the condenser. The subcooled liquid expands through the expansion valve to some temperature level below T_H, the heat source temperature. The evaporator is a heat exchanger which transfers heat from the electronic system to the refrigerant, resulting in ammonia vapor at the evaporator exit. In the absorber, the hot ammonia vapor is absorbed by water (ammonia is soluble in water). This absorption is an exothermic process, so that cooling of the absorber is required to carry off the heat of reaction. The stronger ammonia solution rises to the top of the absorber because of its lower density, and is pumped through a heat exchanger to the generator. An external source of heat in the generator heats the strong ammonia solution, liberating ammonia vapor, together with a small amount of steam. The steam is extracted by condensation in the rectifier, and is returned to the generator. The ammonia vapor from the rectifier re-enters the condenser, and repeats the cycle. The liberation of ammonia vapor in the generator results in a weak ammonia solution, which is circulated by way of the heat exchanger back to the absorber. The heat exchanger cools the weak solution on its return to the absorber, and heats the strong solution on the way to the generator, resulting in less external heat required in the generator and less cooling water required in the absorber.

Since the energy required by the pump is very small in comparison to that required by the compressor of a vapor cycle refrigeration system, very little mechanical shaft work is used to operate the absorption system. It is in fact possible to eliminate even the pump shaft work as in the Servel refrigerator. The Platen-Munters system (not illustrated) uses an inert gas to regulate system pressure and operates without a pump. The absorption system thus may be considered a method of replacing the external shaft work to the compressor in a vapor cycle compression system with an external heat source in the generator.

In general, absorption refrigeration systems as compared to compression refrigeration systems are more costly to install and operate, and require more attention for proper regulation. In addition, the toxicity of ammonia vapor is a deterent to their use. Their use may be considered insome special applications where mechanical shaft energy is not available and/or a generator heat source is present. Refrigeration experts should be consulted before committing to this type system.

12.5 Change of phase cooling. Material is generally recognized to have three states, or phase, of existence: solid, liquid, and gas. In the gaseous state, a material may be termed either a vapor or a pure gas, depending on whether or not the forces of attraction between

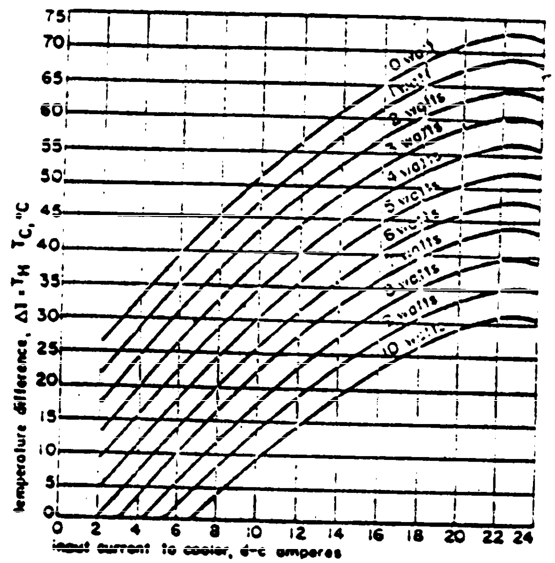


FIGURE 154. Temperature Difference vs. Input Current for Various Heat Pumping Loads

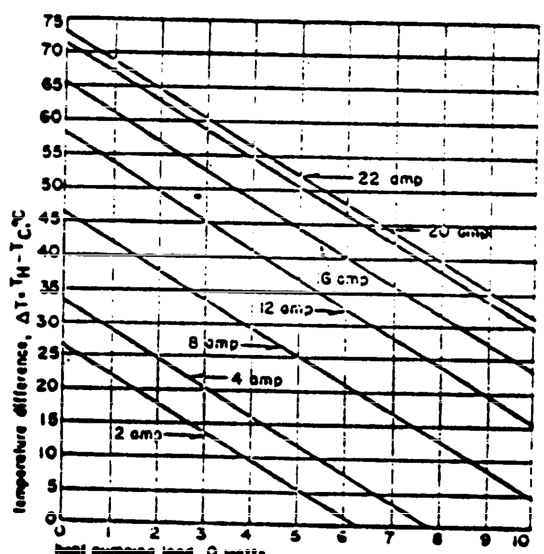


FIGURE 155. Temperature Difference vs. Heat Pumping Load for Various Currents

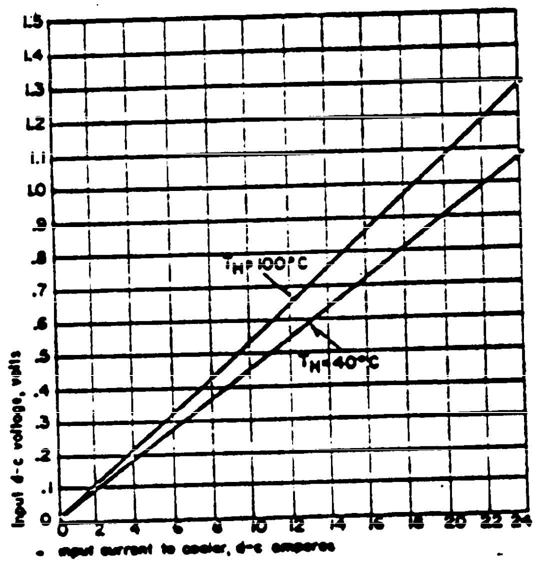


FIGURE 156. Input Voltage vs. Input Current for 5 Watt Heat Pump Load

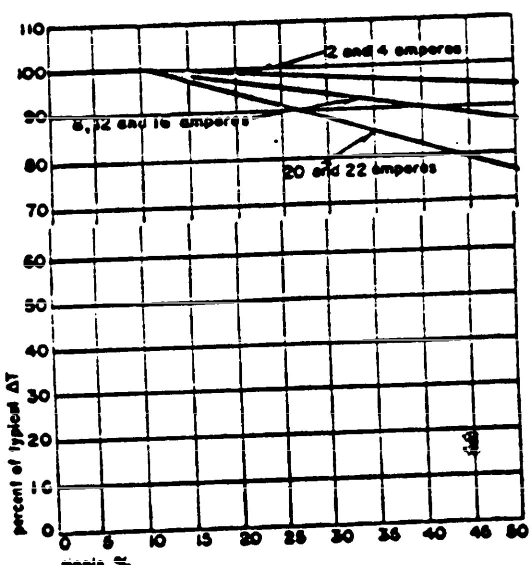


FIGURE 157. Percent of Typical ΔT vs. Percent of Power Supply Ripple

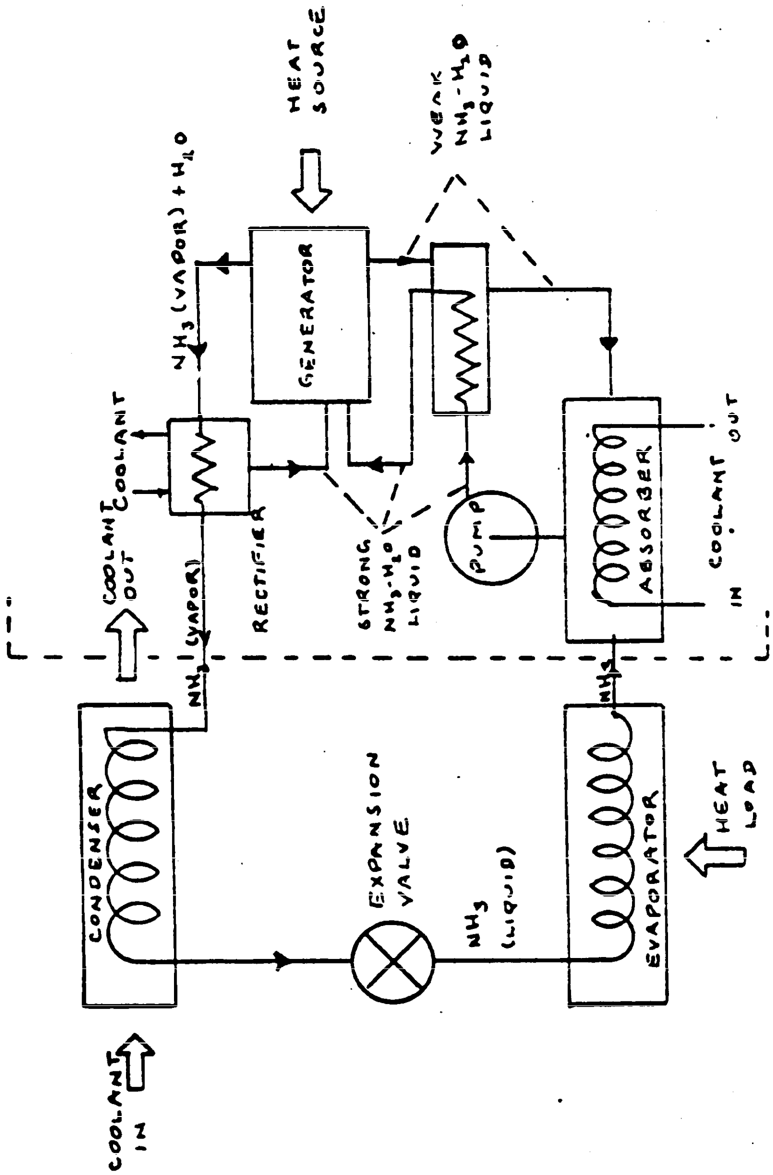


FIGURE 158. Absorption Refrigeration System Schematic

molecules are large enough to affect significantly the behavior of the molecules. Empirically, the criterion for distinction between a gas and a vapor is the degree of adherance to the ideal gas laws of thermodynamics.

In any one of three phases of existence, the temperature increase (or decrease) of the substance is in direct proportion to the amount of heat transferred into (or out of) the substance. This proportionally, on a unit mass basis, is relatively constant, and has been designated specific heat: the amount of heat required to change the temperature of one pound mass of the substance one degree.

When a substance changes from one phase to another, heat is transferred into (or out of) the substance with no significant change in temperature, the only noticeable effect being the progressive change of more of the substance from one phase to another. When the entire mass of the substance has completed the change of phase, the proportionally between transferred heat and temperature change will resume (although not necessarily with the same proportionality constant).

The phase change from liquid to gas (boiling) or gas to liquid (condensation) has been dealt with in chapter 7. This subsection will concern itself with phase changes from solid to liquid (melting), liquid to solid (solidification or fusion) and solid to gas (sublimation).

In each of the above processes, the substance is a solid at some point in the cycle. Because it is impractical to transport a solid in a confined volume, these processes are almost never used in a closed loop, steady state system. To illustrate further, the liquid-gas phase change can readily be accommodated in a physical system. Thus, a liquid can be boiled in an evaporator; the ensuing gas can be transported through ductwork, with a fan if required; the gas can be condensed in the condenser; and the liquid can be returned to the evaporator through piping, with a pump, if required. Mass transport of either phase is readily achieved, enabling a relatively simple heat transfer loop. When the substance exists in the solid phase at any point in the cycle, however, mass transport becomes virtually impossible, and the closed circulation loop is destroyed.

Solid-to-liquid phase systems may be used to advantage in non-steady state, or transient conditions. In transient heat transfer designs, the source heat dissipation and/or the environmental temperature vary with time, so that the temperature at any given point in the system is also time dependent. For transient applications, solid-liquid change of phase systems are preferable to liquid-gas change of phase systems because the volumetric change is much less for a given amount of heat transferred. The amount of heat absorbed by a change of phase material may be determined by:

$$q = m\lambda/t \tag{12-52}$$

Where:

q = phase change heat transfer, watts $\quad$ t = time, hr.
m = mass, lbm.
λ = latent heat, watt-hr./lbm.

Equation 12-52 contains no temperature terms, since the temperature during phase change is constant, and is determined by the material being used. Pressure effects on phase change temperature are usually negligible for solid-liquid systems.

λ in equation 12-52 is the latent heat of the substance. For solid-liquid systems, this is the heat of liquification or of fusion (equal terms); for solid-gas systems, this is the heat of sublimation.

Solid-liquid phase change systems may find application where a high dissipation exists for a relatively short time or where the environmental temperature reaches a high value for a short period.

Solid-liquid phase change systems must be carefully designed to accommodate the volume change associated with the phase change. This generally means that in the solid phase, the substance will not fill its enclosure. The material must be distributed such that the phase change interface progresses properly through the material. For example, localized high heat dissipation sources adjacent to the solid phase should be avoided, since a small pocket of liquid will form, surrounded by the solid phase. If not relieved, excessive pressures can be built up within such pockets. Cellular structures are generally preferred for this type application.

Some substances pass directly from the solid to the gaseous phase, with no intermediate liquid state. Perhaps the most common of these substances is solid carbondioxide, commonly called "dry ice." Other substances sublimate under certain conditoins. For example, a piece of ice exposed to dry air at temperatures below 0°C (32°F) will sublimate, and given enough time, will pass entirely into the air as water vapor, or steam. Because of the near-impossibility of collecting and re-converting the sublimed vapor into a solid, sublimation must be considered an expendable process. The process is not used for normal cooling of electronic systems. However, because of its convenience, and the lack of subsequent messy liquids, "dry ice" is sometimes used as a spot cooler in developmental tests, or along with electric heaters in an insulated box, to provide a crude test chamber for low temperature tests. The sublimation temperature of "dry ice" at atmospheric pressure is -78°C (-109°F). The latent heat of sublimation is 23.9 watt-hr./lb.

12.6 Vortex tubes. A vortex tube consists of a precision-configured "tee" type chamber into which air is forced tangentially. One tube is connected to each end of the tee. The longest of the two has a throttling valve on the opposite end from the tee and is known as the hot tube while the shorter tube, which is known as the cold tube, is reduced to a diaphragm with a small aperature near the tee end (see Figure 159). Compressed air passing tangentailly into the tee forms a vortex in the tee. The lighter air with less thermal energy flows to the center of the vortex and passes through the aperature in the center of the cold tube to be discharged out the open end of the tube. The denser air with more thermal energy flows down the hot tube in a spiral and out through the throttling valve which is used to control the operating temperatures.

The efficiency of the tube is not affected by any particular shape of the container and tubes, provided the whole is of rotational symmetry. One model of .25 inch copper tubing had a warm tube 12 in. long and a cold tube 6 in. long. Optimum results were obtained with the diaphragm as near the nozzle as possible. Since the shape of the aperature of the diaphragm is not significant, a circular opening is used.

Varying ratios of hot and cold air can be obtained by adjusting the throttling valve at the warm air end. The tube is capable of simultaneous temperatures of plus 106°F, and minus 56°F. At "hot" adjustment, it can produce up to 350°F. While the vortex tube provides 15 to 20 times greater cooling than the ordinary laboratory method (expansion of gas through a nozzle under the Joule-Thomson principle), it has a refrigerating efficiency of only 20%.

The general design of the vortex tube is fairly simple but there are many variables which control the temperatures of the tubes.

1. External pressure and temperature of the atmosphere or chamber into which the air is discharged.

2. Tube dimensions: diameters of hot and cold tubes, diameter of nozzle, diameter of orifice in diaphragm.

3. Pressure of air before expansion by nozzle (nozzle pressure) and rate of flow of air from nozzle.

4. The mass of cold air through the cold air tube varies with throttle pressure and setting.

Because the vortex tube has no moving parts, it is particularly adapted to high stagnation temperatures, especially when ceramic materials are used. The vortex tube will maintain its efficiency at high pressure ratios. This may make it applicable for cooling at very high Mach numbers. The vortex tube may find application in installations where very small flow rates are required. At larger flow rates, an expansion turbine in a simple air cycle system is far more efficient.

Only about 1/3 of the air entering the vortex tube reaches the low temperature level to be available for cooling purposes. According to tests, the tube length can be chosen at about 250 times the diameter of the nozzle. Thus, for additional cooling, a tube three inches in diameter would have a length of 14 to 18 feet.

Standard sized commercial models are available.

Reference 110 describes a patented "energy separation" device which in operation is somewhat similar to a vortex tube in that it separates a high pressure gas stream into two streams of high and low temperature. For a given pressure ratio and flow rate, Q, it is considerably more efficient than the standard vortex tube. Its construction is considerably more complex, although, like the vortex tube, the device contains no moving parts. Figure 160, taken from reference 110, indicates the relative efficiency of the two devices in terms of COP (coefficient of performance), together with the COP of a standard gas refrigeration cycle.

12.7 Cryogenic cooling. Cryogenic cooling refers to heat transfer processes taking place at very low temperatures, generally less than the boiling point of methane (112°K, or -161°C). Cryogenic cooling os electronic systems is used for various purposes, some of which are:

1. Electronic devices that require cooling to cryogenic temperatures for proper functioning. For example, impurity-activated indium antimonide detectors for infrared systems normally operate at 1.2°K.

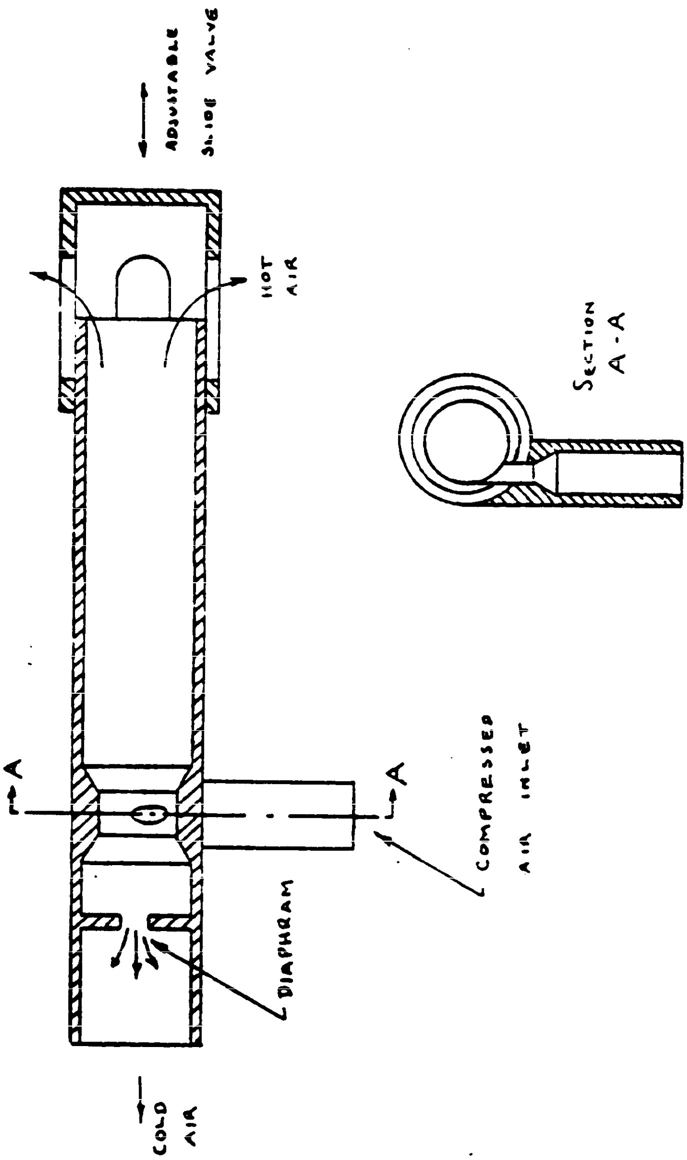
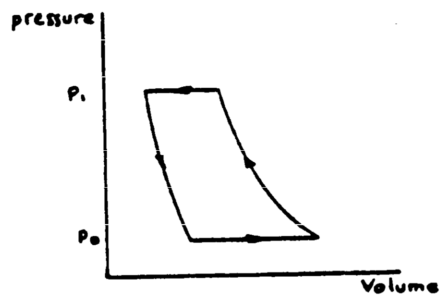


FIGURE 159. Typical Vortex Tube



STANDARD GAS REFRIGERATION CYCLE

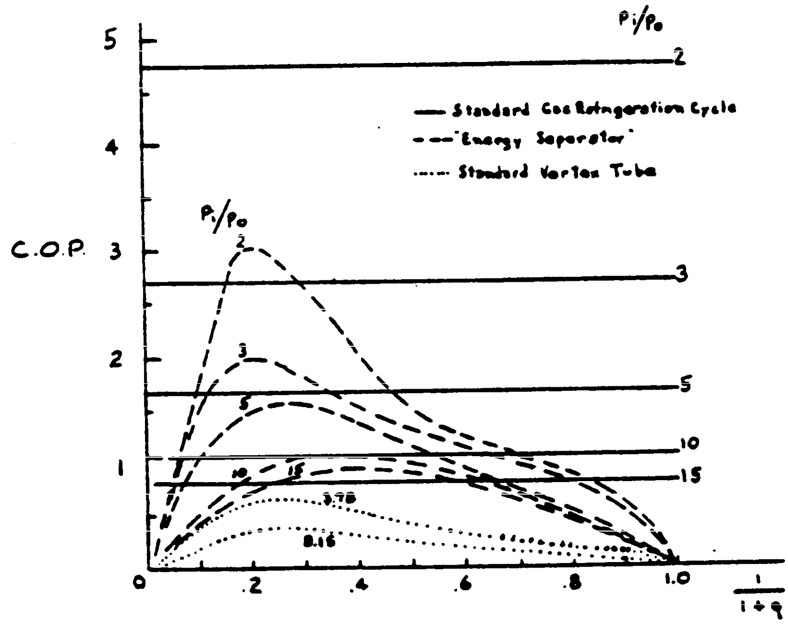


FIGURE 160. C.O.P. Comparison

2. Significant reductions in size and weight are often possible by operation at cryogenic temperatures. Power generating and distribution components are typical examples. Since many conductor materials exhibit low electrical resistivity at low temperatures, operation at cryogenic temperatures can greatly reduce IR^2 losses and conductor sizes.

3. In some vehicles (notably spacecraft, missiles, and some aircraft), cryogenic liquids are carried on board for use in propulsion. Since these liquids are vaporized and heated before being introduced into the reaction chamber, they offer an attractive ultimate sink for the waste heat developed by electronic systems, provided an acceptable heat transfer mechanism can be devised.

4. For space applications, the equilibrium temperature of a passive body in space, not too near a star, is calculated to be about 3°K.

5. Superconducting devices usually operate at maximum temperatures of 17°K.

Cryogenic temperatures are normally produced by utilization of preliquified gasses. In rare instances, cryogenic refrigeration systems employing mechanical expansion engines are used to produce cryogenic temperatures on site, but these are specialized installations, not normally within the scope of electronic equipment system designs. Data on some of the more common liquified gasses are given in Table XXXII. The temperature of the liquid bath can be varied between the triple point and the critical point for any of the liquified gasses by changing the pressure above the liquid. Thermodynamic properties, such as specific heat or thermal conductivity are not given, since they are temperature dependent, and cannot be considered constant, as is usual at normal temperatures.

Operation of heat generating electronic equipment immersed in a cryogenic fluid will almost invariably lead to boiling heat transfer. The correlations and equations of chapter 11 are applicable. Intermediate liquid loops using liquid-to-cryogenic fluid heat exchangers may be used to avoid the design problems associated with direct immersion. While the heat exchanger design criteria of chapter 10 are generally applicable, the design of these transport loops and heat exchangers is best left to specialists.

Cryogenic cooling applications present many design problems peculiar to the field, so that the aid of specialists should always be enlisted. A few of the specific problem areas are very briefly reviewed below:

1. Material properties, thermal. As mentioned previously, properties such as specific heat and thermal conductivity are not constant at cryogenic temperatures, but are temperature dependent. Specific heat generally decreases with temperature, so that at very low cryogenic temperatures, the thermal capacitance effect in "filtering" transient heat loads becomes negligible.

The extreme temperature variation encountered in cryogenic devices can result in severe stresses built up as a result of thermal expansion (or contraction). Minor temperature fluctuations at cryogenic temperatures are not generally harmful, since most metals have virtually finished contracting by the time liquid nitrogen temperature (77°K) has been reached. However, temperature differences through a cross section of a cryogenic device can impose severe stresses on joints, particularly vacuum joints on dewar type containers, unless suitable expansion members are included. Similarly, joined metals of different contraction rates will be stressed in cooling from fabrication to cryogenic temperatures. For example, a stainless steel tube joined to an aluminum tube by a lap joint will experience a differential contraction of approximately 0.1% in cooling from 300°K to 77°K. With an elastic modulus of 10, this strain would result in a stress in the aluminum member of 10,000 psi. Repeated cycling could induce joint failure.

2. Material properties, mechanical. Most common metals show a moderate increase in tensile strength as temperature is reduced, so that strength in iteself is not generally a limiting factor. The governing criterion is that many materials lose their ductility (become brittle) at cryogenic temperatures. Among metallic elements, the pure metals and alloys with face centered cubic structures (generally copper, aluminum, nickel, and austenitic ferrous alloys) are widely used because they retain their strength and ductibility at low temperatures. Most other materials exhibit a temperature range over which a transition from ductile to brittle behavior occurs.

Non-metallic materials also become brittle at cryogenic temperatures (PTFE being a notable exception). This, combined with their very large contraction coefficients, poses severe compatibility problems. Plastics are nearly always used with fillers for cryogenic applications to modify their properties.

3. Safety. The use of cryogenics imposes many safety hazards; however, the history of the industry attests to the fact that adequate precautions will reduce injury potential to almost zero.

Since very low temperatures are required to maintain cryogenic fluids in the liquid state, there will be an inevitable flow of heat into the system from normal environments, and suitable vent systems must be provided to avoid excessive pressure build up.

Vaporization of the liquid, either through venting or through a design process, can result in exclusion of air from enclosed or partially enclosed spaces. The result may be a toxic or asphyxiating environment, dependent on the particular cryogenic fluid. In the case of oxygen, oxygen enrichment may result in a highly combustible environment.

Cold injury is an obvious danger in cryogenics. Tissue damage may result from contact with the low temperature devices. Contact with cold structures or piping is usually more damaging than direct contact with cryogenic liquid (for short periods), since body heat will vaporize the liquid at the contact areas, forming an insulating vapor blanket.

The above discussion presents only a very brief description of the cryogenic field. This field embraces the disciplines of physics and physical chemistry, thermodynamics, fluid flow, heat transfer, and mechanics of materials, among others and each of these disciplines is different from its normal environment counterpart by reason of low temperature effects. References 101-105 are cited as sources of cryogenic data and design criteria. Specific design problems should always be assigned to experts in the field.

TABLE XXXII. Properties of Common Cryogenic Liquified Gasses

Gas	Boiling Point @1 atm (^{o}K)	Triple Point Temp (^{o}K)	Triple Point Press (atm)	Critical Point Temp (^{o}K)	Critical Point Press (atm)	Heat of vaporization at BP Watt-min/lbm
O_2	90.1	54.4	.01	154.8	50.14	1609
N_2	77.4	63.2	.124	126.2	33.54	1506
Ne	27.2	24.6	.427	44.5	26.86	657
H_2	20.4	13.8	.07	33.3	12.8	3386
He4	4.2	-	-	5.3	2.26	155
He3	3.2	-	-	3.3	1.15	-

12.8 Other cooling techniques. In aircraft and spacecraft it is desirable to utilize thermal capacitors. One special variation of the thermal capacitor which undergoes a temperature reduction and a change of physical properties has been investigated by the Naval Air Development Center. Non-Newtonian fluidsundergoing catalytic chemical reactions were investigated (References 116 and 117).

Some of the detailed conclusions reported by NADC are:

"Chemically reacting collagen solutions with their inherently built in control system can be used as a means of reducing temperature cycling of electronic component parts with overall improvement in equipment reliability."

"The non-Newtonian collagen solutions used as a chemical reactant in these investigations are colloidal systems whose physical properties appear to be much different from plain water."

"From the results obtained at temperatures as low as -65°F, it appears that a colloidal water solution (containing as low as 15% colloidal collagen) does not have the deleterious expansion and cracking properties of plain water. This being the case, colloidal water solutions may be utilized for more cooling applications than exist at the present time."

"In a catalytic chemical reactor, non-Newtonian fluids (collagen-protein water solutions) can be made to change (as a function of time) their physical properties even before extensive chemical reaction temperatures are approached."

"The collagen solutions investigated are non-Newtonian and have time dependent properties of viscosity."

"The viscosity of a non-Newtonian fluid is a complex variable and the thermal response of these fluids as a function of time is much different from conventional type Newtonian fluids."

"The change in mainly physical properties (viscosity and specific heat) as a function of time, resulted in the overall effect of temperature drops of fluid exiting from the heated sections of the reactor. Temperature stabilization at reduced exit temperature did occur after a period of time, which ranged from about 12 to 30 minutes depending upon the nature of reactant materials."

"The results of chemical reactions with non-Newtonian fluids showed that it is possible to increase the heat load into the reactor by a factor of 210%. This can be done without increasing the exit temperature of the fluid above that which would normally be obtained with presently availble conventional fluids."

"Even without chemical reaction, the heat transfer properties of non-Newtonian fluids are considerably different from conventional Newtonian types. Viscosity of non-Newtonian fluids is a complex variable and its value is time dependent. The results of this investigation have shown that the overall pressure drop through the reactor could be varied as a function of time and that exit temperatures of these fluids could be considerably reduced with time."

"Most known catalytic reactions occur at temperatures above 300°F and cannot be utilized for cooling avionic and other types of systems."

"The results of chemical reaction tests with non-Newtonian fluids described in this report occurred at temperature levels below +200°F."

"Non-Newtonian fluids of the types described here have the following characteristics:

1. The ability to absorb much larger amounts of heat than conventional fluids without a large temperature rise.

2. The ability to make large changes of physical properties, particularly viscosity, as a function of time."

"The overall thermal effect being to cause a desirable temperature reduction of the fluid leaving the reactor as well as versatility and flexibility of physical properties which can be optimized for transient thermal environmental operating conditions."

"Non-Newtonian fluids differ from Newtonian types because their shear stress values are not proportional to the rate of shear strain. Relatively little theory is available for non-Newtonian fluids, particularly those whose viscosity is a complex variable and varies as a function of time. This discipline in combination with catalytic chemical kinetics is a completely dark area. In view of the potential for subsequently obtaining more effective cooling systems by being able to change physical properties to suit transiet environmental conditions, it is concluded that future investigative efforts should be directed toward developing an overall theory along with obtaining more experimental data on a wider range of non-Newtonian type materials."

13. STANDARD ELECTRONIC MODULE PROGRAM (SEMP) THERMAL DESIGN

Note: The Standard Electronic Module Program is relatively new and thermal data are currently being accumulated from various sources. Consequently, some of the data in this SEMP chapter are preliminary and should be treated as such. Numerous documents on SEM thermal characteristics are referenced and many of these documents are also of a preliminary nature and some of the information therein is invalid and superceded. Therefore, the referenced thermal data should be used with caution. It is planned to prepare and distribute updated supplemental material for this chapter in the near future.

13.1 Standard electronic module program, general. The Standard Electronic Module Program (SEMP) is a tri-service sponsored electronics standardization program. Its purpose is to define and make available a family of commonly used module functions that will reduce the cost and expedite the design and production of military electronics systems, while improving their logistical support posture. SEMP is a relatively new conept, and is presently utilized in a limited number of Navy electronics systems (about 50). However, the use of SEMs is or will be mandated, where possible, on almost all present and future procurements, so that the implementation of SEMPs will be increasingly evident as new systems are deployed.

The basic module of the SEMP system (termed the 1A module) has a span of 2.62 inches, a thickness of 0.290 inch, and a height of 1.95 inches. (See Figure 161) Multiple growth modules may be incrementally larger in span, thickness, or both. Growth increments are 3.00 inches in span and 0.300 inches in thickness. Each approved SEM performs a prescribed electrical function, as defined in the detailed Module Specification. For further description of the SEM consult references 118-125, inclusive.

13.2 Standard electronic module thermal aspects.

13.2.1 General. Two thermal paths for heat removal are recognized within the SEM: (a) by way of the cooling fin (or fins in multiple growth modules) which is integral with each module, or (b) by way of the guide ribs at the side of each module (see Figure 161). The thermal design of the module must be such that the thermal resistance to either of these sink connections (the fin or the guide ribs) is low enough to insure satisfactory parts temperatures. The reason for this requirement is that in order for the SEM to be a true standard, it must be applicable to a variety of installations. Some of these installations will utilize the cooling fin as the primary method of module heat removal, usually by forced air cooling over the fin surfaces. Heat removal through the guide ribs in these applications may be uncontrolled, and could be minimal due to poor guide rib/structure contact, and/or thermal peculiarities of the structure. In other

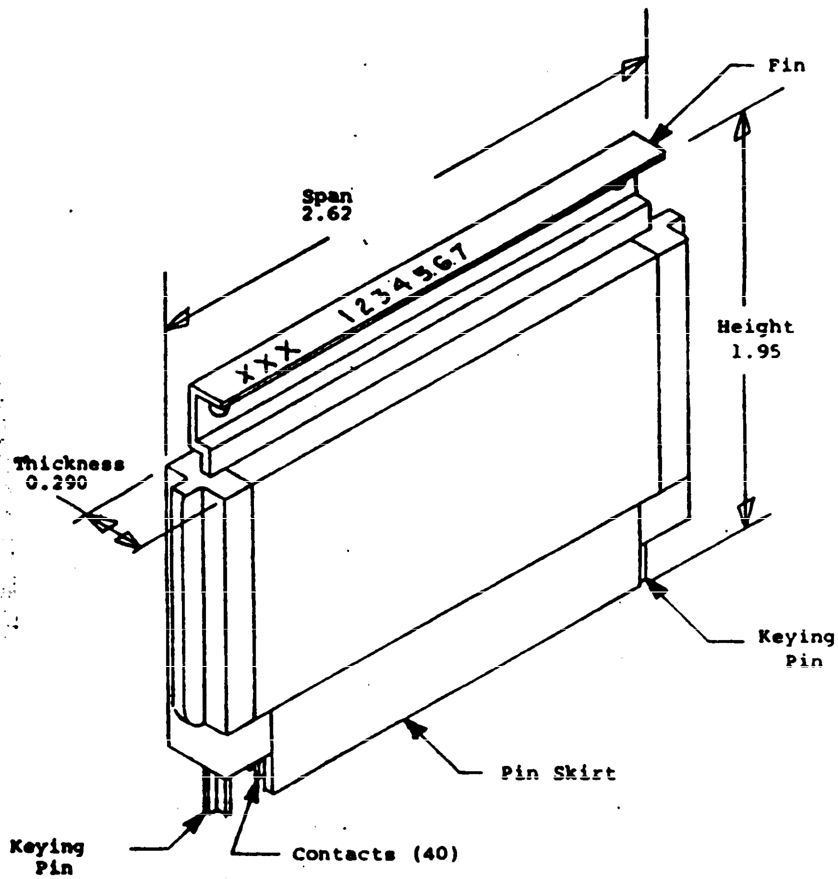


FIGURE 161. SEMP Electronic Requirements

installations, the cooling fin may be coupled with a forced air or liquid cooled structure will emphasize the guide rib thermal path, especially if no forced air is used over the cooling fin surface. In order that the standard module may function satisfactorily in either installation, both cooling paths must be independently capable of removing the modules generated heat.

The SEMP system has been designed to provide precise thermal interface for module/system designers. This interface must provide sufficiently low thermal paths within the module to maintain acceptable parts temperature when the interface is maintained at the temperature specified for the environmental class for the module. The system designer must provide system cooling to maintain the interface temperature of each module at or below the temperature specified for the module environmental class.

Two module environmental classes exist. The interface temperature for Class I is 0°C to 60°C; the interface temperature for Class II is -55°C to 100°C. The interface temperature is that of the cooling fin or guide rib surface. (See Reference 125 for other environmental factors associated with environmental classes I and II).

It is evident, then, that there are two distinct thermal designs inherent in the use of SEMs: (1) the internal design of the module, subject to the applicable interface temperature, dissipation level, and acceptable parts temperature, and (2) the overall system thermal design, so as to maintain the required module interface temperature subject to the system environmental requirements, the individual, and and combined module dissipation levels, and the system packaging, or partitioning. Section 13.2.2 will consider module thermal design; section 13.2.3 will consider system thermal design.

13.2.2 SEM thermal design.

13.2.2.1 Definition and requirements. The criteria to be met in module design are the Critical Component Temperature (CCT) and the Transient Critical Component Temperature (TCCT). CCT is the component temperature beyond which reliability is adversely affected. It is generally equal to 100% of the MIL spec, temperature rating, less 20°C. In order to meet SEMP reliability goals, a SEM must be designed so that all component temperatures are below their CCT when the module is at maximum dissipation at its rated interface temperature.

TCCT is generally equal to 100% of the MIL spec temperature rating. It is required that all component part temperatures be below their TCCT when the module is at maximum dissipation with the interface temperature at 20°C above the interface temperature rating designed for that class. TCCT requirements are intended as a safety factor to allow system useage in abnormal operating conditions, such as loss of coolant in battle conditions. Operating integrity is required under TCCT conditions. However, end-of-life reliability is understood to be adversely affected by prolonged operation under these conditions.

As mentioned previously, two environmental classes are defined for SEMs. Class I implies a maximum interface temperature of 60°C (0°C to 60°C); Class II implies a maximum interface temperature of 100°C (-55°C to 100°C). Class I will be assumed in the discussions and examples following. A similar design philosophy may be used for Class II modules, with substitution of higher interface temperatures.

The thermal design of the module then requires that all component parts be at or below their CCT when the fin or guide rib temperature is maintained at 60°C; and that all component parts be at or below their TCCT when the fin or guide rib temperature is maintained at 80°C for 60 minutes. In addition, the total module dissipation must be compatible with typical system cooling capabilities to maintain the required interface temperatures. MIL-STD-1378, MIL-STD-1389, and MIL-M-28787 are the primary documents governing SEMP modules. MIL-STD-1389 gives 2.50 watts as a recommended maximum dissipation per 1A module.

13.2.2.2 Design method. The recommended approach is to formulate a electro-thermal circuit analog, including all significant temperatures and thermal resistances. It is not normally necessary to include thermal capacitance, since CCT tests are steady state, and operation under TCCT conditions result in thermal equilibrium within 60 minutes.

Consider as an example the DIP module shown in Figure 162. This type of module construction utilizes a cutout metal frame to form a series of ribs which act as low resistance paths for heat from the individual DIP's. A simplified thermal circuit is shown in Figure 163. (Only three DIP devices are shown in the circuit for simplicity.) The significant temperature nodes, as identified by subscripts are:

- j - semiconductor junction
- c - semiconductor case
- fr - frame node at semiconductor mounting
- l - semiconductor leads
- ss - substrate at lead interface
- f - fin
- gr - guide rib

The thermal path to the pin connector via the substrate and associated copper circuitry may be significant in some applications. However, since the connector pin temperature is not a recognized or controlled thermal interface, this path has not been included in the circuit. Deliberately neglecting this path results in a slightly conservative thermal design.

The thermal path through the device leads, the substrate, and the frame can normally be neglected with little error. Semiconductor manufacturers normally provide values of the junction-to-case thermal resistance, but rarely for internal junction-to-lead thermal resistance. Consequently, this thermal resistance would have to be estimated, which can be dangerous, or be determined be test, which can be expensive. Very often, the junction-to-lead thermal resistance is high because of the fine wires used internally in the semiconductor device, as small as 3/4 mil (0.00075 in.) diameter. In addition, the leads are sometimes thermally bonded to the case at their exit, essentially equalizing T_l and T_c. The manufacturers junction-to-case thermal resistance specification then includes the internal junction-to-lead resistance. Finally, the thermal resistance through the substrate is usually difficult to calculate accurately, and in the case of typical epoxy-fiberglass laminates, is so high in comparison with other parallel paths that it may be safely ignored. Higher conductivity substrates, such as ceramics, will present significantly lower thermal resistance.

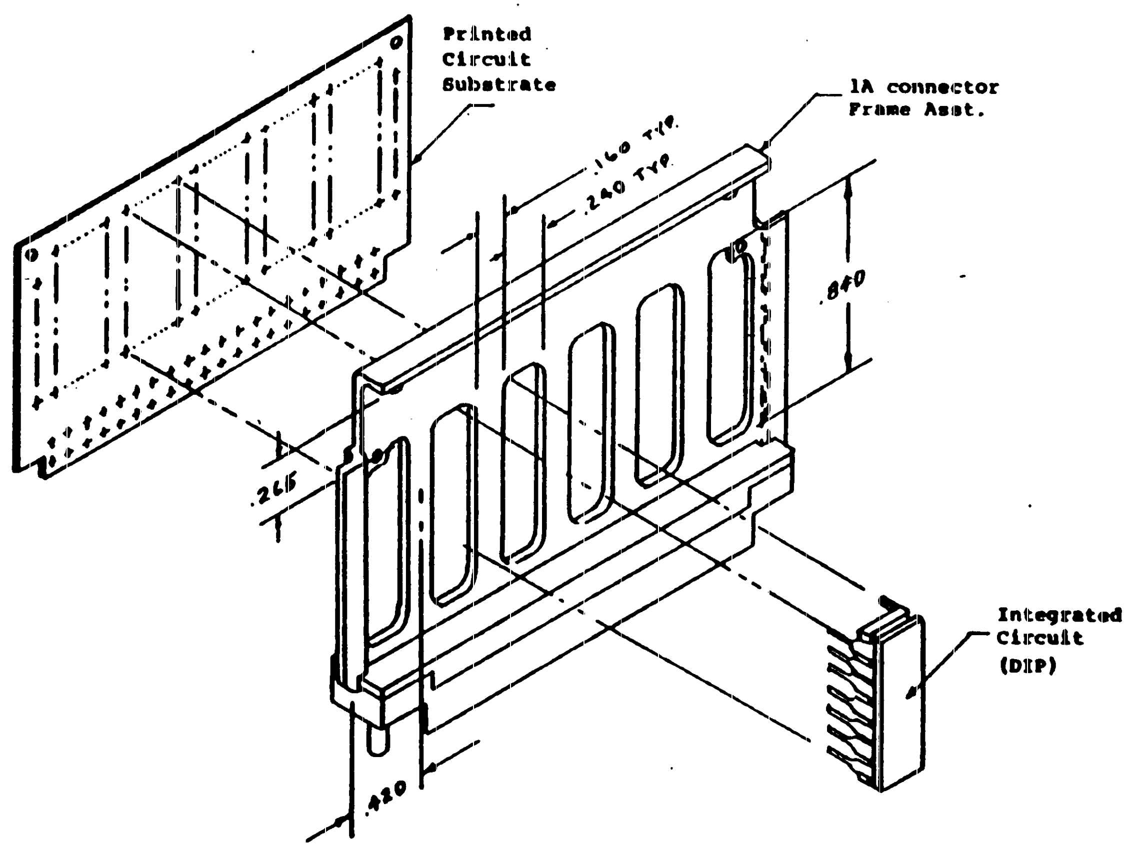


FIGURE 162. DIP Module Construction

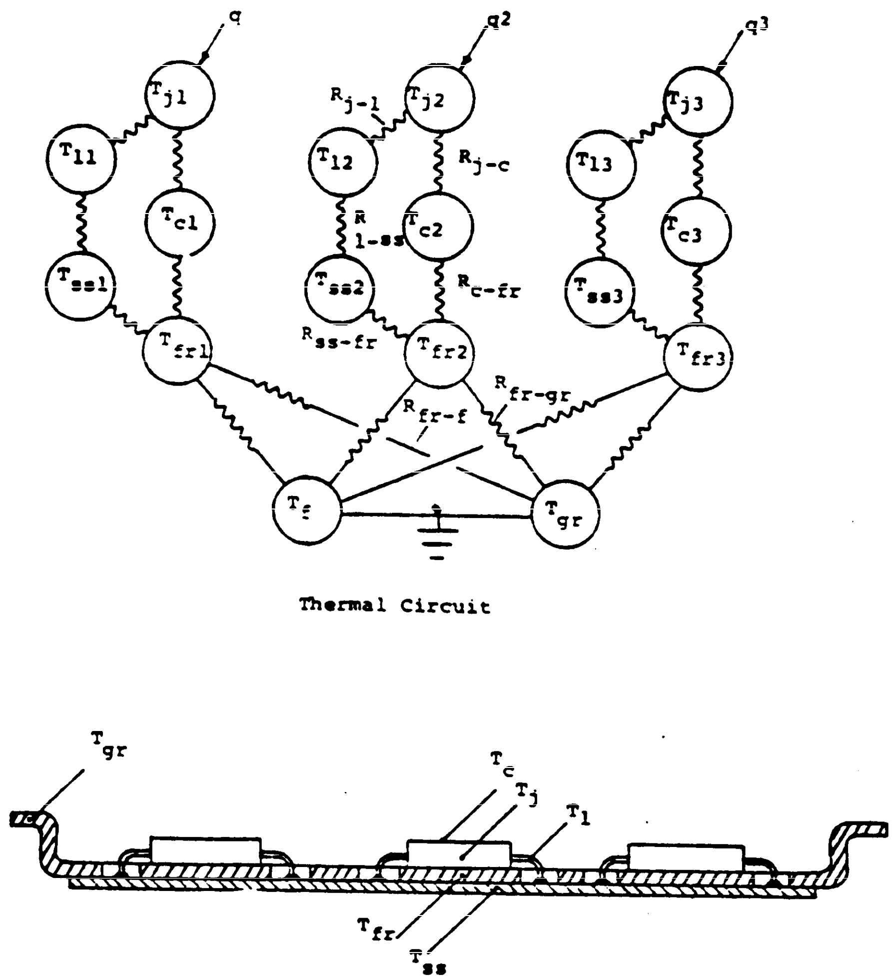


FIGURE 163. Physical Schematic Analysis of DIP Module

paths. In general, however, the junction-lead-substrate-frame thermal path may be safely ignored.

The circuit depicted in Figure 163 indicates independent frame nodes (T_{fr}) for each DIP mounting location. In actuality, of course, each frame node is thermally connected with the other frame nodes, since the frame is a continuous metallic entity. Thus, the DIP on the center rib of Figure 162 has a thermal path to the fin not only through its own mounting rib, but also through the adjacent ribs via the frame structure and the substrate, and to a lesser degree, through the end ribs. The thermal circuit can be simplified if these intra-frame resistances can be ignored. This is generally the case if the DIP dissipation on a frame is relatively uniform, so that all frame nodes are at approximately the same temperature. The module thermal designer must use some judgement as to whether the intra-frame resistance should be included in the thermal curcuit. Usually, only the effects of adjacent ribs need be included, if at all.

The frame nodes are connected thermally to the fin node, T_f, and the guide rib node, T_{gr}, which are specified for these design purposes as having a constant interface temperature. As previously mentioned, the thermal resistance to either of these nodes must be low enough to ensure satisfactory component temperatures with no heat removed from the other interface node. Generally, the average component-to-fin resistance and the average component-to-guide ribs resistance will be approximately equal.

If all the above assumptions are included, the circuit of Figure 163 may be further simplified to the circuit of Figure 164, where the subscript i signifies the interface, which may be the cooling fin or the guide ribs. (Again, only three devices are shown for simplicity.)

This simple circuit can be solved for node temperatures quite easily, provided that the various thermal resistances can be defined. Frequently, more complex circuits are over-simplified in such a manner to obtain first approximations to temperatures and/or resistances. If the required resistances fall within a generally practicable range, circuit design and layout can proceed with reasonable confidence in the thermal adequacy of the design. Eventually, of course, a detailed thermal analysis should be performed. Ideally, the thermal model should be updated and re-evaluated coincidentally with the electrical design and physical layout of the module.

Figure 165 shows a solid-fin module construction. In this construction, the PC board is bonded to the solid metal frame backbone, and components are mounted to one side of the board. Figure 166 is a thermal analog circuit. Again, only three semiconductors are shown for simplicity. As before, the thermal path through the pin connector has been ignored. Node-to-node resistances within the PC board and within the frame are shown, and may or may not need to be included, depending on the proximity and relative dissipation levels of the devices. Normally, the node-to-node resistance within the frame is much less than the equivalent resistance in the circuit board, and the circuit board resistance (lengthwise) may be ignored. Exceptions may occur if ceramic substrates are used rather than epoxy-fiberglass laminate PC boards, or if a large portion of the area of the PC board between nodes is covered with continuous copper circuitry.

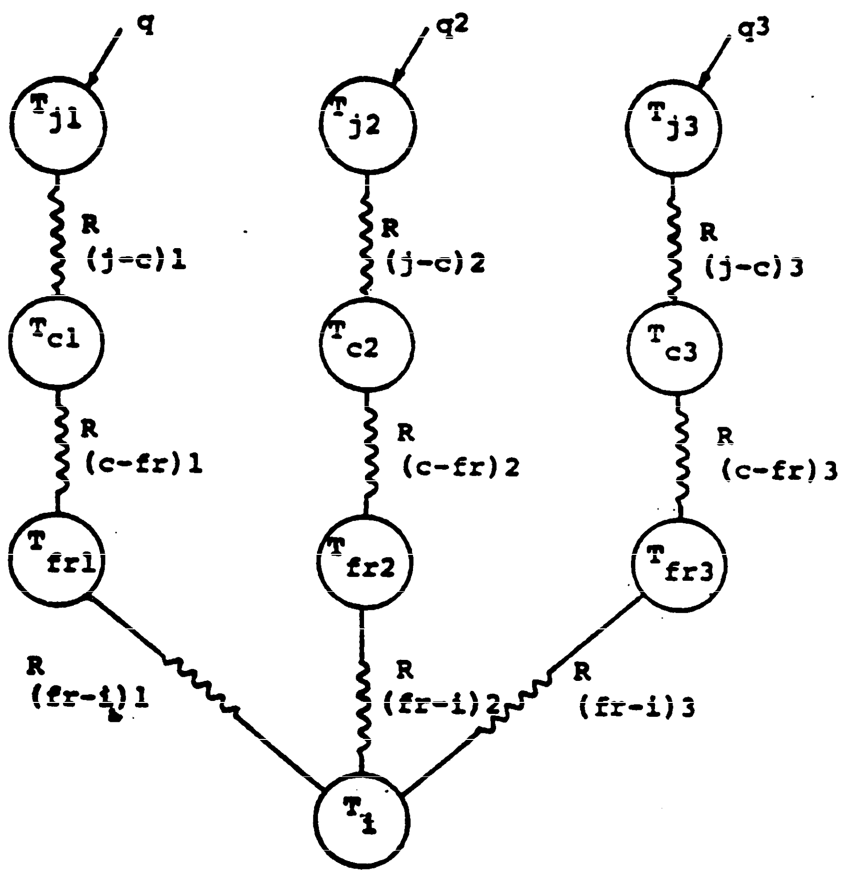


FIGURE 164. DIP Module Simplified Thermal Circuit

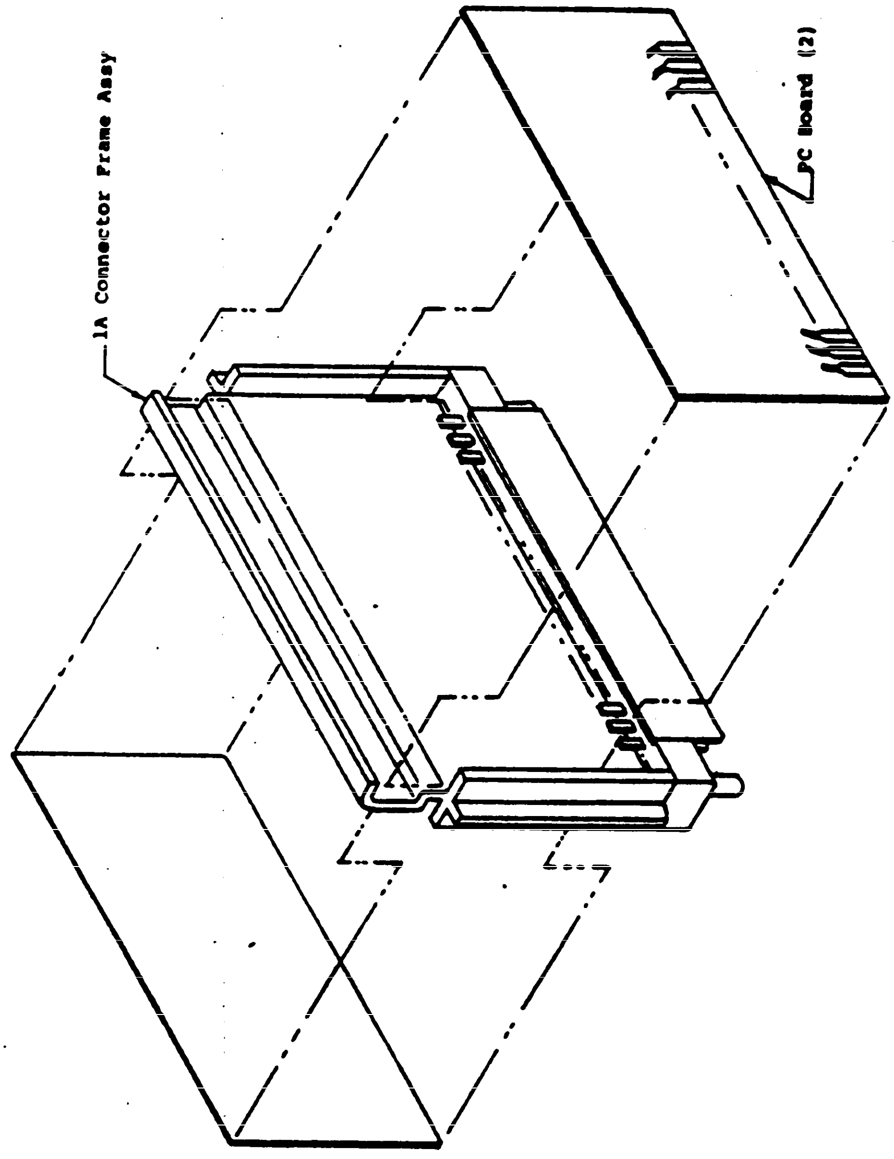


FIGURE 165. Solid Fin Module Construction

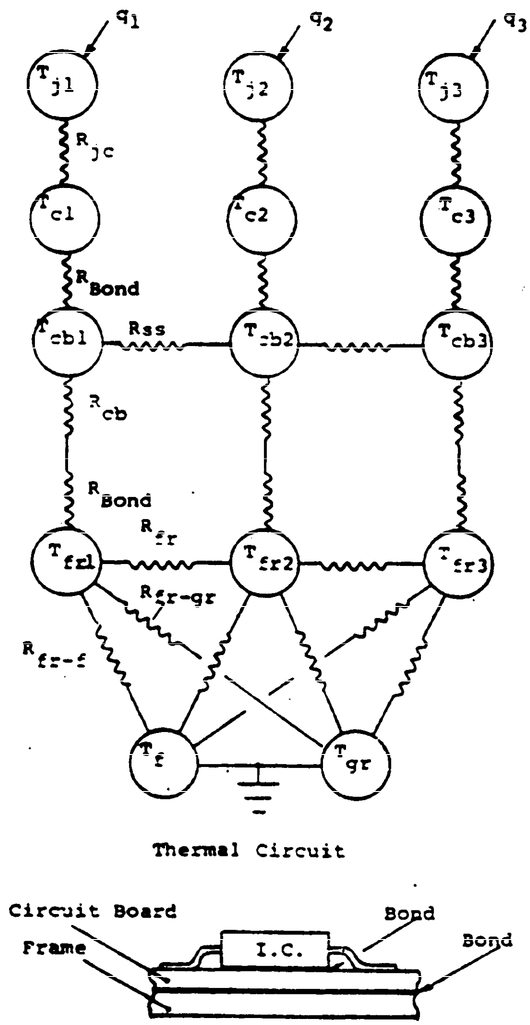


FIGURE 166. Physical Schematic Analysis of Solid Fin Module

If the node-to-node thermal path within the board may be ignored, Figure 166 may be simplified to Figure 167, where again, i is the interface (fin or guide rib). Of course, those resistances shown in series may be simply added. The node-to-node resistances within the frame may or may not be significant. Note that in this construction the thermal path from any component to the interface node includes resistances equivalent to two bonds and the thickness of the PC board.

Figure 168 depicts still another commonly used module construction. This consists of a die-cast aluminum frame, with printed circuit boards attached to both sides of the frame. Figure 169 is a thermal analog circuit. The resistance from case to substrate (c-ss) may be either an interface resistance or a bond, dependent on the construction technique used. The substrate resistance (both node-to-node within the substrate and in the thermal paths to the interface node) may be difficult to define, and is dependent to a large extent on the copper coverage of the substrate. Ceramic substrates provide significantly lower thermal paths than epoxy-fiberglass or similar substrates. The substrates to interface thermal resistance (ss-i) is dependent on the contact between the substrate and the frame around the perimeter of the substrate. Normally, the substrate has a metal clad border in the contact area, to minimize the thermal resistance.

The frames depicted in the previous examples are typical of those used in SEMs, but should not be considered restrictive. Provided the requirements of Reference 124 are met (outline size, cooling fin, and guide rib details, connector details, etc.), the physical construction of the module is the prerogative of the module designer.

The examples given included a variety of thermal resistances. These are summarized below, and will be individually discussed in the following, since minimizing one or more of these resistances may be the primary thermal task:

Symbol	Thermal Resistance
R_j-c	Junction to case, internal to semiconductor
R_c-fr or R_c-ss	case to frame or case to substrate. This is the thermal resistance encountered at the mounting of the device.
R_{Bond}	This is the thermal resistance encountered where an adhesive is used in the module construction. Typically, in mounting a device to a substrate, or in bonding a circuit card to a frame.
R_{fr}-i	Frame to interface. This is typically the conduction resistance of the metallic path through the frame from the point of device mounting to the interface (fin or guide ribs)
R_{cb}	Circuit board (substrate) resistance, through thickness
R_{ss}	Substrate resistance, normal to thickness
R_f	Frame resistance, node-to-node internal
R_{ss}-f	Substrate to frame. Interface resistance
R_{jc}	

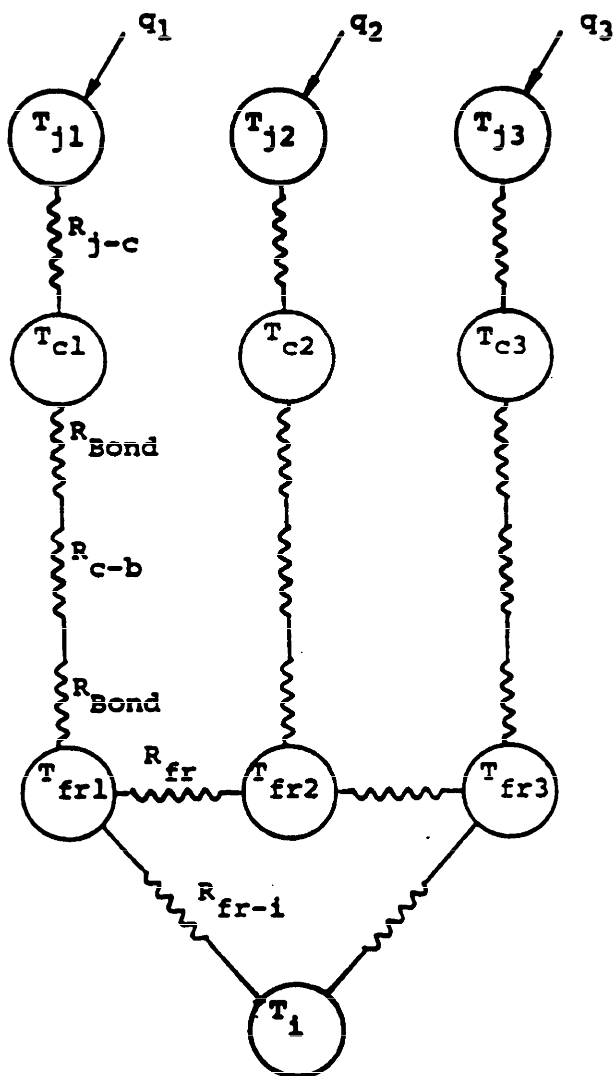


FIGURE 167. Solid Fin Module Simplified Thermal Circuit

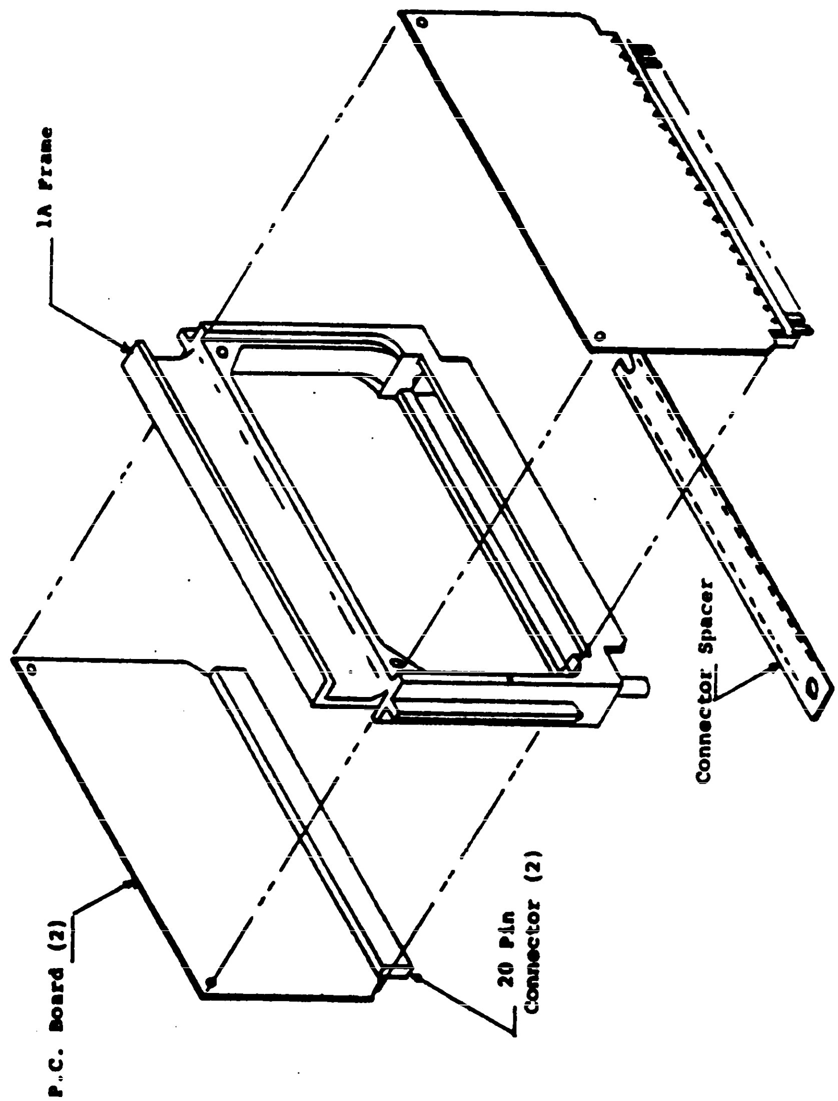


FIGURE 168. Die Cast Module Construction

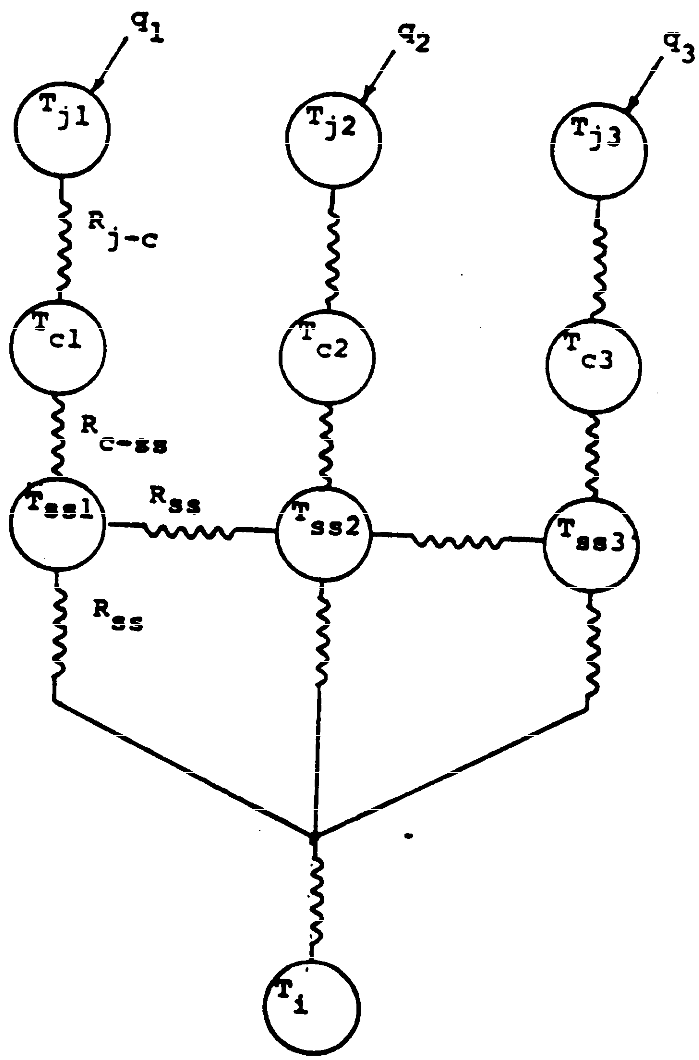


FIGURE 169. Die Cast Module Thermal Circuit

The junction to case thermal resistance of a semiconductor is a value normally defined by the semiconductor manufacturer in °C/watt. It is usually determined by measuring some temperature dependent electrical parameter of the junction at varying power levels, while holding case temperature constant with an appropriate heat sink. The junction temperature minus the case temperature divided by the power is R_{j-c}. Some devices (particularly small signal devices) are rated in terms of junction-to-air thermal resistance. This value is determined by deriving the junction temperature while the device is operated when suspended in free air. This is a poor rating technique, since it includes not only the device internal resistance, but also the effects of external natural convection, radiation, and lead conduction. It is particularly inappropriate for SEM design, since SEMs are intended to transfer dissipation heat through specific interfaces. When devices with this type of rating are contemplated for a design, it is often possible to obtain junction-to-case thermal resistance values by contacting the manufacturer. Otherwise, R_{jc} must be estimated, provided a sufficient sample is tested and statistical derating of results is employed to ensure that manufacturing tolerances in internal resistance paths are accommodated. Occasionally, a derating curve similar to Figure 170 is provided in lieu of a value of R_{jc}. The derating curve generally specifies a maximum power (P_{max}) at some base temperature of the case, T_b (usually 25°C), and a maximum case temperature. R_{jc} may be determined by

$$R_{jc} = \frac{T_{MAX} - T_{BASE}}{P_{MAX}}$$

For example, if, as shown in Figure 170, P_{MAX} = 5 watts, T_b = 25°C, and T_m = 175°C, then

$$R_{jc} = \frac{175 - 25}{5} = \frac{150}{5} = 30°C/watt$$

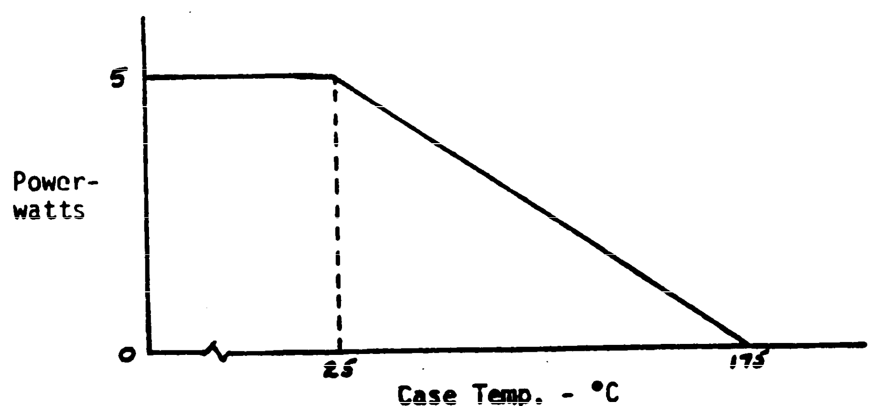


FIGURE 170. Typical Derating Curve

The junction to case thermal resistance concept originated with transistors, and is generally applied as well to integrated circuits, with a worst case R_{j-c} being specified. For medium and large scale integrated circuit, the concept becomes nebulous, however, since there may be hundreds or even thousands of junctions on a single large chip. Since these circuits are almost always digital, the dissipation per junction is generally small, but the total device dissipation can be significant. These devices essentailly become modules themselves. Usually a maximum substrate temperature can be specified for reliable operation. In any case, close thermal liaison between the manufacturer and the user is essential for these devices.

R_{c-fr} or R_{c-ss}, R_{Bond}

This resistance factor represents the resistance of the component mounting interface, which may be to the frame or to the substrate, depending on the module design. The data of chapter 8 are applicable to determination of these resistances. It is almost always advisable to use either a filler material at the interface, or some type of conduction clamp. Filler materials may be high conductivity grease-like materials, such as Dow Corning 340 heat sink compound (or equal) or may be adhesives such as epoxies. Adhesives should be used cautiously, since those which cure to a hard substance can stress components due to differential expansion. Flexible or pliable adhesives are generally preferable. The purpose of any filler material is to displace the air interstices normally present at an interface with a higher conductivity material. The filler material should completely cover the interface area, and should be of minimum thickness, usually about 5 mils (.005 in.). Most common adhesives have a thermal conductivity on the order of .009 watts/°C sq. in./in. This is roughly an order of magnitude better than air, so that the benefits of a filler become evident. Fillers specially formulated for heat transfer applications have higher conductivities. Manufacturers' data should be consulted for specific values. Where no filler is used, the interface thermal resistance is a function of the nature and surface finish of the mating materials, and of the contact pressure. Since small electronic components are usually held in place by their leads, it is impossible to develop significant contact pressure, and the interface resistance will not only be high, but will also be subject to wide variations. Simple metallic spring clips are of great advantage in that they produce an effective and predictable interface contact pressure, and at the same time, provide a parallel high conductivity path to the substrate.

R_{fr-i}

This is the conduction resistance through the frame from the point of device mounting to the module thermal interface. It is generally a low thermal resistance, but not necessarily insignificant. Most frames are made of aluminum with a conductivity of about 5 watts/sq. in.-°C/in. (Note: the conductivity of aluminum alloys can vary widely, from a low of about 2.25 w/sq. in.°C/in. for 220T4 casting alloy to 6.1 w/sq. in.-°C/in. for pure aluminum. In specific applications, be certain to use the conductivity for the alloy being used, rather than a generic conductivity for "aluminum alloys.") Typical frame thickness is about 0.050 inch.

Since $R = \frac{L}{KA}$

Where:

R = path thermal resistance, °C/watt
L = path length, inches
K = conductivity, (watts/sq. in.)/°C/inch)
A = path cross sectional area, inches

then for a typical frame, $R = \frac{L}{(5)(.05)(w)} = 4\frac{L}{w}$

Thus, for example, the resistance of one of the ribs of the frame shown in Figure 162, from the vertical center of the rib to the top of the rib is $4\frac{.840}{(2)(.160)} = 10.5$°C/watt.

$\underline{R_{cb}}$

This is the thermal resistance of the substrate, through its thickness. Conductivity of glass-epoxy laminated printed circuit borads is about .0092 watts/sq. in.-°C/in. Thus, the resistance through a 0.050 inch thick board is $R = \frac{L}{A} = \frac{.05}{.0092A} = \frac{5.43}{A}$, where A is determined by the mounting area of the device. Taking a typical integrated circuit as 0.4 by 0.25 inch, the thermal resistance through the circuit board is $\frac{5.43}{(.4)(.25)} =$ 54.3 C/watt. Significant reductions in thermal resistance can be made by using ceramic substrates. Typical conductivity of a ceramic is 0.4 watts/sq. in.-°C/in.

$\underline{R_{ss}}$

This is the substrate resistance, normal to the thickness. This can be a high resistance path through epoxy-glass laminates. For example, for a .050 inch thick printed circuit board, $R = \frac{L}{KA} = \frac{L}{(.0092)(.05)W} = 2175\frac{L}{W}$.

The resistance of this path, however, is generally reduced by the effect of the copper circuitry on the board. Two-ounce copper cladding on a circuit board is .0027 inch thick. The conductivity of copper is about 10 watts/sw. in.-°C/in. Thus, the resistance of a continuous path of 2 ounce copper is $R = \frac{L}{KA} = \frac{L}{(10)(.0027)W} = 37\frac{L}{W}$, in parallel with the $2175\frac{L}{W}$ resistance of the .050 thick circuit board. Of course, the copper does not exist in continuous paths, since its purpose is to provide interconnecting circuitry, and the copper thermal resistance must be increased in inverse proportion to the percentage area covered by copper. Even so, the copper circuitry provides a significant thermal path, and from a thermal viewpoint, as much copper should be left on a circuit card as possible. In particular, copper pads should be left directly beneath component mounting locations. When the substrate offers too high a thermal resistance to assure reliable component temperatures, consider using either (a) a different module construction, which provides a metallic thermal path to the interface, usually through the frame or (b) a high conductivity substrate, such as ceramic, or (c) metallic straps between the components and the frame to provide a high conductivity path.

R_f

R_f is the node-to-node thermal resistance within the frame. It is similar to R_{fr-i} described above. These resistance paths are included in more detailed module thermal circuits.

R_{ss-f}

This is the interface resistance between a substrate and the frame. It can be significant in a "window" type frame, in which the substrate contacts the frame only around its periphery. It can be reduced by enlarging the contact area and by leaving a metal clad border on the substrate to form the interface. A typical specific thermal resistance with a metal clad border is about 0.85°C/(watt/sq. in. of contact). Thus, if a substrate is 2.15 inches wide and 1.050 inches high with an interface width of 0.06 inches the thermal resistance at the top of the substrate (in path to fin) is $\frac{.085}{(0.06)(2.15)} = 6.589$°C/watt, and the thermal resistance at either side of the substrate (in the path to the guide rib) is $\frac{.085}{(0.06)(1.050)} = 13.49$°C/watt.

13.2.2.3 Typical module thermal resistance values. Considerable data regarding thermal resistance values of typical SEMs have been generated from the applications to date of the SEMP. Additional data are being accumulated as more systems using SEMP are designed, tested, and deployed. The data in Table XXXIII represent typical thermal resistance values for a variety of presently utilized SEMs.

In some instances, there are considerable variations in reported thermal resistance values for the same type of module. For example, the calculated thermal resistance, component-to-fin, for a die cast frame with a printed circuit board ranges from 21.5 to 52.6°C/watt. (Some of this is attributable to the percentage of copper remaining on the PC board.) Also, in general, thermal resistance values determined by tests tend to be considerably lower than the corresponding computed values.

13.2.2.4 Design examples, module thermal design.

Design Example 13-1: For the construction depicted in Figure 162, derive expressions for the thermal resistance values from the case to the fin and to the guide rib for the integrated circuit mounted on the center rib, assuming the following:

Frame thickness = 0.050 inches
Frame materials: aluminum; k = 5.6 (watts/in.^{2})/(°C/in.)

TABLE XXXIII. Typical Module Thermal Resistance Values.

CONSTRUCTION	SIZE	THERMAL RESISTANCE, °C/W			
		COMP-FIN		COMP-GUIDE RIB	
		CALC.	TEST	CALC.	TEST
DIP/PC (Fig 11-2)	1A	10.9	6.85	9.7	8.83
	2A	14.7	-	15.5	-
Solid Frame/PC(Fig 11-5)	1A	9.8	8.9	6.25	6.25
Solid Frame/Ceramic (Fig 11-5)	1A	6.52	6.02	3.33	8.00
Die Cast/PC (Fig 11-8)	1A	21.5	14.7	27.8	15.0
Die Cast/PC (Fig 11-8)	1A	21.9	-	19.6	-
Die Cast/PC (Fig 11-8)	1A	23.1	-	26.6	-
Die Cast/PC (Fig 11-8)*	1A	52.6	-	52.3	-
Die Cast/PC (Fig 11-8)**	1A	31	-	-	-
Die Cast/PC (Fig 11-8)	2A	-	13.8	-	13.8
Die Cast/PC (Fig 11-8)	2A	-	9.1	-	10.4
Die Cast/Ceramic (Fig 11-8)	1A	14.5	6.8	10.4	6.7
Die Cast/Ceramic (Fig 11-8)	2A	25.4	-	23.4	-
Die Cast/PC, potted	1A	-	9.7	-	10.1
Special Heat Sink Design	2A 2A	-	2.4 4.4		3.5 5.1

* double sided board, 50% copper
** double sided board, 90% copper

Bonding material, component-to-frame and frame-to-subsrate:
thickness = 0.005 inch
k = 0.009 (watts/in.^{2})/(°C/in.)
Substrate: glass-epoxy laminate, thickness = 0.050 inch,
k = 0.0092 (w/in.^{2})/(°C/in.)

There are a total of 5 integrated circuits, of equal dissipation.

Solution:

Case-to-fin thermal resistance. Since all I.C.'s dissipate equal amounts of power, ignore any heat flow paths parallel to the fin, and the simplified circuit of Figure 164 may be used. The significant resistances to be determined are from the case to the frame, R_{c-fr} and from the frame to the fin, R_{fr-f}.

R_{c-fr} = Area = (0.160)(0.625) = 0.10 sq. in.
Length = 0.005 inch
K = 0.009 (watts/in.^{2})/(°C/in.)

$$R_{c-fr} = \frac{L}{KA} = \frac{.005}{(.009)(.1)} = 5.55°C/watt$$

R_{WEB} = Area = (0.160)(0.05) = 0.008 sq. in.
Length = $\frac{0.840}{2}$ = 0.420 inch (to top of web)

k = 5.6 (watts/sq.in.)/(°C/in.)

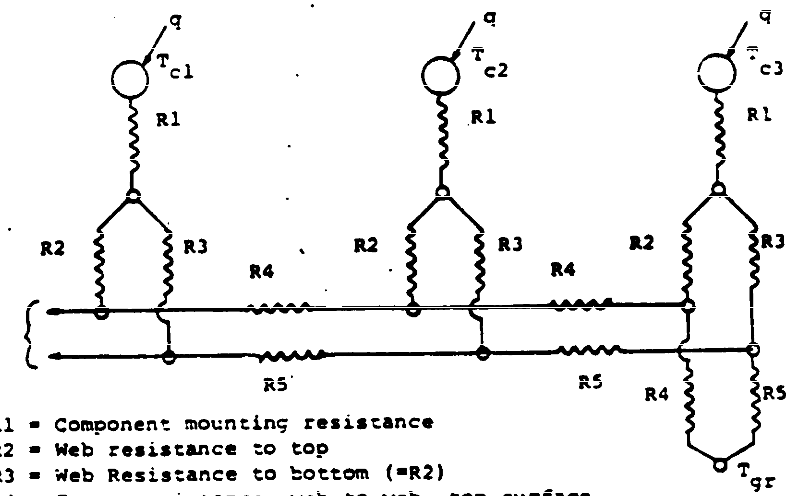
$$R_{WEB} = \frac{L}{kA} = \frac{0.420}{(5.6)(0.008)} = 9.39°C/w$$

R_{FIN} = Area = (0.05)(1.60 + .240) = 0.02 sq. in.
Length = 0.265 inch
k = 5.6 (watts/sq. in.)/(°C/in.)

$$R_{FIN} = \frac{L}{kA} = \frac{0.265}{(5.6)(0.02)} = 2.37°C/watt$$

Total resistance = 5.55 + 9.38 + 2.37 = 17.30°C/watt

Case-to-guide rib thermal resistance. Since all the I.C.'s of interest dissipate similar power, the heat flow is symmetrical about the module vertical centerline, and the thermal resistance from the center IC may be calculated on the basis of the temperature rise due to one-half the dissipation flowing to either guide rib. The major thermal resistances encountered are the component-to-web resistance, the web resistance, and the frame resistance. Note that there are two parallel paths through the frame, one across the top, and one across the bottom. Assume equal thermal resistances through both paths. The thermal network is slightly complicated, since additional heat sources are introduced at each intervening web between the IC in question and the guide rib. The network is shown in Figure 171.



R1 = Component mounting resistance
R2 = Web resistance to top
R3 = Web Resistance to bottom (=R2)
R4 = Frame resistance, web-to-web, top surface
R5 = Frame resistance, Web-to-web, bottom surface

FIGURE 171. Thermal Circuit, Design Example 13-1

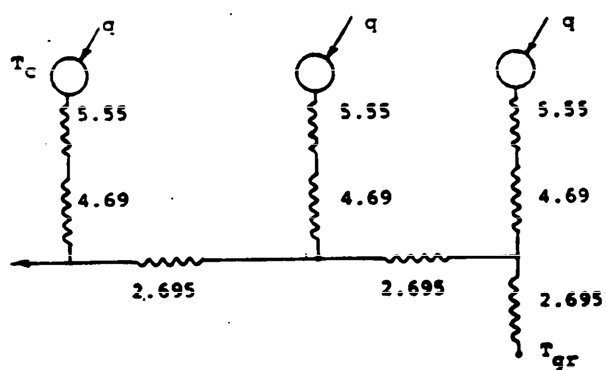


FIGURE 172. Simplified Thermal Circuit, With Values of R in. °C/W

Resistance values are:

$R1 = 5.55$ °C/watt (from computations above)
$R2 = 9.38$ °C/watt (from computations above)
$R3 = 9.38$ °C/watt

$$R4 = \frac{L}{kA} = \frac{(.160 + .240)}{5.6\ (.256)(.05)} = 5.39\text{°C/watt}$$

$R5 = 5.39$ °C/watt (assumed = R4)

Because of the similarity between the top and bottom heat flow paths (R2 = R3, and R4 = R5), the network can be further simplified to that shown in Figure 172. The component case temperature, T_c, is computed to be:

$$T_c = T+2.5q(2.695)+1.5q(2.695+2.695)+0.5q(2.695)+1.0q(5.55+4.69)$$

The thermal resistance is the temperature rise over the guide rib divided by the dissipation.

$$R = \frac{T_c - T_{gr}}{q} = 2.5(2.695)+1.5(5.39)+0.5(2.695)+1(10.24)$$

$$R = 26.41\text{°C/watt}$$

and for the outermost IC,

$$R = \frac{T_c - T_{gr}}{q} = 2.5(2.695)=16.98\text{°C/watt.}$$

If the network were not symmetrical as assumed, due to non-identical heat dissipations per I.C., or different thermal resistances in the top and bottom paths, it would be necessary to establish a set of simultaneous linear equations and solve them. The resultant matrix, for even simple networks, generally involves 10 or more equations, making computer or mini-computer solutions desirable, and often mandatory.

Design Example 13-2: Compute the thermal resistance to the fin and to the guide rib for a component (I.C.) centrally located on a substrate using the construction of Figure 165. Consider both the case of a glass-epoxy laminate substrate, and a ceramic substrate. Ignore other heat loads on the substrate. Use the following data:

Frame: Thickness = 0.050 inch
Material = aluminum, k = 5.6(watts/sq. in.)/(°C/in.)
Substrate: Thickness = 0.025 inch
Case I = glass-epoxy, k = 0.009 (watts/sq. in.)/(°C/in.)
Case II = ceramic, k = 0.4 (watts/sq. in.)/(°C/in.)
Bonding agents: Thickness = 0.005 inch
k = 0.009 (watts/sq. in.)/(°C/in.)

Solution:

Thermal resistance, component-to-fin. Significant thermal resistances are the component-to-substrate resistance, R_{c-ss}; resistance through the

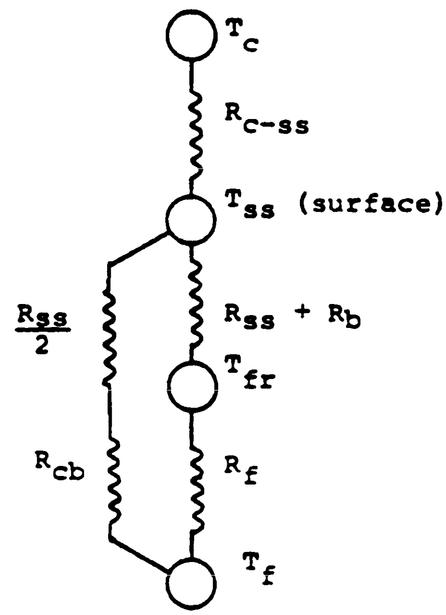


FIGURE 173. Design Example 13-2, Thermal Network, General

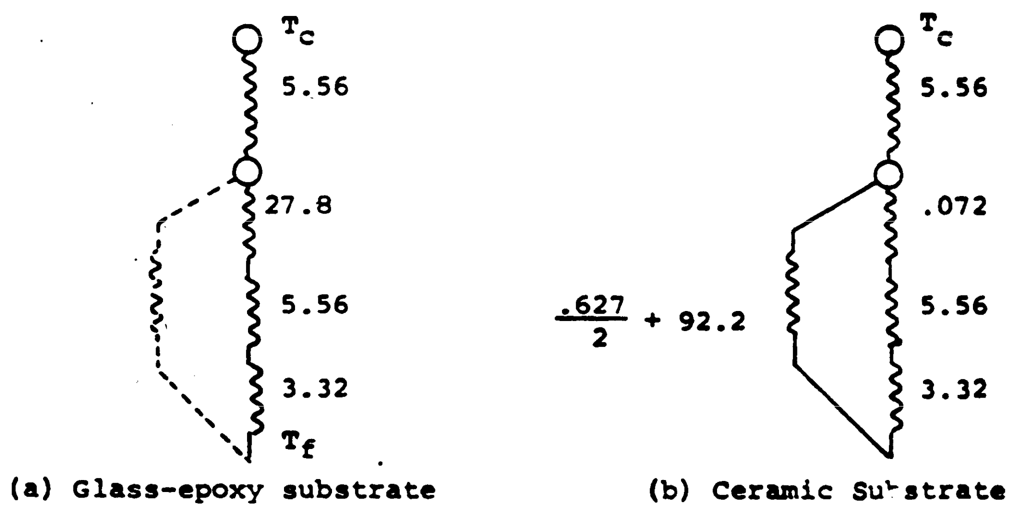


FIGURE 174. Design Example 13-2, Thermal Networks, With Values of R in. °C/W

substrate, R_{ss}; resistance along substrate, R_{cb}; resistance through the substrate-to-frame bond R_b; and resistance of the frame, R_f. The thermal network is shown in Figure 173.

Case I

Component-to-substrate, R_{c-ss}

Area = (0.250)(0.4) = 0.10 sq. in.
Length = 0.005 inch
k = 0.009 (watts/sq. in.)/(°C/in.)

$$R_{c-ss} = \frac{L}{kA} = \frac{0.005}{(0.009)(0.10)} = 5.56°C/watt$$

Normal substrate resistance, R_{ss}

Area = (.250)(0.4) = 0.10 sq. in.
Length = 0.025 inch
k = 0.0092 (watts/sq. in.)/(°C/in.)

$$R_{ss} = \frac{L}{kA} = \frac{0.025}{(0.0092)(0.10)} = 27.3°C/watt$$

Bond resistance, substrate-to-frame. While the bond is continuous over the entire surface, the substrate resistance is so high that the effective area for a given component may be taken as the component base area. Thermal diffusion through the substrate is ignored.

$$R_b \doteq R_{c-ss} = 5.56°C/watt$$

Frame resistance R_f. The thermal path characterisitcs (length and area) for a small source diffusing to a wide fin are difficult to determine. Assume the path width as half the fin width (average, assuming a point source), and the path length as the half the module height. Then,

$$\text{Area} = \frac{2.16}{2}(0.05) = .054 \text{ sq. in.}$$

Length = 0.975 inch
k = 5.6 (watts/sq. in.)/(°C/in.)

$$R_f = \frac{L}{kA} = \frac{0.975}{(5.6)(0.054)} = 3.22°C/watt$$

Substrate transverse resistance, R_{cb}. This is determined similarly to the frame resistance, except the conductivity is 0.0092 instead of 5.6, and the thickness is 0.025 inch instead of 0.050 inch.

$$R_{cb} = \frac{L}{kA} = \frac{2(.975)}{(0.0092)(0.054)} = 4012°C/watt$$

This path may obviously be ignored. The thermal circuit with numerical resistance values is shown in Figure 174a. Total thermal resistance (ignoring substrate path)

$$R_{cf}=R_{c-ss}+R_{ss}+R_b+R_f=5.56+27.8+5.56+3.22=42.14°C/watt$$

Case II Ceramic Substrate

For a ceramic substrate, R_{ss} will be reduced by the ratio of glass-epoxy/ceramic conductivity.

$$R_{ss} = 27.3 \quad \frac{.0092}{0.4} \quad = 0.627°C/w$$

Similarly, R_{cb} will be reduced by the same factor

$$R_{cb} = 4012 \quad \frac{.0092}{0.4} \quad = 92.2°C/w$$

R_{cb} is no longer a negligible factor.

Solving the network of Figure 174b yields R_{cf} = 13.72°C/watt.

The thermal advantages of using a high conductivity ceramic substrate are apparent. The main intent of this example was to demonstrate the effect of using high conductivity substrates, even when a high conductivity parallel path (the frame) exists. Actually, both the glass-epoxy and ceramic substrate thermal resistance will be higher than indicated, because of the additional heat introduced by other sources on the board. This is a common error in module analysis. A thermal resistance, R, is calculated for a given component. Later a temperature rise for that component is calculated from $\Delta T = qR$, where ΔT is the rise, and q is the component dissipation. Portions of the thermal path will carry more heat, q, than that of the single component, and the temperature rise will be higher than computed.

13.2.3 System thermal design using SEMP

13.2.3.1 Requirements. The thermal design of a system utilizing SEMP begins with the assumption inherent in SEMs-that the module will be adequately cooled if either of the thermal interfaces (cooling fin or guide rib) is at 60°C for Class I or 100°C for Class II. The simplest, and most conservative approach, is to choose either the fin or guide rib interface, assume all the heat is extracted from the module through this interface, and design the cooling system so as to maintain the required interface temperature. A number of methods may be used for heat removal at the module-system interface:

1) Natural convection cooling from the module cooling fin. This is generally limited to low dissipation modules.

2) Forced air cooling of the module cooling fin. This is probably the most extensively used cooling method. Considerable data are available relatively to forced air heat transfer coefficients over the fins.

3) Cooling fin in contact with a metal plate (or extended fin) which in turn is cooled by natural or forced air convection. This system has the disadvantage of imposing the additional interface thermal resistance between the fin and the plate. It has the advantage of not allowing cooling air to impinge directly on the module, and thus may be useful in airborne systems, where cooling air contamination could be determental to the electronic circuitry. The metal plate may be finned on the air side, if required. This system is essentially an SEMP adaption of the cold-plate technique described in chapter 9.

4) Module cooling by conduction from the guide ribs to the chassis. The chassis in turn is cooled by either forced air or by liquid coolant pumped through integral cooling passages, with liquid cooling being more common. Spring clips which provide good thermal contact at the guide rib interface are essential when the guide ribs are used as the thermal interface.

In extreme cases, combinations of the foregoing system cooling methods may be used.

13.2.3.2 System design methods and typical values

13.2.3.2.1 Natural convection cooling. Natural convection cooling of the cooling fin is used only for relatively simple systems with low dissipation modules. The heat transfer formula for natural convection is given in chapter 8, equation 8-28 as:

$$q/A = Ck\,(a/L)^{1/4}\,\Delta T^{5/4}$$

Assuming a 1A module with the 2.62 inch dimension vertical (vertical fin), and with typical fin dimensions:

c = 0.55 for a vertical surface (from Table IV)
$k = 6.945 \times 10^{-4}$ (watts/sq. in.)/(°C/in.) for air
$A = (2.32)(2)(.4 + .2) = 2.784$ sq. in.
$a = 1.25 \times 10^3$ units/(cu. in.-°C) for air
$L = 2.32$ inches

Then $q = 5.123 \times 10^{-3}\,\Delta T^{5/4}$

The temperature rise of the fin over the cooling air may be computed for any given level of power dissipation. Since by definition, $R = \frac{\Delta T}{q}$, where R = thermal resistance in °C/watt, R may also be computed for any given dissipation, q. Note that R is not a constant because of the non-linear relationship of q and ΔT. Thus, to maintain a 60°C fin temperature in a 25°C environment using fin free convection, a maximum of $(5.123)(10^{-3})(60-25)^{5/4}$ = 0.436 watts could be dissipated in the module. The computed thermal resistance is $R = \frac{60-25}{0.436} = 80.4$°C/watt. Actual thermal resistance values will be considerably lower, since natural convection from the module surfaces will also be significant. Nevertheless, the amount of dissipation per module allowable using this method is severely restricted, with 0.250 to 0.500 watts per module being typical. The design must ensure free circulation of the cooling air.

13.2.3.2.2 Forced air cooling of the module fins. The principles of forced air cooling developed in chapter 9 are applicable to analysis of systems employing SEMs with forced air system cooling. Typical system cooling arrangements are shown in Figure 175 and 176. Typical test thermal resistance values (fin-to-air) as a function of air velocity are given in Figure 177. Computation of the temperature rise of the fin over the air for a given module is relatively straightforward. A typical system enclosure, however, may contain on the order of 2500 to 3000 modules, with widely varying dissipation levels. Such systems use a series-parallel uniflow arrangement, whereby the system air is ducted into parallel paths, each serving a number of modules in series. For each module, the fin temperature rise over the cooling air temperature may be computed; however, the cooling air temperature for a given module will be determined by the system cooling air supply temperature, plus the temperature rise of the cooling air due to the dissipation of upstream modules, plus the temperature change of the cooling air if any, due to heat transfer to or from the duct-work. When several thousand modules are involved, computer solutions are recommended to obtain the "thermal map" of the system. Computer solutions also allow rapid evaluation of proposed changes, such as interchange of modules, or varying air flow rates. If the system thermal design progresses concurrently with the system electrical and physical design, as it should, a means of rapidly evaluating the effects of changes, such as a system computer thermal program, can prove invaluable.

13.2.3.2.3 Cooling by fin contact with a metal plate (cold plating). Shipboard system enclosures normally supply filtered cooling air, either by using filtered compartment air or air conditioning system air, or by employing a self contained closed system, with an integral air-to-water heat exchanger. Such systems allow the use of direct forced air cooling over electronic modules. Other air supplies (such as airborne systems utilizing ram air or engine bleed air for electronic system cooling) can contain relatively large amounts of entrained moisture, dirt, and other contaminants, and are not suitable for direct forced air cooling of circuitry. Cold plate techniques presented in chapter 9 are the recommended approach for this type application. This necessitates a low resistance thermal path from the module fin to the surface of the cold plate heat exchanger. This thermal path must additionally accommodate the variation in distance between the top of the cooling fin and the cold plate surface resulting from accumulative mechanical parts tolerances in the structure and assembly. This variation can be very significant in most typical constructions.

One approach to providing the required thermal path is by use of mechanical leaf or cantilever springs. These are commercially available items, or may be fabricated especially for a given application. Some typical designs are shown in Figure 178. All designs of this type offer at least three thermal resistances:

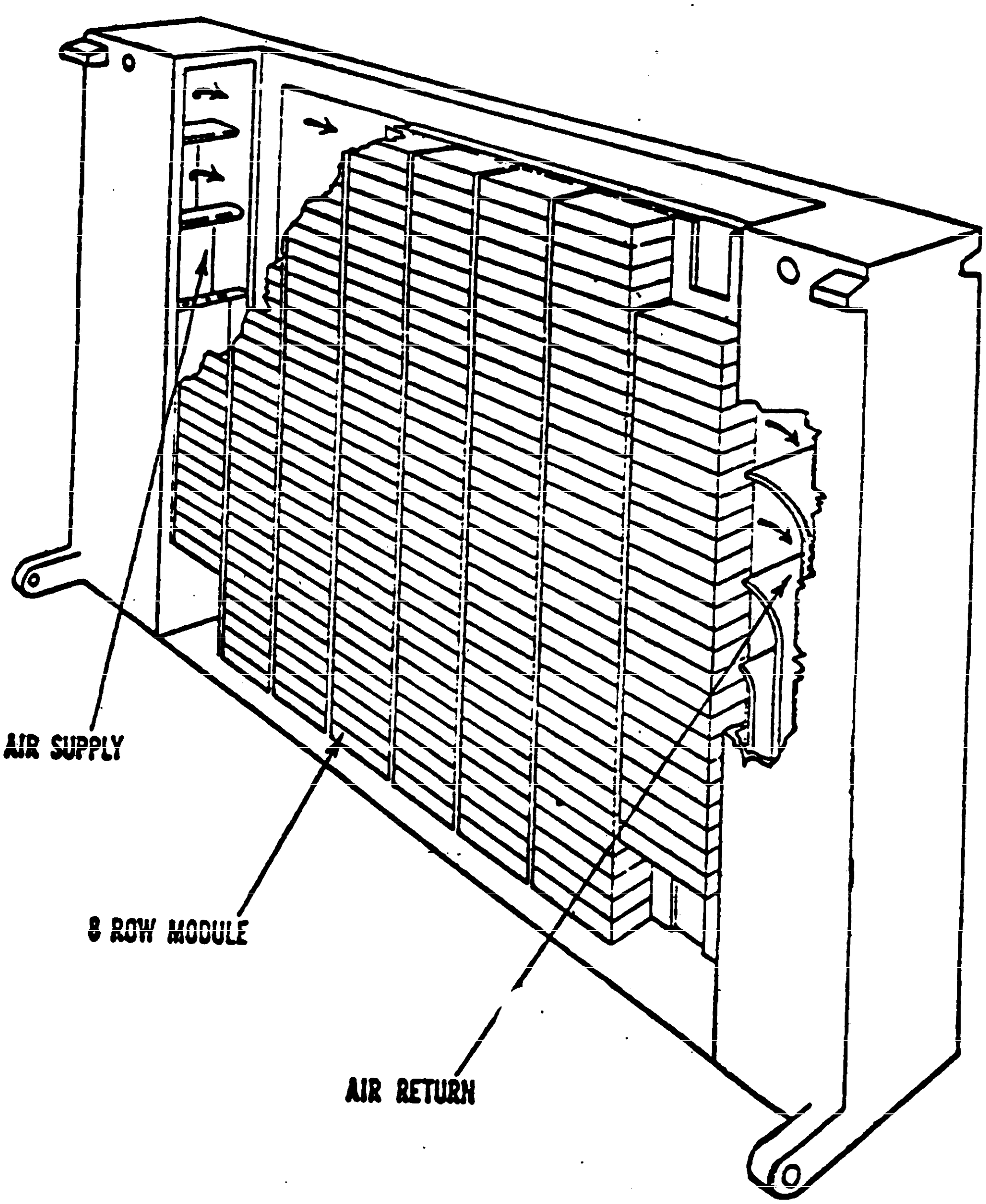


FIGURE 175. Typical System Cooling Arrangement

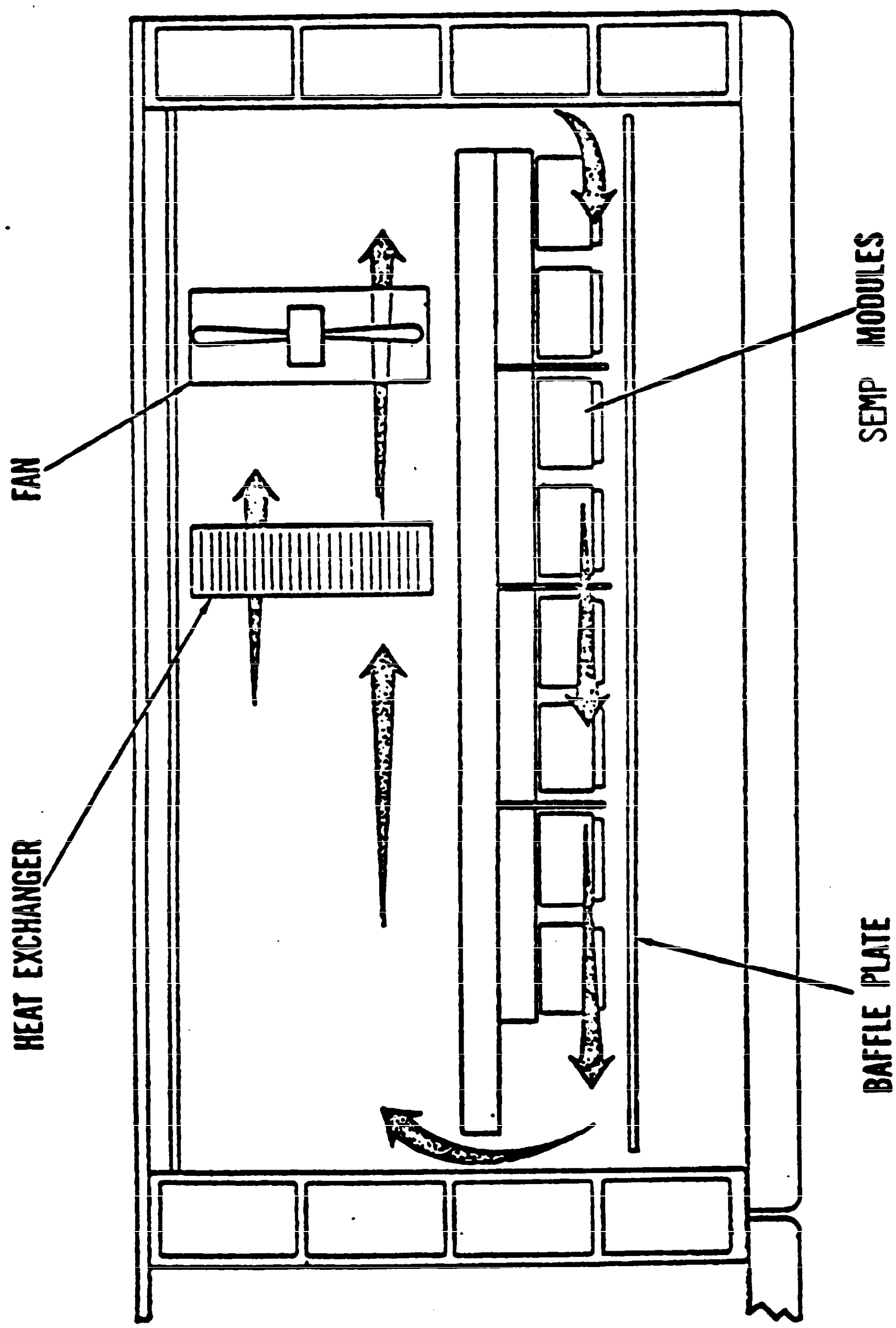


FIGURE 176. Typical System Cooling Arrangement

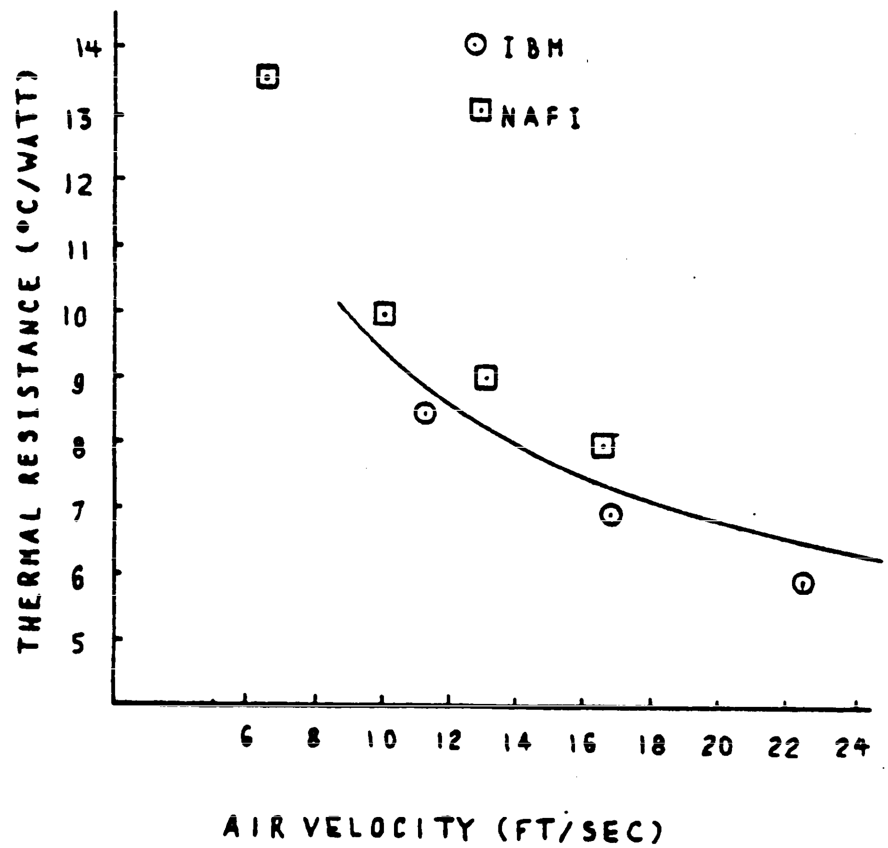


FIGURE 177. Thermal Resistance (Fin-Air) Cooling Air Velocity

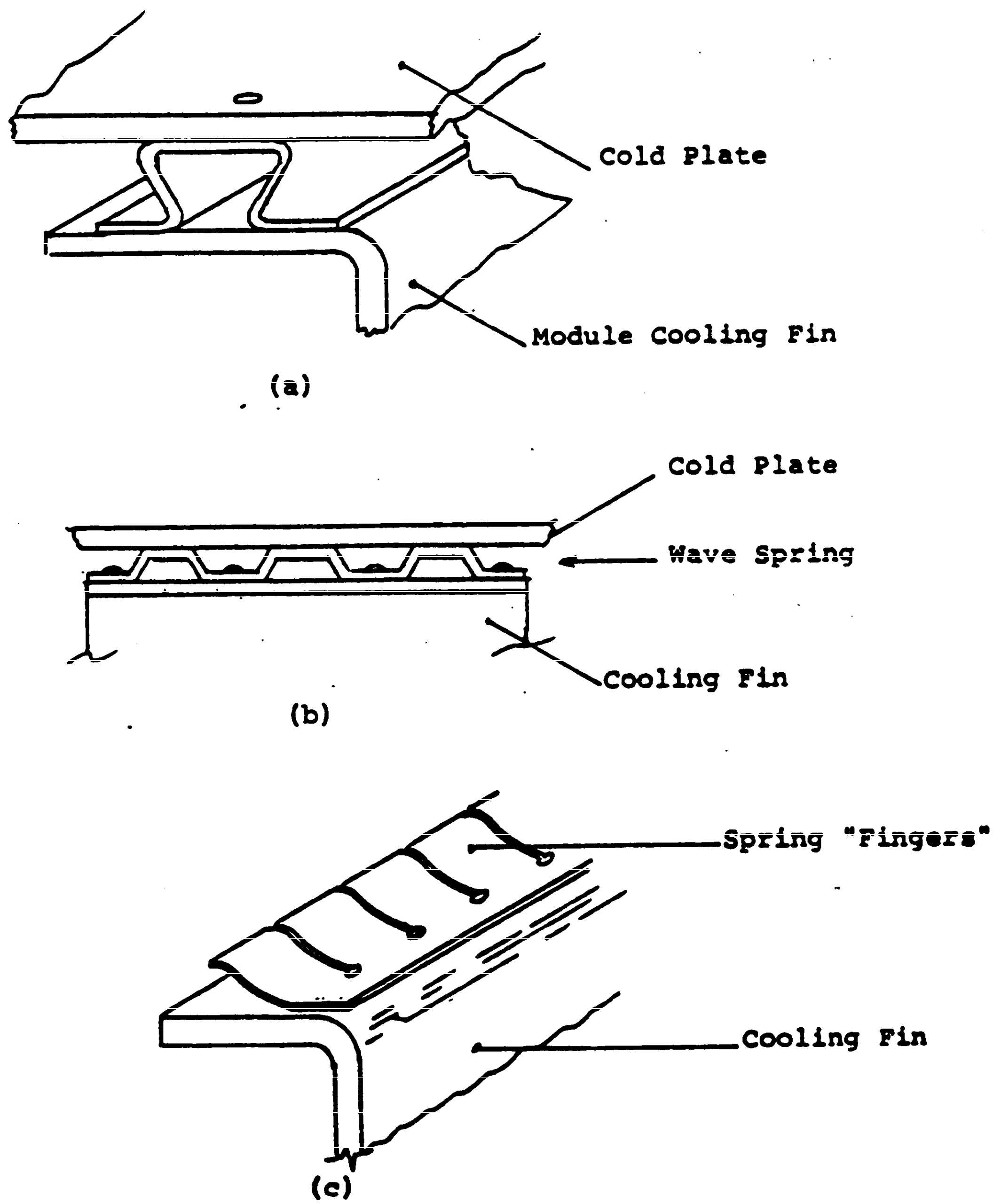


FIGURE 178. Typical Spring-Type Thermal Links

1) The interface resistance between the device and the module fin.

2) The conduction thermal resistance of the device.

3) The interface resistance between the device and the cold plate.

One of the interface resistances may be minimized by direct attachment of the device (welding, reveting, bonding, etc.), but the other interface resistance must be a contact resistance to allow for disassembly and for tolerance makeup. This contact resistance can be high, particularly if the contact pressure is low, or if the contact exists only along a line or a series of points. This latter is likely to occur unless extreme care is taken in alignment of parts. Thermally conductive grease-like compounds are helpful at the interface. The device conduction thermal resistance, like any conduction resistance, is proportional to the thermal path length, and inversely proportional to the thermal path cross sectional area and the thermal conductivity of the device material. Since the devices are likely to be made of thin stock material to achieve the required tolerance makeup without excessive forces involved, the cross sectional area is likely to be low, resulting in a relatively high thermal resistance. Well designed and properly applied thermal links of this type may display overall thermal resistances on the order of 15-20°C/watt. Poorly designed or poorly applied devices may exhibit thermal resistances of 50°C/watt or greater.

Another approach sometimes used to provide a thermal link is by use of wire mesh, similar to that commonly used for EMI shielding. Typically, however, a mesh dense enough to achieve a sufficiently low thermal resistance will be too "stiff" to accommodate tolerance variation, and conversely, a mesh which is sufficiently elastic will be so thin as to have an unusable thermal resistance.

A third approach is to utilize one of the conductive elastomers. These are elastomers formulated (usually in sheet form) with a high content of high conductivity filler, typically silver powder. Thermal conductivities may be on the order of 0.1 (watts/sq. in.)/(°C/in.). Stiffness may be varied within limits by the choice of the basic elastomer. Overall thermal resistances for an area typical of an SEM fin top surface is on the order of 20°C/watt. Some of these materials may be quite expensive.

Once a thermal link is designed from the module to the cold plate surface, there is, of course, an additional thermal resistance to the cooling air. The cold plate may be finned on the air side to increase the forced convection heat transfer area. Thermal resistance and air pressure drop in cold plates as a function of air velocity may be determined by use of the material presented in chapter 9.

13.2.3.2.4 <u>Cooling through guide ribs</u>. SEMs are designed such that they will be adequatly cooled if the interface temperature for their operating class is maintained at either the cooling fin or the guide ribs. Thus, the guide ribs offer an alternative heat removal path for the system designer. The guide ribs are used as a thermal path to the system chassis. The chassis in turn may be cooled by any of a variety of methods, such as natural convection, forced air, or liquid cooling; liquid cooling is probably the most common method of heat removal from the chassis when the guide rib cooling interface is predominant. The design methods presented in chapter 10 are applicable for computation of thermal resistances, temperature rise, liquid flow rates, and liquid pressure drops for forced direct liquid cooled chassis. Because of the typically low thermal resistance of a metallic chassis, and the high heat removal rate of direct forced liquid cooling, the chassis temperature may normally be considered uniform, at least in the localized area of a single module interface.

The major thermal resistance to be determined in such a system is that between the module guide rib and the structure. Spring retention clips are used to reduce the guide rib-to-chassis thermal resistance. A typical clip configuration is shown in Figure 179. The function of the clip is to force one side of the guide rib against the chassis plate. This rib-plate interface is the primary thermal path, as opposed to conduction through the clips itself. The overall thermal resistance is thus, considerably lower than that of the spring clips used for cold plating of the module fin as previously described, in which the clip provides the thermal conduction path. Measured thermal resistance of typical commercial clips are shown in Figures 180 and 181. Another series of tests indicated a thermal resistance of 3.8°C/watt for perfect guide rib contact, and 4.4°C/watt for a controlled module (1A) skew of 0.018 inch. A thermal resistance value guide rib-to-chassis, of between 4 and 5°C/watt appears reasonable. Some variation may occur with use and wear, but the wiping action of the guide rib to the chassis on removal or insertion of the module should assure relatively consistent values. To this thermal resistance, there must be added the resistance through the chassis to the liquid coolant passage, and the forced convection resistance into the coolant. These, of course, will vary with the particular chassis construction but typically amount to 0.5 to 1.0°C/watt.

Module cooling through the guide ribs into a forced liquid cooled chassis provides the advantages of:

1. Generally greater heat capacity per module.

2. Absence of contamination.

3. Acoustically quiet.

4. Simple filtering and little maintenance (at electronic enclosure).

5. Generally higher module component density.

These are offset by the disadvantages of:

1. More complicated chassis construction due to liquid cooling requirements.

2. More complicated distribution system. Liquid cooling lines must not prevent servicing and maintenance of electronics.

3. More complicated chassis removal and/or replacement, due to liquid cooling lines.

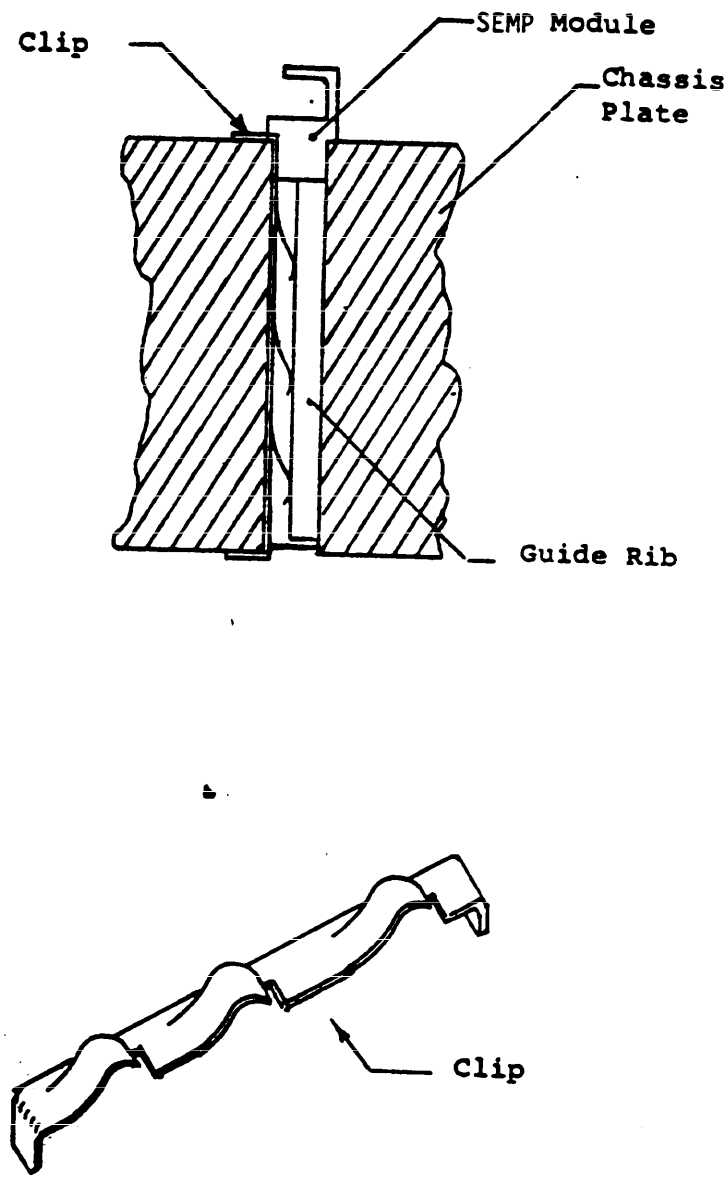


FIGURE 179. Typical Clip Installation and Configuration

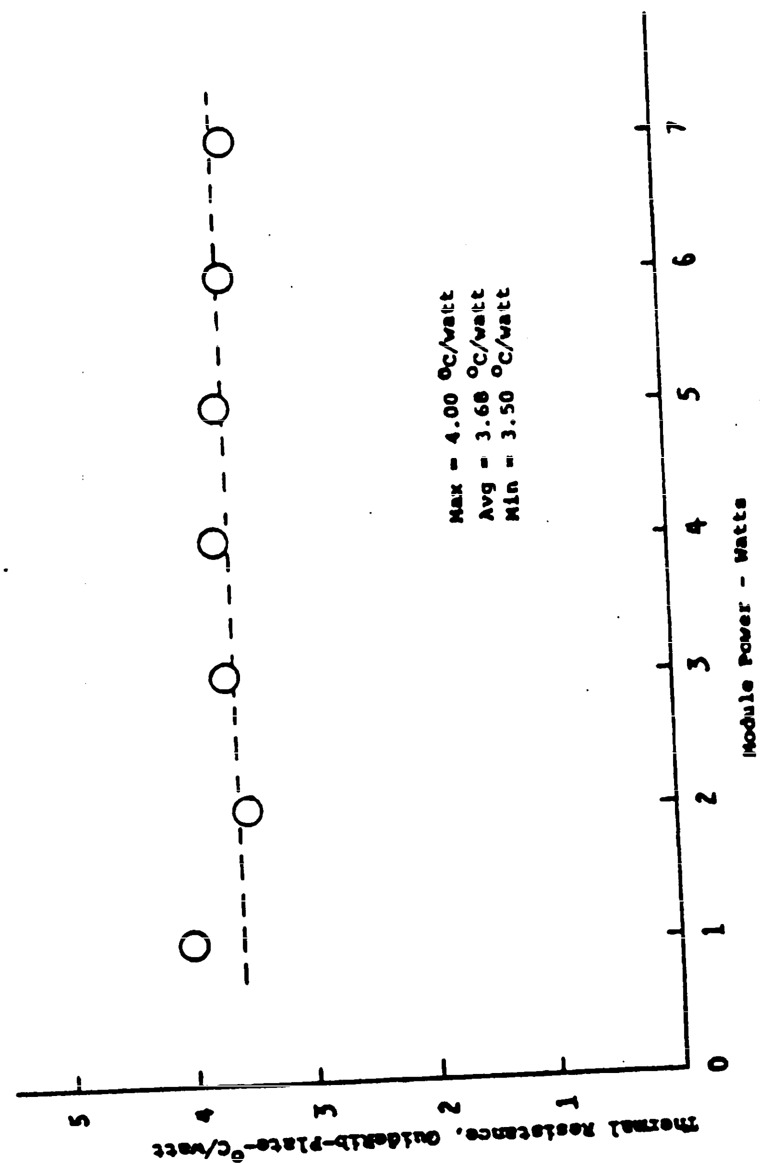


FIGURE 180. Thermal Resistance vs. Module Power
IERC Retention Clip Test Data

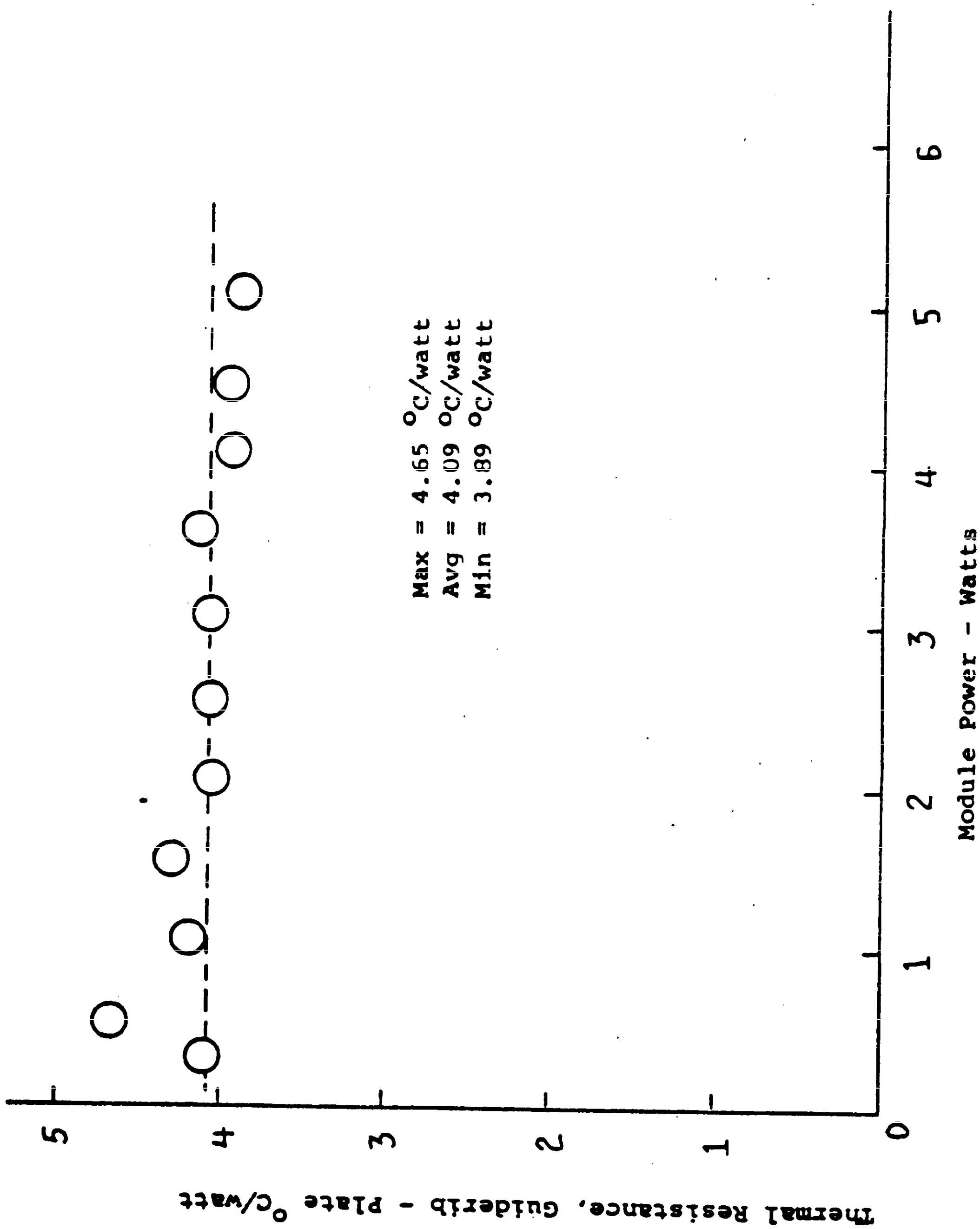


FIGURE 181. Thermal Resistance vs. Module Power Birtcher Retention Clip Test Data

4. Dependence on source of liquid coolant, with possible deleterious effects due to poor maintenance or fouling of heat exchangers.

5. Possibly increased weight.

NOTE: The guide rib-to-chassis thermal resistance is normally given in terms of a module, and represents the combined effects of both guide ribs, not the individual resistance of two guide ribs in parallel. Thus, for example, with a module dissipating 3 watts, and with a given guide rib-to-chassis thermal resistance of 4°C/watt, the guide rib temperature will be (3)(4) = 12°C above the adjacent chassis temperature.

13.2.3.2.5 Combined cooling methods. The preceeding paragraphs on system thermal design emphasized one particular method of heat removal (for example, forced air cooling of the module fin, or cooling through the guide ribs to a liquid cooled chassis). In a given system, even though one cooling method may be predominant, some heat may be dissipated through other paths. Thus, in a system using heat transfer through the guide ribs into a liquid cooled chassis, there will still be a natural convection process occurring at the fin. Likewise, in a system employing forced air cooling of the fin, there will be conductive heat transfer through the guide ribs to the structure. These extraneous heat paths (other than the primary design method) should be ignored in system design. They will normally be negligible, or at most, transfer only a small percentage of the total module heat. Ignoring them will add a slight degree of conservatism to the design.

Similarly, modules should not be relied upon to transfer heat into adjacent modules. It is true that a module dissipating significantly more heat than its neighbors will transfer heat to them by radiation and convection in the intervening air gap, and by conduction through the guide ribs and chassis structure. However, this is an uncontrolled thermal path, and subsequent design changes to the neighboring modules could have a serious effect on the high dissipation module. All module dissipation should be designed to be removed through the primary heat transfer path (or paths). On the other hand, it may be necessary to consider this heat transfer in terms of the thermal load on the adjacent modules. If they are of low dissipation, a significant portion of the thermal load may be due to the effects of the high dissipation module. In extreme cases, it may be necessary to provide high conductivity radiation shields (which must be thermally attached to the chassis) between modules.

There are instances wherein more than one thermal path from a module may have to be utilized to maintain acceptable component temperatures in high dissipation modules. These modules are often identified as SEMP Special Modules, i.e., modules conforming to SEMP standards, having completed and approved SEMP SCD's, and design qualified, but not appropriate for Navy-wide inter-system usage. Such a module may, for example, require forced air cooling of the fin plus conduction cooling through the guide rib to the chassis (which may or may not be liquid cooled), so as to be adequately cooled without excessive air velocity over the fin, with associated high noise levels and high blower pressure requirements. In such applications, both the cooling fin and the guide rib temperatures may exceed the interface temperature for the module class, and still result in sufficient cooling. Such designs, however, must be substantiated by detailed thermal analysis, including not only the system thermal resistances, but also the internal module thermal resistances. This is due to the parallel heat removal paths. Verification of the design by thermal testing of preliminary mockups is highly recommended.

Design Example

A system has been designed (with SEMP) for forced air fin cooling. Heat exchanger design necessitates operation with cooling air at 40°C entering the system. Cooling air velocity over the module fin is 18 ft./sec. Thermal resistance vs. airflow is in accordance with Figure 177. A special module within the system has the following characteristics:

Dissipation: 5 devices, 1 watt each
Junction-to-module frame thermal resistance = 35°C/watt
Frame-to-fin thermal resistance = 5°C/watt
Maximum allowable junction temperature (CCT) = 125°C (Class I operation)

The thermal circuit is shown in Figure 182, where

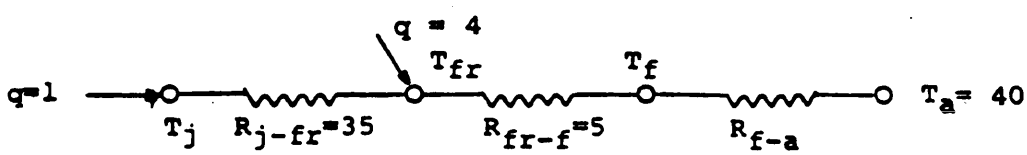


FIGURE 182. Thermal Circuit-Design Example

q = dissipation, watts
T_j = junction temperature, °C
T_{fr} = frame temperature, °C
T_f = fin temperature, °C
T_a = air temperature, given as 40°C
R_{j-fr} = thermal resistance, junction-to-frame, °C/watt
R_{fr-f} = thermal resistance, frame-to-fin, °C/watt
R_{f-a} = thermal resistance, fin-to-air, °C/watt = 7.5 @ 18 ft./sec. air velocity.

Solution of the circuit yields:

T_f = 40 + (7.5)(5) = 78.5°C (exceeds Class I allowable)
T_{fr} = 78.5 + (5)(5) = 103.5°C
T_j = 103.5 + (1)(35) = 138.5°C (exceeds CCT)

The air velocity limitation prevents achieving an acceptable fin temperature. Assume that R_{fr-gr}, the thermal resistance from the frame to the guide rib is equal to R_{fr-f}, the thermal resistance from the frame to the fin. Assume further that retention clips yielding a module resistance of 5°C/watt are added to the guide ribs. Finally, assume that the chassis temperature is maintained at 50°C. The thermal circuit, with appropriate values, is shown in Figure 183.

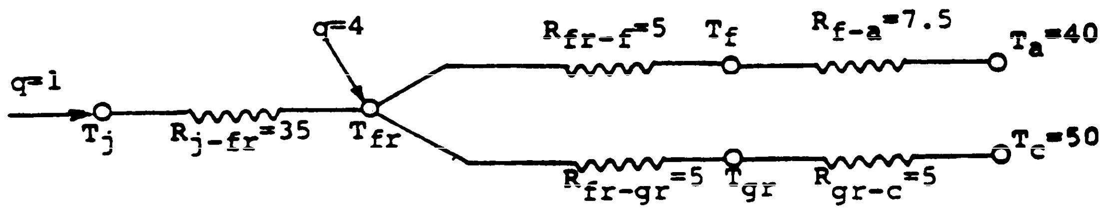


FIGURE 183. Thermal Circuit, Combined Cooling Paths

Solution of this circuit yields:

$$(T_{fr} - 40) = (5 + 7.5)q_A$$
$$(T_{fr} - 50) = (5 + 5)q_C$$
$$q_A + q_C = 5$$

From which $q_A = 2.4$ watts
$q_C = 2.6$ watts
$T_{fr} = 76°C$

Also, $T_f = 40 + (7.5)(2.4) = 58°C$
$T_{gr} = 50 + (5)(2.6) = 63°C$
$T_j = 76 + (1)(35) = 111°C$

Thus, the addition of a conductive path through the guide ribs yields acceptable junction temperatures. Of course, the assumption that the chassis is maintained at 50°C while heat is being transferred to it must be verified.

14. EQUIPMENT INSTALLATION REQUIREMENTS AND CONSIDERATIONS

14.1 General. The thermal interface between electronic equipment and the environment in which the equipment operates is extremely important. Equipments with adequate thermal designs can overheat when installed improperly or in a thermal environment which differs from that the equipment was designed for. This has been a particularly severe problem on shipboard.

14.2 Anticipated and actual thermal environments. The anticipated thermal environment is that which is expected to exist around equipment and that which equipment is usually designed to operate in. This includes, on shipboard for example, the local air temperature of the space the equipment is installed in, and that of the fresh water coolant, if used.

The actual environment in which the equipment is installed should not significantly differ from the anticipated environment, otherwise a thermal mismatch will occur. Unfortunately, this mismatch has lead to serious equipment overheating. Typical examples are:

14.2.1 Equipments designed for natural cooling of all external surfaces are intended for isolated operation; i.e., sitting alone on a desk or bench. When such equipments are installed in a cabinet along with other equipments, the natural cooling heat flow paths the equipment was designed for are modified. The air temperature in the cabinet increases, often to excessive values, and additional thermal resistances are introduced. Result - the equipments overheat! Supplemental cooling must be provided.

14.2.2 Small forced air cooled equipments are stacked side by side or one above the other so that the hot exhaust air from one unit enters the cool air intake of another unit. Second, third, or fourth handed air will have a temperature rise far above that of the anticipated environment. Also, equipments should not be installed so that the air intakes and exhausts are partially "blocked."

14.2.3 When many equipments are installed in a given space and collectively reject their heat into the space, the space can become overheated. The air conditioning systems, which are often designed only for personnel comfort heat loads, become overloaded. This is particularly applicable to ships, vans, and small buildings.

14.2.4 Equipments should not be installed in humid locations where moisture can collect on the electronic parts and wiring. The equipment should not operate at temperatures below the dew point of the air.

14.2.5 The installation of equipments in thermal environments which are obviously more severe than the anticipated thermal environment must be avoided. For example, equipments should not be installed in poorly ventilated small compartments (including those with steam lines), alongside of ships funnels or steam lines, adjacent to jet engines, or other major heat sources.

14.3 Provisions for the removal of heat.

14.3.1 The ships ventilation and cooling systems must include provisions for at least marginal cooling under certain emergency or battle conditions. Vital equipments must function even though portions of the cooling system may be damaged or inoperative. For example, in closed loop forced air to fresh water shipboard cooling systems, when the water system is inoperative, provision must be made for the circulation of room or cabin air through the equipment, using the existing blowers or fans in conjunction with emergency air inlet and exhaust openings in the cabinet. Under these conditions cool air, preferably in excess of that normally supplied, must be provided.

14.3.2 The use of supplemental deck mounted air conditioners to provide increased cooling to overheated spaces aboard ships is basically a "fix" of a local thermal mismatch. Certainly, if most spaces were provided with such cooling systems, the decks would be overcrowded with air conditioners. The ship's basic cooling systems must be designed to match the thermal requirements of the electronics.

14.3.3 The thermal adequacy of the installations on ships should be determined by tests. All the equipments in a given space should be operated simultaneously and the temperatures and flow rates of coolant should be measured under environmental conditions approaching the maximum anticipated. Only installation tests and measurements can prove that a satisfactory thermal match exists. (See chapter 15 for details of thermal evaluation.)

14.3.4 The cooling systems of each equipment must be checked for proper operation prior to installation. Thermal problems have been traced to defective cooling systems.

14.4 Installation aspects for maintainability. Provision must be included in the equipment installation for adequate cooling during maintenance and test periods. For example, adequate cooling must be provided when drawers or chassis are withdrawn from cabinets, when chassis are bench tested, when the sides of cabinets are removed, or when avionics equipment is serviced in a hanger or on a flight deck. The lack of or inadequacy of cooling during maintenance periods under conditions such as the aforementioned has led to serious premature failures due to thermal overstress, even though the time periods were relatively short. The installation thermal design must include consideration of the redistribution of heat during equipment maintenance.

The installation must not impair rapid and easy maintenance of the equipment or its cooling system.

Easy access to air and other coolant filters is extremely important. Equipment has overheated too often as a result of dirty air filters and blower scrolls.

Heat exchangers, especailly forced air to fresh water shipboard exchangers, must be readily accessible for removal and cleaning.

Fresh water systems for cooling electronic equipment must be designed and installed for ease of maintenance. Water filters, flowmeters, deionizers, etc., must be accessible.

14.5 Aircraft equipment and cooling systems interfaces.

14.5.1 In general, the cooling systems in aircraft are integrated and matched to the thermal requirements of the equipments to a much greater degree than on shipboard. However, there is also a need for thermal improvement.

14.5.2 The supply of coolant to electronic equipments is for short periods often much less than specified. In aircraft the flow of jet engine bleed air used for cooling can be drastically reduced during a high speed dash and the thermal problem is compounded by the increased aerodynamic heating of the aircraft. In some aircraft, the flow rate and temperature of the available cooling air matches that specified as being available for equipment cooling under ideal flight conditions only.

Aircraft manufacturers carefully ration the supply of coolant to electronic equipment because of the penalties in aircraft performance and weight involved. Decisions in these matters have been arbitrary. Tradeoff studies are recommended, particularly in view of recent changes in policy and philosophy towards trading performance for reliability. In general, the supply of coolant or aircraft is marginal and the electronic parts are stressed near their maximum levels.

14.5.3 Minimization of moisture condensation is important, especially in airborne equipment. The wide variation in altitude, temperature, and humidity encountered during a mission often results in moisture laden equipment. If possible, indirect forced air cooling should be used.

14.5.4 Thermal mismatches also occur in aircraft with standard electronic equipments which are common to many aircraft. Units housed in standard AIRINC cabinets should not be mounted in locations where the local thermal environment (especially altitude aspects) exceeds that the unit is designed for.

15. THE THERMAL EVALUATION OF ELECTRONIC EQUIPMENT

15.1 General. The testing of electronic equipment to determine the adequacy or effectiveness of the cooling system is not expensive or difficult but does require careful measurement of coolant temperature and flow rate. Parts temperature measurement is not individually difficult, but with many parts can become somewhat of a chore, particularly if thermocouples are used. However, other techniques are available and these are discussed later in this section.

It is important to recognize that parts surface temperature is the major consideration in the thermal evaluation of an equipment. The purpose of the cooling system is to cool the parts and thus parts temperature is the fundamental basis for evaluating the adequacy of a design.

This chapter also includes detailed information on the measurement of fluid (and gas) pressure and flow rate which is particularly pertinent to forced air and liquid cooling techniques. Some of this information relates closely to design activities and the remainder to test and evaluation. Rather than scatter the fluid flow information throughout this handbook, it is assembled into a single section at the end of this chapter.

The material presented in this chapter is directed towards the thermal evaluation (testing) of electronic equipment, not inspection. For example, infra red radiometers or pyrometers are of limited use in the thermal evaluation of electronic equipment, since the radiation from heat producing parts cannot be seen through equipment covers, and doors, etc., and individual parts are usually within a tightly packaged assembly of subassemblies. On the other hand, an optical pyrometer or radiometer is an excellent device to inspect, for example, a single printed circuit card to observe temperature distributions. Defective parts and other malfunctions can be readily detected with IR sensitive devices because the defects cause an abnormal temperature distribution.

15.2 Thermal indices and criteria. It is often desirable, in the interest of simplicity and economy, to determine the condition of an equipment by performing only one or several temperature measurements. This can be accomplished by measuring the temperature at one or several key locations in an equipment. The temperatures of these key locations are an index of the overall temperature distribution within the equipment, since all of the parts are thermally connected to each other by relatively constant thermal resistances. Thus, a convenient thermal index of the operating condition of an equipment can be established. However, the complete temperature distribution in the equipment must be initially determined by measurement to ascertain the actual thermal resistances between the parts. Also, the key locations must be carefully indentified. Usually the surface temperature of the parts having the highest thermal stresses can be used as thermal indices.

15.3 Methods of temperature measurement (from reference 29).

15.3.1 Accuracy desired. Whether or not the design of an electronic equipment meets the thermal specifications will be indicated largely by temperature measurements. Such temperature measurements fall into two general classes: (1) the temperature of a fluid (gas, vapor, or liquid), either still or in motion, and (2) the temperature of solid bodies, either at the surface or within the body. Examples are the temperature of the air within an enclosure, the surface temperature of the envelope of an electronic tube or a transistor. Each class requires a different measuring technique depending upon the desired accuracy.

A high degree of accuracy in measuring temperature is difficult to attain. The degree of accuracy required depends on the purpose of the measurements. For example, in experimentally determining the film coefficient of heat transfer of a fluid, an accuracy of $\pm$ 0.5°C may be insufficient. Here precise temperature measurements must be made. On the other hand, an accuracy of $\pm$ 2 to 3°C is adequate in determining whether an electronic equipment meets the specifications. In general, it appear reasonable to expect such accuracy in test work on electronic parts, subassemblies, and assemblies.

15.3.2 Temperature indicating devices. Of the many types of temperature measuring devices, the thermocouple remains the most practical since it is simple, rugged, and accurate. The most common method of employing the thermocouple is in conjunction with a sensitive potentiometer which can be either of the indicating or recording type. Such potentiometers are standard items available from several instrument manufacturers. Several excellent digital temperature indicating instruments are now available.

Other devices or instruments of value are the glass thermometer and temperature indicating paints and waxes. It has been found that most all other devices such as optical pyrometers, thermistors, radiometers, and resistance thermometers are relatively impractical for the type of temperature measurements required in the testing of electronic equipment. Certain applications may arise wherein use of these devices may be advantageous.

15.3.2.1 Thermocouples. The theory of thermoelectric thermometry is well known. Hence, this subject will not be discussed in detail although general considerations and certain techniques will be considered later. It is important to note that thermocouples must be calibrated. This is discussed in detail later in this chapter.

There are many types of thermocouples but, for the evaluation of electronic equipment either the iron-constantan or copper constantan thermocouple is adequate. Iron-constantan thermocouples are generally used below 760°C (1400°F) and copper-constantan below 350°C (662°F). The range of either of these two thermocouples is indicated by the foregoing maximum temperatures is adequate for electronic test work. Copper-constantan has the advantage of being much more resistant to corrosion. The iron in iron-constant thermocouples rusts when exposed to moisture which may change the thermoelectric characteristics. When used with sensitive indicating potentiometers, or digital indicators, these thermocouples provide high accuracy of temperature measurement.

relatively small amount of vapor than ... ture. The low temperature liquid and r

15.3.2.2 Glass thermometers. The use of the liquid-in-glass thermometers in electronic testing is confined to the measuring of fluid temperatures and the checking of other temperature devices such as the thermocouple.

15.3.2.3 Temperature indicating paints and waxes. Temperature indicating paints and wax-like substances are available and cover a wide range of temperatures. In general, these devices are of two classes: (1) paints which exhibit a color change at a given temperature and (2) wax-like substances which melt at specific temperatures. The paints may have as many as four color changes, each color change indicating a different temperature. In addition to its use as a temperature indicator, these paints can be used to show the temperature or heat flow pattern over an area.

A wide assortment of the wax-like substances which melt at specified temperatures is available. They can easily be applied to a surface, and, upon application of heat, will melt if the specified temperature is exceeded. Upon cooling the liquid smear solidifies but has a different appearance than the original wax coating. Small paper or plastic tabs with adhesive backing and temperature indicating paints are also available.

15.3.2.4 Other devices. Resistance thermometers are widely used to accurately measure temperature. Their use largely parallels that of thermocouples and they are subject to the same errors such as radiation effects. They are not readily applicable to measuring surface temperatures (with the exception of thermistors). It appears that their use is relatively limited. In view of the ease of use and application, and the wide spread knowledge of thermoelectric thermometry, it is recommended that thermocouples be used rather than resistance thermometers.

Thermistors, or thermally sensitive resistors, have been used as thermometers. Their application follows the same principles of resistance thermometry but their large value of temperature coefficient permits a new order of sensitivity to be obtained. The thermistor can be made extremely small to reduce its heat capacity so as to rapidly follow changing temperatures. The main disadvantage of thermistors is that each must be carefully calibrated after having been well aged.

IR scanning systems to provide thermal profiles of unpackaged I.C.'s as well as larger assemblies such as P.C. boards are also available.

Optical pyrometers which measure the infrared radiation from a heat source are useful in special situations. Optical pyrometers with fields of view as small as 0.0005" spot diameters with accuracies of $\pm$ 0.5°F are available. These devices are excellent for laboratory measurements such as the temperature distribution on the surface of unpackaged IC's.

Optical pyrometers suffer two major disadvantages, namely: (1) the emissivity of the surface being measured must be known accurately and (2) the surface must be visible. Unfortunately, it is extremely difficult to "see" the parts in a densely packaged electronic equipment. Optical pyrometers and radiometers are discussed in a later section.

15.3.3 Thermocouple applications.

15.3.3.1 General considerations. A thermocouple consists of two dissimilar metal wires joined togehter at one end by soldering, welding, or twisting. The junction thus formed is termed the "hot junction." If the remaining two ends of the wires are joined together to form the "cold junction," and the two junctions are maintained at different temperatures, an electric current will flow in the closed circuit. This could be indicated by connecting the free ends of the wires to a voltmeter in which case the voltmeter terminals and/or wiring becomes the cold junction. The generated electromotive force is a function of the metals used and the temperature difference of the two junctions. Due to the latter cause, the cold junction must be maintained at a constant known "reference temperature," usually 0°C, by inserting the cold junction in an ice bath. An alternate method is to include the cold junction within the instrument, usually a sensitive potentiometer. In this instance the reference temperature becomes that of the instrument (room air temperature). Since the room temperature is seldom constant, means must be provided to compensate for changing reference temperature. Temperature indicating digital potentiometers are available with either manual or automatic compensation, the latter being preferable. For greater accuracy, the ice bath should be used for the cold junction and no compensation is required. Potentiometers may be calibrated either in millivolts or directly in degrees C or F. In the former case, various metals can be used for the thermocouple since the reading is in millivolts and a calibration table for any particular thermocouple can be used to obtain the temperature. Potentiometers calibrated in temperature degrees are for only one type of thermocouple and thus, for example, an iron-constantan couple cannot be used with a potentiometer if it is calibrated on the basis of copper-constantan.

Figure 184 shows the arrangement of thermocouple and potentiometers wherein the instrument includes the cold or reference junction. If an ice bath is used as the reference junction, the arrangement is as shown in Figure 185.

If the thermocouple wire is homogeneous (as it should be), temperature gradients in the wire itself do not affect accuracy of reading. It is important to realize that only the temperature of the junction or bead affects the reading.

15.3.3.2 Calibration. For accurate temperature measurement, thermocouples should be calibrated, or thermocouple wire should be purchased from a manufacturer who guarantees the wire to follow a standard calibration. This is especially important for iron-constantan thermocouples since it is difficult to control the uniformity of the iron composition. There are two general methods used to calibrate thermocouples, one being calibration at fixed pointed such as the melting point of ice, boiling points of water and naphthalene, and the freezing points of tin, lead, and zinc. A second method is by comparison with a standard thermocouple whose calibration curve is known. This method is sufficiently accurate for the temperature measurement of electronic parts. The former method, that of calibration at fixed points, involves special furnaces and considerable skill and is not recommended. In general, thermocouple wire as small as #30 gage (0.010 inch diameter) can be obtained which will follow standard calibration data with sufficient accuracy. However, it is recommended that at least two standard calibrated thermocouples be purchased so that thermocouple wire may be easily and quickly checked by the comparison method. Such thermocouples would be used only as "laboratory standards" for calibration purposes.

If thermocouple wire is purchased in significant quantities, such as by the roll, the manufacturer usually supplies calibration data. Alternatively, since the wire is all from the same run, the calibration need only be made at several temperatures (against the standard) for one sample thermocouple from each roll. Thermocouples can be fabricated from the wire on the calibrated roll and need only be checked at one temperature. For example, when a group of thermocouples have been fabricated and installed as instrumentation in an equipment; while the equipment is still cold prior to energization, if all thermocouples read room temperature within $\pm$ 0.5°C, the thermocouples are satisfactory.

Thermocouple wire smaller than #30 gage should be calibrated by the user. Since heat transfer theory indicates that, decreased wire size increases accuracy due to such effects as minimizing heat transfer from a surface through the leads, small wire size should be used. Further, temperature measurements of very small parts require fine wire size, #36 gage or smaller.

A variety of insulation for thermocouple wires is available. Duplex wire with glass insulation has been found to be satisfactory. This insulation is heat resistant and mechanically strong and its use is recommended. Since bench testing usually involves relatively short wires, it is not recommended that independent lead wires be used. Also inductive coupling into radio frequency circuitry will be minimized. Use of calibrated thermocouple wire for both thermocouple and lead wire makes for maximum accuracy and use of the cheaper lead wire is not recommended.

15.3.3.3 Methods of forming the junction. Any method of fabricating the junction which produces a good clean contact without overheating (oxidizing or burning) the thermocouple leads may be used. Examples are electric welding, low temperature brazing and soldering. Overheating and strong blows with a hammer may cause a change of fine structure and result in a shift in the calibration of fine thermocouple wires. A satisfactory junction can be made by twisting and soldering the cleaned wires. Soldering the leads is recommended to insure a metallic bond between the two wires. Silver solder must be used for elevated temperature since common solder can be used only up to about 150°C. Further, the use of low temperature solders using bismuth and antimony alloys is to be avoided. At elevated temperatures these materials may soften slightly to form an electrolytic cell instead of a thermocouple. The tip of the junction should be clipped off so that the actual junction is very short.

15.3.3.4 Thermocouples for measuring liquid temperatures. No difficulty is encountered in measuring temperatures of liquids since radiation effects are usually negligible. Hence, a thermocouple immersed in a liquid should give an accurate measure of the temperature. The length

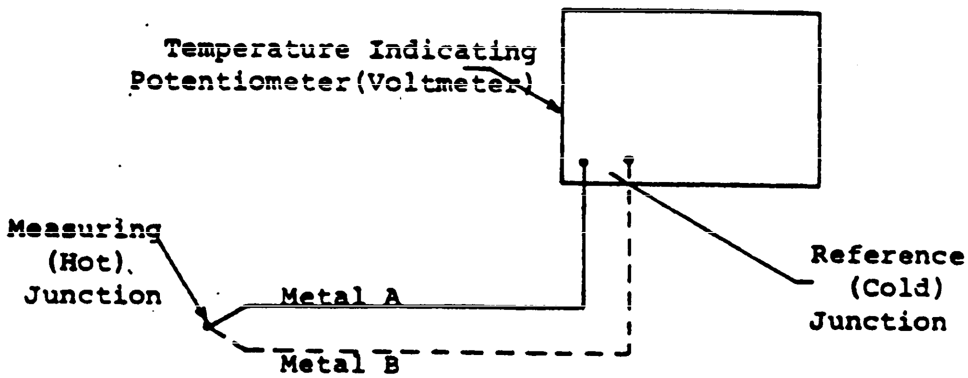


FIGURE 184. Thermocouple Circuit With Instrument Terminals Serving as Reference Junction

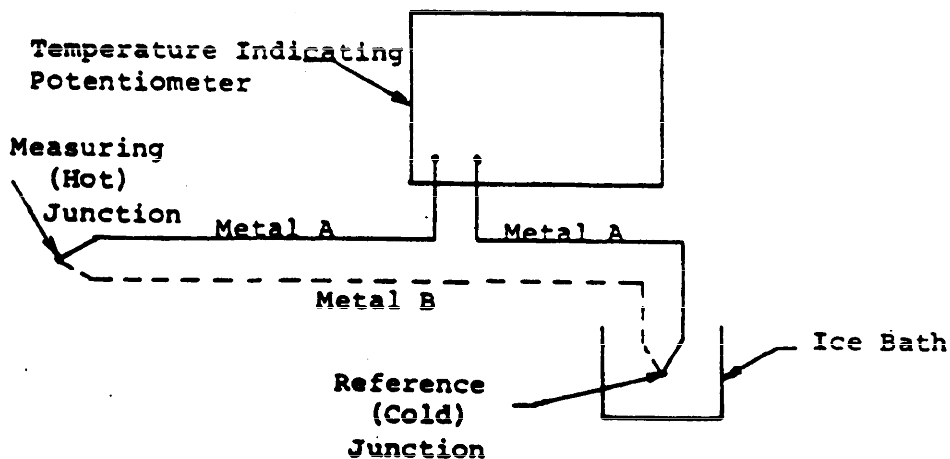


FIGURE 185. Thermocouple Circuit With Reference Junction in an Ice Bath

of lead wire immersed in the liquid should be long to minimize conduction through the wires. Thermocouples are insensitive to mechanical and thermal shock.

15.3.3.5 Thermocouples for measuring gas temperatures. In measuring the temperature of gases, radiation from surfaces which the thermocouple "sees" may cause a thermal effect which results in considerable error. If a thermocouple is placed within an enclosure to meausre the air temperature and the thermocouple "sees" a surface at a different temperature than the air such as a wall whose temperature is lower than the air, there will be a net exchange of radiant heat from the couple to the walls. Since the couple reaches thermal equilibrium with its surroundings, the couple will indicate a lower temperature than the true air temperature. The couple gains thermal energy from the air by convection at the same rate that it radiates to the walls. Convection implies a temperature difference and the couple will assume a temperature between the true air temperature and that of the wall. The magnitude of the error will depend on the temperature difference (deficiency or excess) between the air temperature and that of the surface which the couple "sees," the character (radiationwise) of the radiating surfaces (both couple and wall surfaces), and the relative areas and configurations of these surfaces. For example, a thermocouple with junction diameter of 0.02 in. and emissivity of 0.20 is placed in an enclosure containing still air at 127°C, with the temperature of the enclosure walls at 82°C. When the couple attains thermal equilibrium, it will indicate a temperature of about 124°C or 3°C below the true air temperature.

The difference between the true air temperature and that indicated by the thermocouple can be decreased by two methods: (1) decreasing the diameter of the thermocouple junction, and (2) decreasing the emissivity of the thermocouple wire. Decreasing the diameter of the junction or bead increases the film coefficient of free convection which results in a decreased temperature difference for the same heat flow per unit area. Decreasing the emissivity of the couple decreases the net heat transfer by radiation. This can be accomplished by either brightening the junction and the adjacent wire surface or by shielding the couple with one or more shields of highly polished metal cylinders. The following table shows the influence of these two factors on the error caused by radiation.

TABLE XXXIV.

Example of the Influence of Junction Diameter & Emissivity on the Error Due to Radiation

Air Temperature 126.7°C, Wall Temperature 82.2°C

Junction Dia.-in	Wire Emissivity	Temp. Indicated by Thermocouple	Approx. Error
0.05	0.10	123.9°C	2.8°C
0.03	0.10	124.7°C	2.0°C
0.02	0.10	125.3°C	1.4°C
0.02	0.20	123.9°C	2.8°C
0.01	0.10	125.8°C	0.9°C
0.003	0.10	126.4°C	0.3°C

Table XXXIV gives representative values only and illustrates the probable order of magnitude of the error in temperature measurement caused by radiation for the specified condition. The errors caused by high temperature heat sources such as electron tubes could be considerably greater. The values given were computed using appropriate convection and radiation equations.

If the enclosure contains electronic parts such as transistors and resistors of various emissivities and at various temperatures, then a bare thermocouple would "see" and exchange radiation with such parts as well as the enclosure walls. Here, it is very desirable to use a shielded thermocouple to obtain the temperature of the gas within the enclosure.

There are many designs of radiation shields. The literature indicates that best results for shielded thermocouples in still air have been obtained with shields approximately one inch or greater in diameter. It has been found that if the shield is too close to the couple, the error may be greater than that for a bare couple due to decreased free convection. A shield consisting of two concentric cylinders with an inner diameter of one inch and an outer diameter of two inches, the cylinders mounted in the vertical position, gives quite accurate results. This type of shield is applicable for use only in relatively large spaces or containers. For small containers or boxes, the size of the shield would prohibit its use. Hence, some other means must be used in very small spaces.

There is an alternate method of temperature measurement using unshielded thermocouples which has been used by several investigators. Since the film coefficient is an inverse function of the diameter, the smaller the thermocouple junction, the closer will the couple indicate true air temperature. The method consists of making temperature readings with at least three thermocouples made of three different junction diameters. The temperature readings are then plotted against its diameter. The true air temperature is then estimated by extrapolation to the zero diameter. This method is claimed to be more accurate than most of those employing radiation shields. The multiple bare couple method lends itself well to measuring the temperature of the interior of a small box or any space wherein a shield cannot be used.

15.3.3.6 Thermocouples for measuring surface temperatures. One of the most accurate methods of measuring surface temperature is to embed the thermocouple in a slot or scratch in the surface and seal it flush with the surface with cement. The wire size should be small, say #36 gage, and the length of lead embedded should be at least one-half inch. This method has the disadvantage of restricting its use to relatively large surfaces and to surfaces which can be marred or scratched. Further, the application in itself requires skill and considerable time.

The most common practice has been to place the junction and a short length of lead wire on the surface and to apply a small piece of adhesive tape to hold the couple in contact with the surface. There are several faults with this method, one being that conventional adhesive tapes have poor bond characteristics at relatively low temperatures. Further, the tape constitutes an added resistance to heat transfer between the gas and surface, the emissivity of the surface may be changed, and the characteristics of the surface affecting convection may be changed. The resulting error may or may not be small but the fact that an error exists should be realized. In measuring the temperatures of large surfaces it appears that this method is satisfactory provided the adhesive tape does not deteriorate.

For high temperature applications the use of a small amount of cement which will withstand high temperatures has been found successful. Again the use of fine wire and a short length of leads in contact with the surface should be used. In measuring the temperature of the glass envelopes of vacuum tubes, it was found that a small amount of "sauereisen" or equal, adhesive cement or "dental porcelain"* could be used to attach the couple to the glass in a strong bond. Because this method is subject to the same errors as is the adhesive tape method, care must be exercised in applying as thin a coating of the cement as possible. Also the junction should be in contact with the glass surface. The surface of the cement should be smoothed out so that the physical characteristics of the surface to be measured remains approximately the same. The cement would not be permitted to form an insulating layer between the surface to be measured and the thermocouple.

*Available under various trade names from Dental Supply houses

With metal surfaces, the junction can be peened into a very small hole drilled into the surface. Again, care must be used to insure that the junction makes intimate contact with the surface material.

Solder may be used as an alternate method of attaching a thermocouple to a metallic surface. An extremely small spot of solder can attach the leads to the surface. The leads should enter the solder separately so that the junction is made at the metal surface and not externally. It is sometimes desirable to solder the two wires separately but in close proximity to avoid the possibility of contact outside the surface. Figure 187 shows two methods of soldering thermocouples to metal surfaces.

With any of these methods, an important consideration is the junction and its position relative to the surface. Consider a couple peened into a hole in a hot metal surface surrounded by relatively cool air. If the two dissimilar thermocouple wires make contact away from the surface as well as at the surface, a relatively large error may be introduced since the couple will indicate a temperature between that of the surface and that of the air. The junction of the wires should be at the surface only.

15.3.4 Temperature indicating paints and waxes.

15.3.4.1 General considerations. Temperature indicating paints and wax-like substances are available and cover a wide range in temperatures. In general, these devices are of two different types: (1) paints which exhibit a color change at a specified temperature, and (2) wax-like substances which melt at specified temperatures.

The paints may be applied to a surface by brush or by spraying. The original color changes to another when the temperature of the surface reaches or exceeds the temperature change value of the particular paint. The manufacturer states the accuracy to be ± 5°C (± 9°F). While most of the paints exhibit only one color change, some exhibit as many as four changes, each at a different temperature. The temperature change is

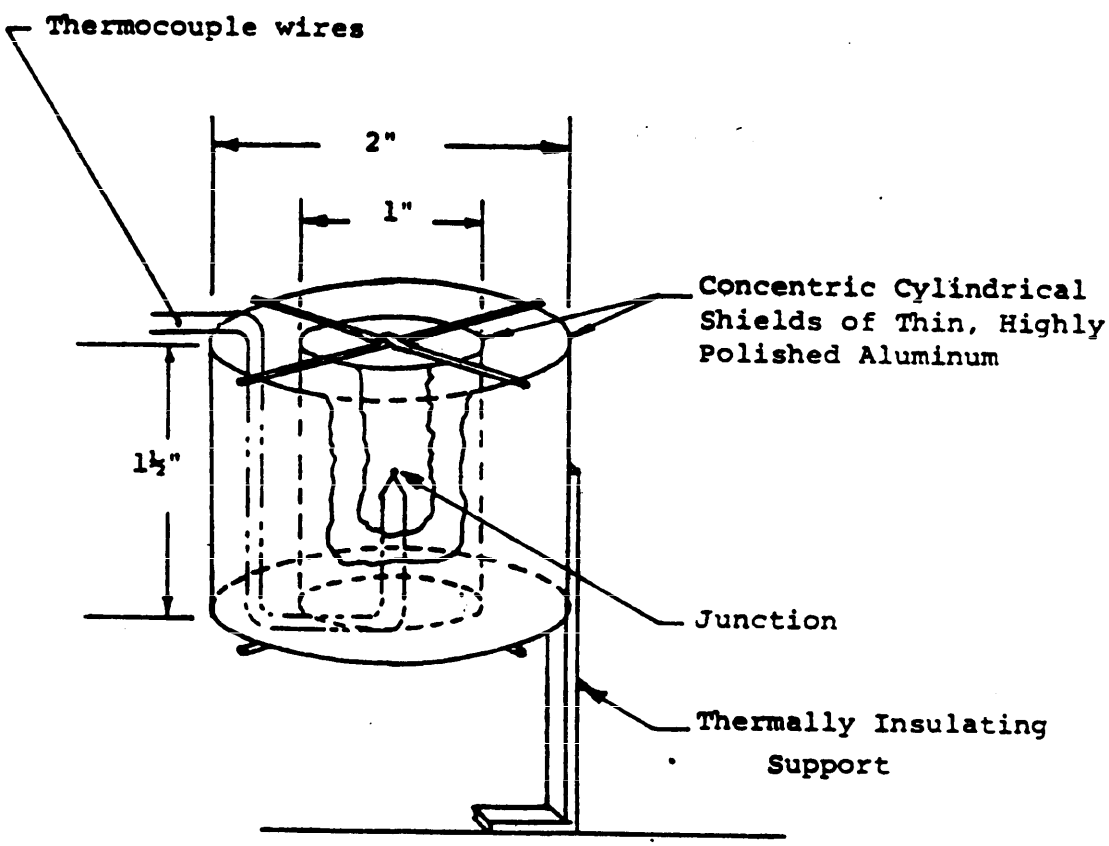


FIGURE 186. Typical Thermocouple Radiation Shield

sensitive to the duration of heating period. For example, one paint is claimed to change color at 103°C for a duration of heating of 5 seconds, 88°C for 30 seconds, finally decreasing to 80°C for 30 minutes. A color change only indicates that the temperature of the surface has exceeded the rating of the paint. Use of a second but higher rating paint which does not change color when applied to the surface indicates that the surface temperature is between the two temperature ratings. All but one of the paints exhibit a permanent color change and do not revert to the original color upon cooling. Several show a tendency to gradually revert due to the influence of atmospheric moisture.

The wax-like substances are available in crayon form and in lacquer form. A surface is touched or stroked with several of the crayons noting the lowest temperature crayon which just melts at the surface. Hence, the temperature of the surface is between that of the crayon which melts and the next higher temperature crayon.

The lacquer may be sprayed or brushed on a surface or may be coated on an object by immersion in the liquid. The lacquer dries almost immediately and, as the surface temperature rises to or exceeds the temperature rating of the particular lacquer, it melts rapidly to a liquid smear. Thus, the melting indicates that the temperature rating has been exceeded.

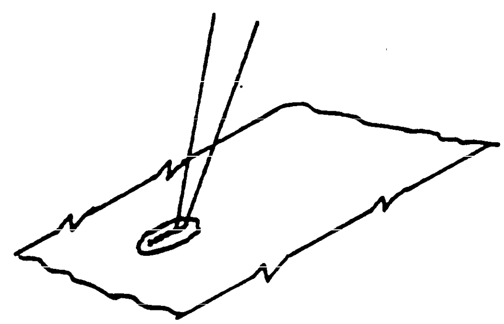
Upon cooling, the liquid smear solidifies with a glossy or vitreous appearance distinctly different from the original mark. The manufacturer states that the accuracy of both the crayons and lacquers is one percent of the temperature rating.

15.3.4.2 Temperature indicating paints.

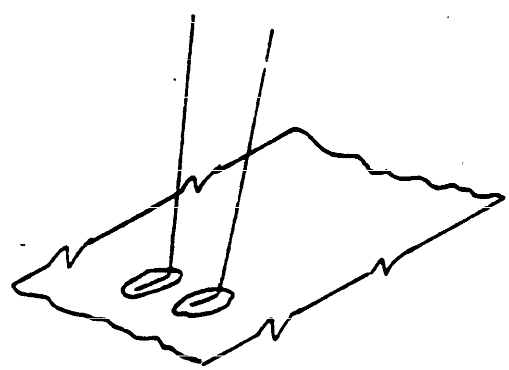
(1) Range availalbe. The paints are available in the range from 40°C (104°F) to 1100°C (2012°F). There are approximately twenty-four paints in the range from 40°C to 560°C and the intervals range from as little as 5°C to as high as 120°C. The manufacturer lists 14 paints with one color change, five with two color changes, five with three color changes, and two with four color changes. A typical paint with four color changes will change from pink to light blue at 65°C, light blue to yellow at 144°C, yellow to black at 175°C, and black to olive green at 340°C.

(2) Application of paints. The paints are packaged in powder form and alcohol is used as the solvent. Each paint must be mixed carefully in accordance with instructions. It has been found that an even coat can be applied on a small object such as a transistor for temperature distribution observations by immersing it in the paint. Tiny dots of paint on each part are adequate for thermal evaluation purposes. The individual paints as well as the brushes must be kept separate from one another so as to avoid contamination. The paints can be removed with alcohol.

(3) Accuracy and readability of temperature indicating paints. The temperature indications of these paints are in terms of color changes which occur gradually and blend into various shades and hues. It is therefore, necessary that only personnel with good color perception be permitted to utilize them. In addition, the color change is a function of the heating time and the temperature. If the heating periods are for only several minutes, extreme care in recording the heating time and



Junction Soldered to Surface



Thermocouple Leads Soldered Separately

FIGURE 187. Methods of Soldering Thermocouples on Surfaces

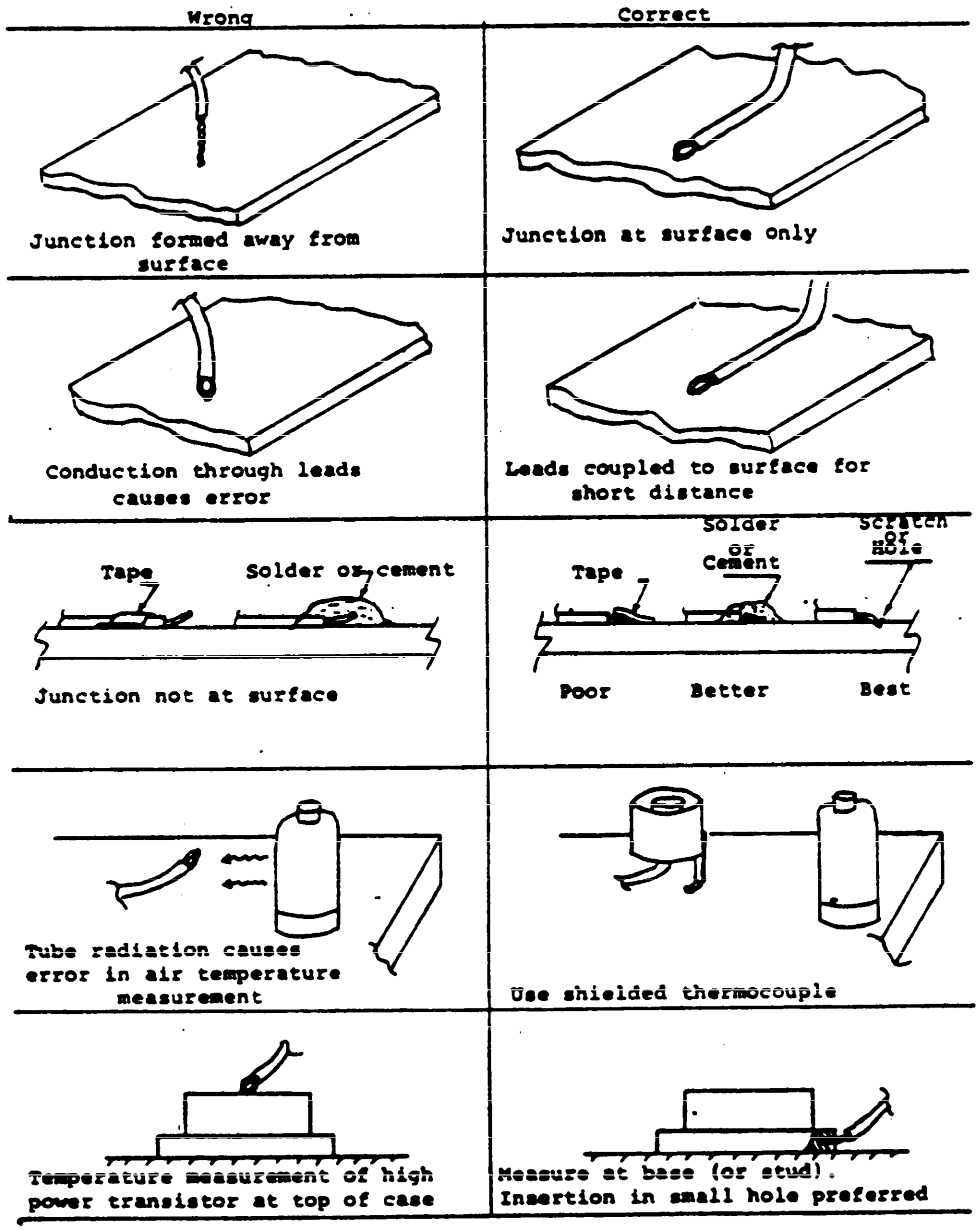


FIGURE 188. Some Common Errors in Thermocouple Application

interpreting the manufacturer's curves must be exercised in order to obtain reasonable accuracies. When heating periods of the order of thirty minutes are required to reach thermal equilibrium it is not necessary to record the heating time accurately.

Several randomly selected temperature indicating paints have been evaluated for accuracy. It was found that the accuracy of these paints was not as good as that claimed by the manufacturer. The findings of these tests are presented by Figures 189 and 190. Note that the more pronounced color changes occur above the manufacturer's calibration curves and that many shades of each color are present. Unfortunately, the evaluation of color and color change is largely dependent upon the color perception of the individual. Another person may report completely different findings. The manufacturer of these paints has identified the color changes only with terminology. It is strongly recommended that the paint manufacturer prepare and publish color charts for each paint which accurately present in full color the color and variations of its hues at each temperature which results in a color change. Thus, the accuracy and utility of the paints will be increased. A typical application of the paints to the temperature exploration of electronic equipment is presented by Figures 191 and 192. This subminiature power supply was operated under full load at 110°C thermal environment for several hours. The temperature gradients and hot spots are clearly defined. Note that color No. 3 did not always change even though all temperatures exceeded its rated 110°C.

15.3.4.3 Temperature indicating waxes and lacquers.

(1) Range available. Both waxes (crayons and lacquer) are available in the range from 45°C (113°F) to 1093°C (2000°F). In the range below 204°C they are available in 7°C intervals and above 204°C in 28°C intervals. Thus, there are 24 temperatures below 204°C and 32 above.

(2) Accuracy and readability. Tests conducted have revealed that the crayons, lacquer, and waxes indicate temperature within the claimed one percent accuracy. The melting was well defined and easily recognized. It was found that the accuracy of these waxes is excellent and within the range desired by electronic engineers. Futhermore, the rate and duration of heating had no effect on the melting point. Upon cooling, the melted areas assumed a glossy appearance. The waxes can be removed by a solvent, such as toluol.

15.3.4.4 Conclusions. Both temperature indicating waxes, paints, and lacquers are helpful tools in the design and study of the thermal aspects of electronic equipment. The waxes and lacquers can be used for relatively accurate temperature measurements as well as indicating regions of areas of high temperature. The paints can be used primarily for indicating regions of high temperature and temperature patterns. Also they give an indication of approximate temperature. However, it appears possible to calibrate the individual paints in terms of temperature and their various shades of color. Samples of painted surfaces exposed to a known temperature period could be utilized as color charts for each particular paint.

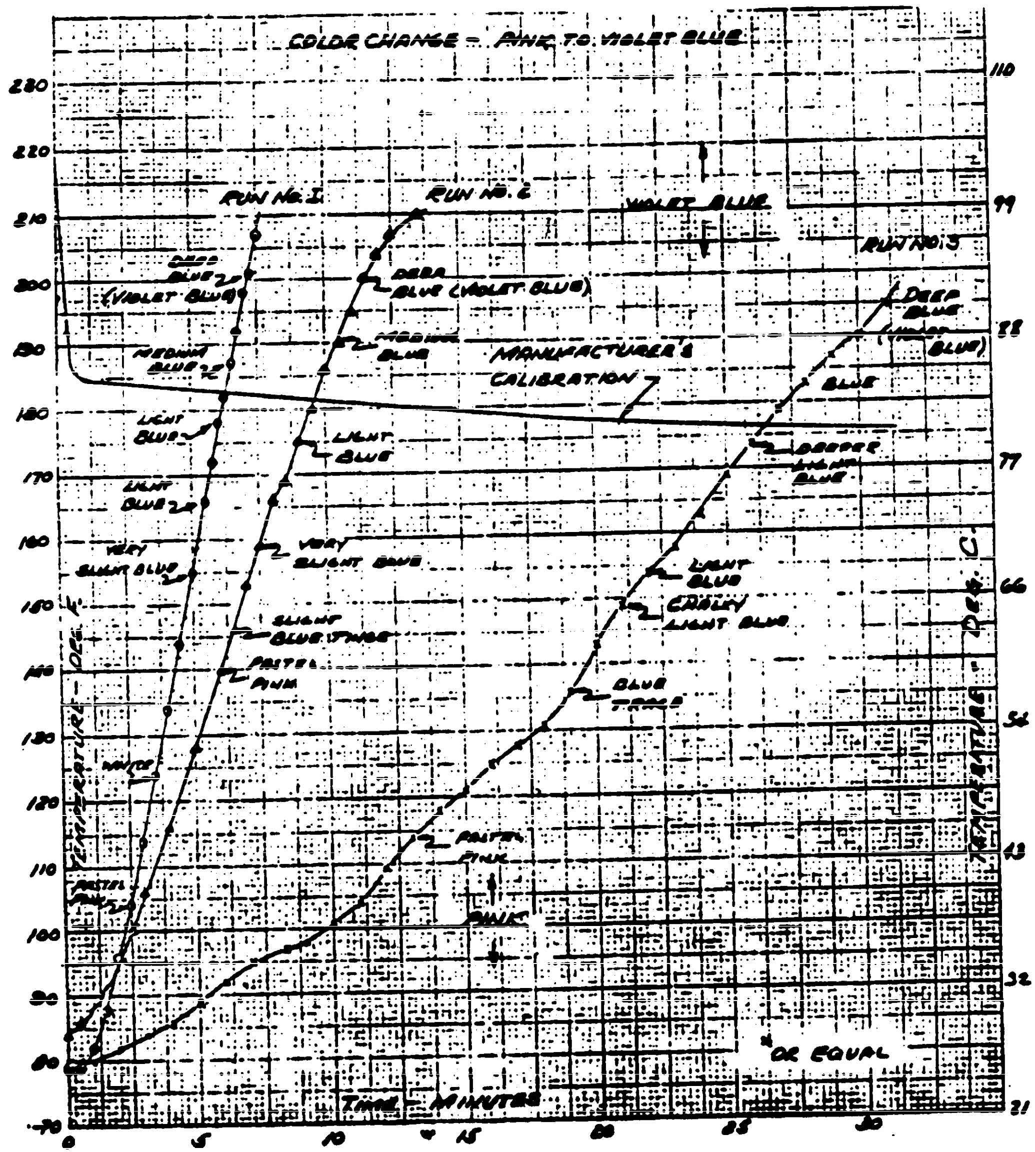


FIGURE 189. Color Change Temperature-Time History of Thermocolor*

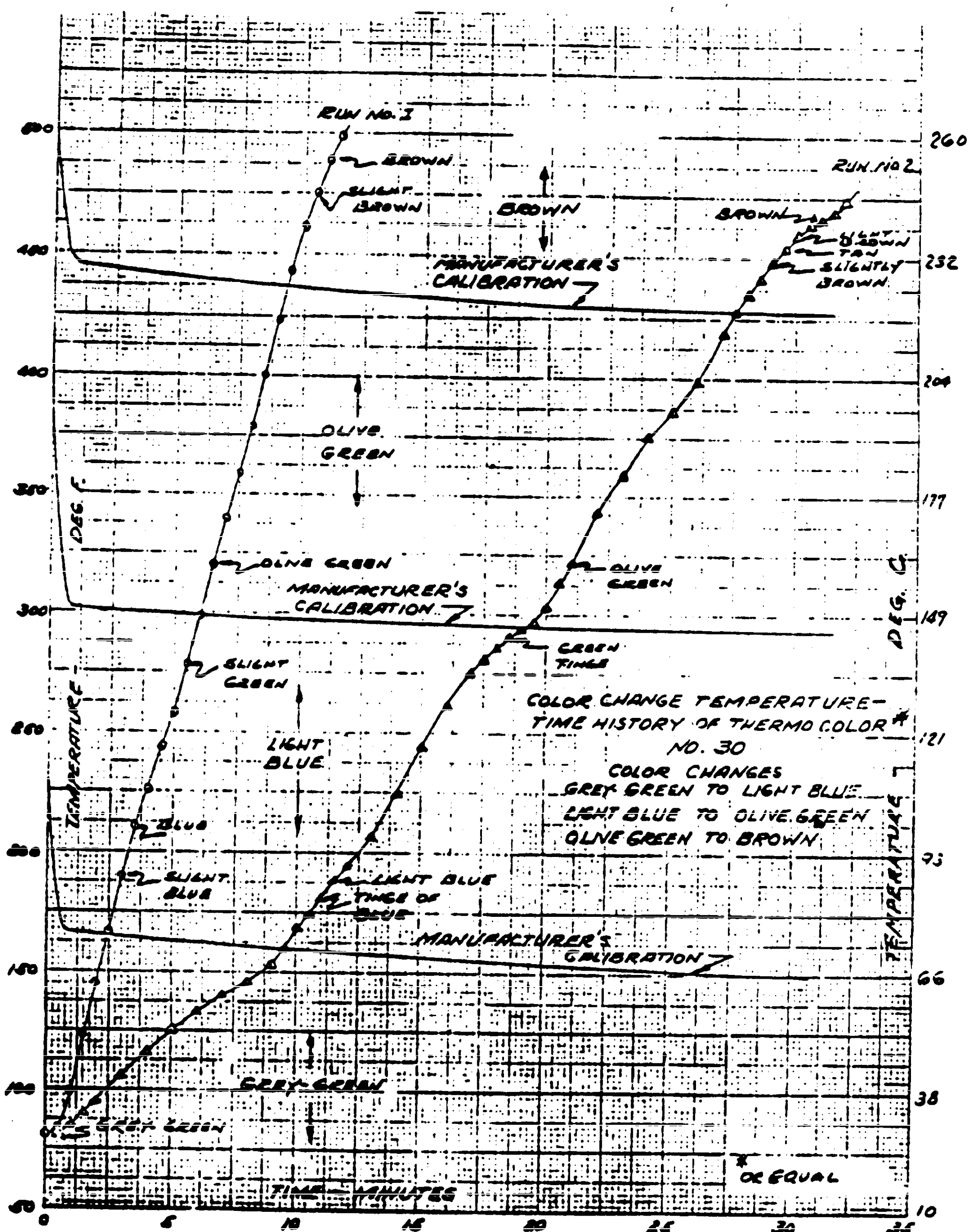
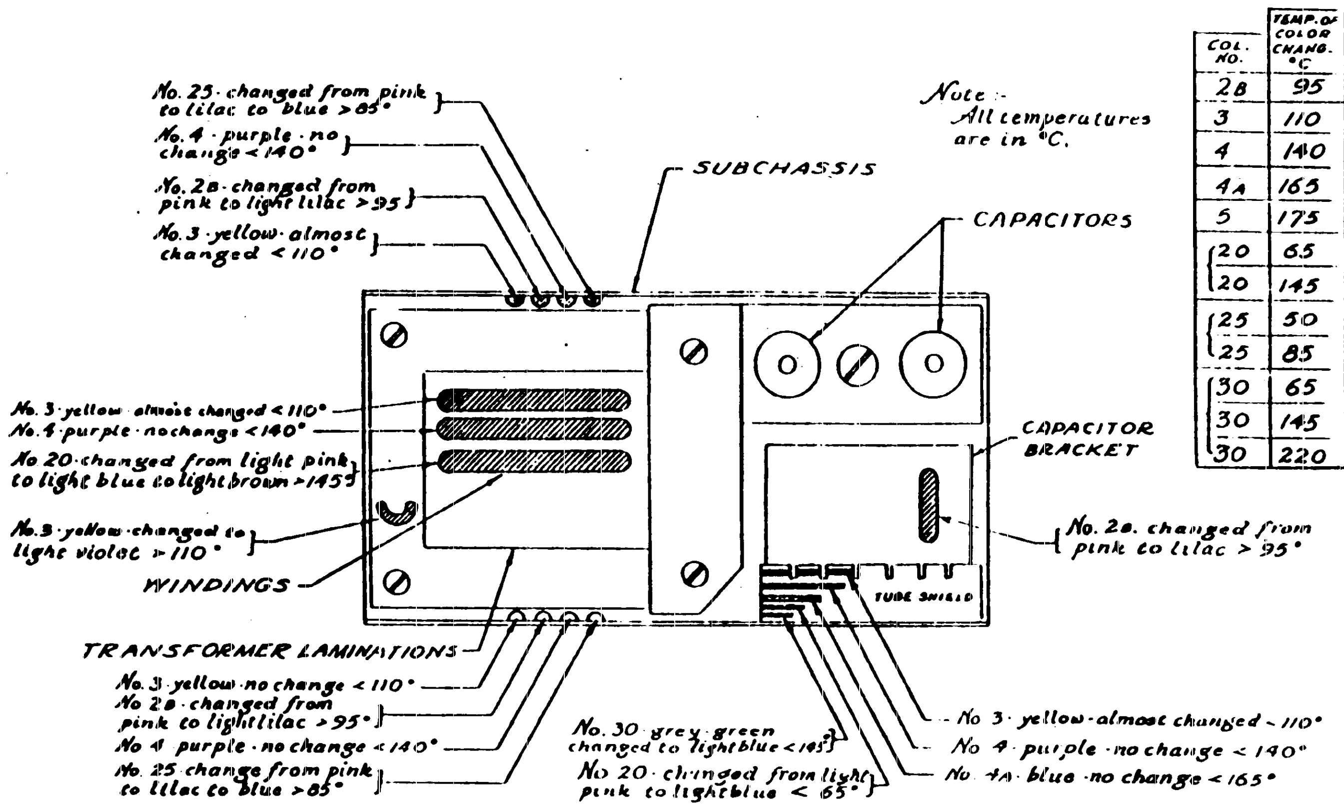


FIGURE 190. Color Change Temperature-Time History of Thermocolor*

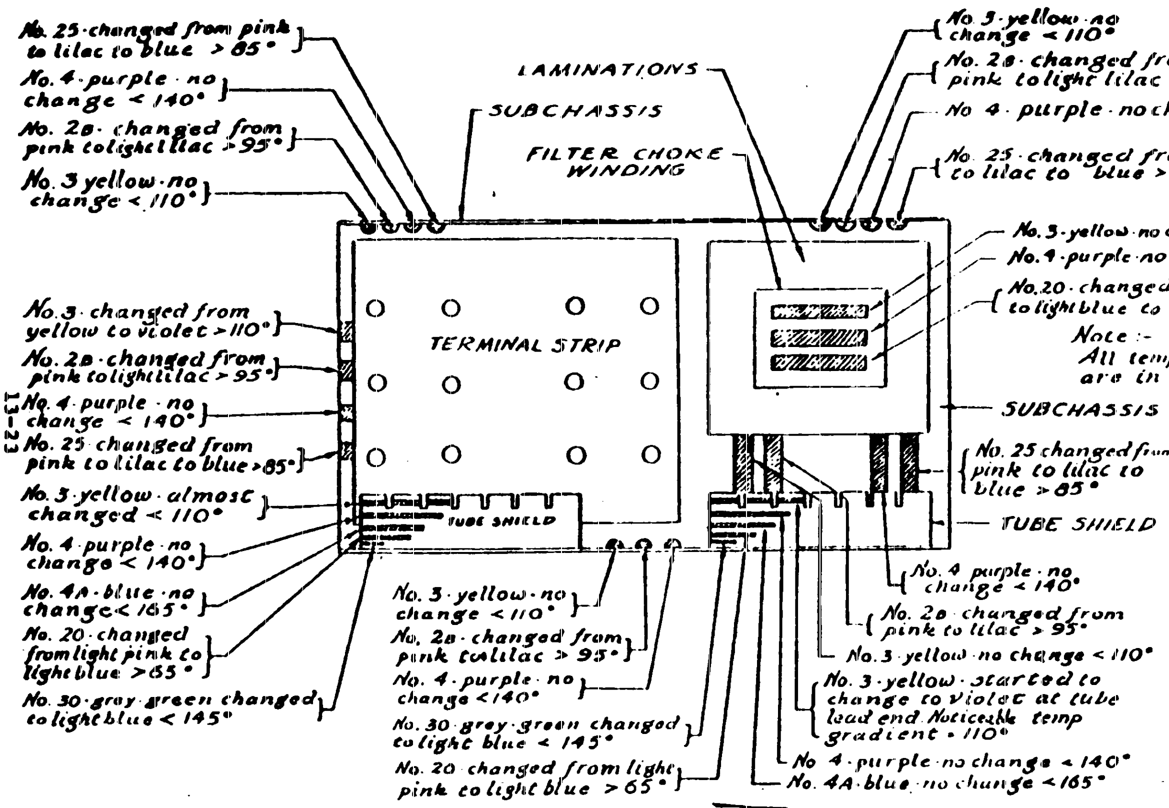


COL. NO.	TEMP. OF COLOR CHANG. °C
2B	95
3	110
4	140
4A	165
5	175
20	65
20	145
25	50
25	85
30	65
30	145
30	220

FIGURE 13-8 191

TOP VIEW SUBMINIATURE POWER SUPPLY WITH TEMPERATURE INDICATING PAINTS AFTER OPERATION AT 110°C. ENVIRONMENTAL TEMP.

FIGURE 191. Top View Subminiature Power Supply with Temperature Indicating Paints After Operation at 110°C, Environmental Temperature



COLOR No.	TEMP. OF COLOR CHANGE °C
2B	95
3	110
4	140
4A	165
5	175
20	65
20	145
25	50
25	85
30	65
30	195
30	220

FIGURE 13-9 192
BOTTOM VIEW - SUBMINIATURE POWER SUPPLY WITH TEMPERATURE INDICATING PAINTS AFTER OPERATION AT 110°C, ENVIRONMENTAL TEMP.

FIGURE 192. Bottom View-Subminiature Power Supply with Temperature Indicating Paints After Operation at 110°C, Environmental Temperature

15.3.5 Temperature measurement of electron tubes.

15.3.5.1 Measuring techniques.

(1) Thermocouples. The use of very fine thermocouple wire #36 gage or smaller appears to be the best method provided it can be used. Heat conduction along the thermocouple lead wire is reduced. The smallest possible amount of cement should be used to minimize interception of the radiant energy which would normally be transmitted through the glass. This radiant energy is absorbed by the cement raising its temperature so that the thermocouple may indicate too high a temperature. The cement should be applied in a very thin layer and smoothed on the surface so as to minimize both insulating and convective effects.

The accuracies of several thermocouple methods which were used to measure the envelope surface temperature of a subminiature tube are presented as typical of their type.

Description	Indicated Temperature
#40 AWG I.G. thermocouple	207°C
#30 AWG I.C. thermocouple	195°C
St'd. thermocouple ring recommended by tube manufacturers	206 to 194°C
Temperature indicating lacquer	204°C

NOTE: These tests were conducted under controlled conditions to minimize error. The true temperature is believed to be approximately 207°C.

The temperature of the thermocouple ring, varied with its rotation and pressure. This device tends to indicate average temperature. It is subject to certain inaccuracies due to its relatively large size which causes a change in the local radiation and convection effects. Also, error may be introduced due to the variations in contact between the ring and the glass. This thermocouple cannot, of course, be used when the tubes have tight fitting metal shields.

The hot spot on the surface of the tube envelope is of primary importance. With free convection cooling the hot spot is usually opposite the anode on conventional miniature and subminiature tubes. With tight-fitting metal shields that are well grounded thermally, the hot spot may be at the base of the envelope near the seal.

(2) Temperature indicating paints and waxes. There are applications wherein it is impossible or impractical to use thermocouples. Temperature sensitive lacquer may be used to advantage in such cases. Here again precautions must be taken to minimize inherent sources of error. The lacquer is relatively opaque and the radiation transmitted through the glass will be absorbed thus raising the temperature locally. Therefore, a very thin film of lacquer should be sprayed on over a small area on the envelope to increase its transparency and minimize error due to local radiation effects.

Another source of error is that an erroneous impression may be obtained relative to the melting of the lacquer when forced air cooling is used. Since the lacquer is a poor heat conductor, the inner surface of the lacquer (in contact with the surface whose temperature is being measured) may be molten while the outer surface to all appearances may be still in the unmelted state.

The above comments, in general, also apply to temperature indicating paints when used on electron tube glass envelopes.

Both paint and lacquer can easily be used to locate areas of high temperature and temperature gradients and patterns. They are especially useful when precise measurements are not required.

15.3.6 Temperature measurements of transistors. The temperature of power transistors should ideally be measured along the conductive heat flow path, that is at the mounting base surface in contact with the heat sink or on the mounting stud. Measurement of the temperature of the case is relatively meaningless in precision measurement unless case temperature has been "calibrated" with respect to the base or stud temperature.

It is sometimes difficult to insert a thermocouple in the thermal interface between a transistor and its heat sink. As an alternate, a thermocouple can be attached to the exposed side of the transistor mounting base or on the end of the mounting stud. The measurement error introduced will be very small.

Small dots of temperature indicating paint or lacquer can be placed on the side of the transistor mounting base or on the end of the stud. In a thermal evaluation or reconnaissance (discussed later in this chapter), only a determination of whether the maximum temperature is exceeded or not is desired. In such a "go-no go" situation the case temperature can be considered to be near the base temperature. Thus, a small dot of paint or wax which changes color or melts at the maximum temperature can be placed on the case of the transistor. This permits "adequate" temperature measurement without disassembly of the transistor mounting.

15.3.7 Temperature measurements of small parts in equipment. Temperature measurments of very small parts present several difficulties. Temperature gradients on the surfaces of small heat producing parts such as resistors may be large making it difficult to determine the temperature at a point. It appears that, at best, only an average temperature over an area can be obtained.

The heat conducting effect along thermocouple wires is magnified and use of fine wire, at least #36 gage or smaller, is essential. The junction must be on the surface and not extending into the surrounding air. This is very important.

Temperature indicating paints and lacquers are useful when applied in the form of a thin small dot. Several such dots, each of a different temperature rating, can be applied to a small area to facilitate measurement. This is particularly applicable to small transistors and IC's.

Small transistors and other semi-conductor devices fall in this category. Such circuit elements do not produce large quantities of heat and extreme care must be used in their temperature measurement in order to avoid error. In general, the temperature of the case or glass seals near the lead wires will provide a reasonable indication of internal temperature.

15.4 Thermal reconnaissance. Thermal reconnaissance is the term which has been given to a "quick and dirty" thermal test of an equipment, usually to determine if an existing equipment has thermal problems. The temperature measuring procedures can be the same as those used for any other test, but are limited to a few selected parts.

There are these important steps in conducting a thermal reconnaissance, namely:

(a) Inspection. The equipment should be carefully inspected for signs of overheating. Look for discolored and blackened parts, swollen or distorted parts, discolored printed circuit boards and paint, and labels on parts which are wrinkled or dirty. Experience is valuable in such an inspection. Frequently, a zone of overheating can be located within a chassis.

(b) Indentification of critical parts. If possible the key electronic parts which provide an index of the thermal condition of the equipment should be identified, (see paragraph 52). If data of these parts and their thermal resistances to other parts are not available, then a study of the schematic wiring diagram of the equipment should be made. From the schematic, the critical parts can be identified and then dissipations (electrical stress level) determined. Reference to MIL-HDBK-217 will provide guidance on the maximum safe temperatures for these parts.

(c) Measure the temperature of the selected critical parts with the equipment operating in the mode which results in the largest heat dissipation and under normal environmental conditions. For example, an equipment which is suspected to have thermal problems on shipboard at 50°C ambient temperature should not be tested in an air conditioned lab at 20°C. Rather, it should be tested near or at 50°C ambient, preferably on the ship.

Any of the recommended temperature measuring techniques can be used. Small dots of temperature indicating waxes or lacquers which melt or change color at the maximum temperatures of the critical parts can be applied to the parts. Dots which are approximately 1/16 to 1/32 in. dia. are sufficiently large. The wax or lacquer will not damage the parts and need not be removed when the equipment is restored to service.

Ideally, the thermal reconnaissance should be performed on the equipment in normal service. The dots of paint or wax or lacquer can be applied, the equipment reassembled and reinstalled, and then placed back in service. At least one normal flight mission is required for avionic equipment, and fixed shore stations and shipboard equipment can be operated for a day or so in its normal environment. If avionic equipment must be tested in a lab, the guidelines of section 15.7 are applicable. Inspection of the lacquer or wax dot after the normal operating period will show if the parts are overheating.

15.5 Thermal evaluation. A thermal evaluation of an equipment is a complete test to determine the temperature distribution and the temperature of every electronic part. The purpose of a thermal evaluation is to determine the adequacy and effectiveness of the thermal design and cooling system. It is an extremely important test since it is the proof of successful design and operation. Every part must be monitored or tested. Usually, the evaluation is performed in a lab or test facility. Shipboard equipment can sometimes be thermally evaluated on ship. Avionic equipment is seldom thermally evaluated in the aircraft.

The thermal evaluation of a large equipment or system can be complex or relatively simple depending upon the requirement for temperature distribution or temperature time history data. If the purpose of the evaluation is only to determine the adequacy of the thermal design and cooling system, then only dots of temperature indicating paints, waxes, or lacquers which change at the maximum safe parts temperatures need be used. This is a "go-no go" test. On the other hand, if temperature time histories and thermal resistance data pertaining to the various resistances in the thermal analog circuit are required, then thermocouples must be used. In this instance numerous thermocouple wires, switches, and recorders are required and the evaluation can be complex with a large system. Fortunately, the complete thermal evaluation of an equipment or system need only be performed once on a single equipment which is thermally representative of all other similar equipment. For example, although the AN/URT-XX radar is planned for use on many ships, only one equipment requires thermal evaluation. All other AN/URT-XX are thermally identical with the unit or system tested. However, if major design changes are made, especially in the cooling system, then the thermal equivalency no longer applies and again a representative modified unit should be reevaluated.

15.6 Thermal testing of avionic equipment. Since avionic equipment is subject to varying altitude, temperatures, and coolant flow rates, the thermal testing requires tests which will include or "take into account" these effects. Thermal tests under all conditions of altitude, flight speed (ram air temperature), air temperature, and coolant temperature can be performed in a well equipped laboratory. However, such tests are expensive and complex. Reference 46 presents recommended testing procedures for avionic equipment which involve thermal performance predictions based on selected tests which are much less expensive and complex than complete tests under all flight conditions. Also, tests can be simplified by taking advantage of the internal thermal characteristics of most modern avionic equipment; namely that the equipment is "cold plated" or liquid cooled and the internal thermal resistance between the cold plate or liquid coolant is almost constant under all conditions. Thus, only a single test under a single set of conditions will establish these resistances. The parts temperatures can be reliably predicted for other conditions of flight based on fixing these resistances and measuring the other varying thermal resistances under varying flight conditions.

Valuable information for thermal data format, important for reliability and test studies, is contained in MIL-T-23103.

15.7 Transient-temperature time history tests. Temperature time history data are of interest for testing equipment which has a very short operating duty cycle, such as equipment for a missile or satellite, and for equipment which operates for short periods in a severe environment, such as in an aircraft in a high speed dash (aerodynamic heating). In general, such equipment is designed to utilize the heat capacity of the hardware as the local heat sink, later slowly cooling off when the equipment is de-energized. In some instances the increased environmental temperature contributes to the equipment heat input. Thus, it is important in a thermal evaluation to obtain an accurate time history.

The instrumentation will require the extensive utilization of thermocouples since the temperatures of parts and heat flow paths need to be recorded. Temperature indicating paints and waxes are limited to determining if selected critical parts exceed their maximum temperatures. The thermocouple outputs must be connected to temperature recorders. Multipoint recorders can be used, but care must be exercised in their selection. Make sure that the point to point cycling and printing time of the multipoint recorder is sufficiently rapid for the thermal transients of interest. It is also important that the thermal environments for such equipments be exactly duplicated or simulated during the tests. Supplemental heat leaks (and resultant cooling) can seriously distort the test results.

15.8 Check list. The following check list (excerpted from MIL-HDBK-217 is applicable.

15.8.1 Check list for thermal testing as a basis for reliability prediction.

Basic Equipment. Does the equipment to be tested represent a reasonably finalized design (thermally close to the production model on which the reliability prediction will be projected) as follows:

In functional circuitry and dissipation?

In circuit part population, types, and ratings?

In physical circuit layout?

In mechanical aspects such as packaging and chassis layout and details, materials, and techniques of construction, antenna action, means of mounting?

In heat tansfer devices and features such as blowers, vents, ducts, plenums, baffles, thermal shunts, tube shields, heat exchangers, surface finish, and emissivities?

Basic Environment. Are the conditions of prospective use known with regard to:

Temperature of incoming and surrounding air and the range thereof?

Pressure, density, and humidity of incoming and surrounding air and the range thereof?

Air flow rate in cubic feet per minute and pounds per minute and, if variable, the range thereof?

Pressure drop through the equipment at pertinent mass flow rates?

Extent of solar radiation and nocturnal irradiation?

Can the above variables be simulated in the test? If they are not to be simulated, can their effect on thermal performances be calculated and incorporated in the test results?

Installation Effects. Are the external conditions of installation likely to affect the thermal conditions? i.e.

Temperature of structural surroundings?

Is the installation environment essentially constant or does it vary? If it varies, are its variations known or predictable?

Is the equipment proximate to heat sources or sinks which might influence interior equipment temperatures through radiation, conduction, or convection?

Can the above effects be simulated in the test? If not, can their effect on temperatures within the equipment be calculated and incorporated in the test results?

Modes of Operation. If the equipment has more than one mode of operation, i.e., transmit, receive, standby, search, track, etc., the following points should be considered.

Are the basic environment and installation effects the same for all modes of operation or is it likely that each mode will be employed under its own characteristic conditions, i.e., of altitudes, air speeds, supply air temperatures, and flow rates?

Can these differences in environment be simulated or their effects on equipment temperatures calculated?

Do the modes of operation vary in dissipation so as to vary the heat load and temperature distribution? Are these variations to be determined by test or can they be calculated?

Is the typical time distribution between modes known or predictable in terms of a typical period or cycle of operation?

15.8.2 Thermal design check points.

Natural Cooling Factors

1. Are specific heat-flow paths used?
2. Is the heat-flow path as short as possible?
3. Are metal heat-flow paths provided?
4. Do hot points, such as banks of tubes, form a bank of minimum height?

5. If vertical stacking is used, are the parts staggered?
6. Are heavy metal chassis used to conduct heat?
7. Are good heat conducting joints used throughout?
8. Are temperature sensitive parts isolated from heat sources?
9. Do polished and unpainted heat shields protect sensitive parts from the radiation sources less than 2 inches away?
10. Are parts dissipating more than 1/2 watt mounted on metal chassis or provided with metal heat paths to a heat sink?
11. Do heat sources have high emissivity?
12. Are embedded heat sources provided with metal heat conductors?
13. Are ventilation louvers provided?

Forced Air Cooling Factors

1. Is cooled filtered air directed to the hot parts?
2. Are hot parts cooled by parallel air flow?
3. Is recirculated air used for cooling? (It should not be)
4. Are air flow paths free and unobstructed?
5. Are intakes and exhausts far apart?
6. Are air flow paths the proper size?
7. Is the blower capacity adequate?
8. Has the air flow in the equipment been measured and mapped?
9. Is induced draft used?
10. Are air filters adequate and accessible for easy cleaning and replacement?
11. Are blower motors cooled?
12. Is protection provided if a blower fails?
13. Have critical temperatures been measured?
14. Is the air flow noise low?
15. Do critical power tubes have adequate air flow?
16. Are fragile fins protected?
17. Does the air first pass over the seals of critical tubes?
18. In avionics equipment, is protection against water provided? (Cooling plates or heat exchangers are recommended with avionics equipment.)

Liquid Cooling Factors

1. Is the coolant noninflammable and non-toxic?
2. Is the coolant chemically neutral?
3. Can the coolant expand freely?
4. Can the container withstand the expansion pressure?
5. Is the equipment hermetically sealed?
6. Are drain and filter plugs provided?
7. Will the coolant boil at under maximum temperatures? (It should not)
8. Is piping adequate?
9. Are the exchangers of proper design and capacity?
10. If necessary, is temperature control provided?

Vaporization Cooling Factors

In addition to liquid cooling factors, which also apply here, vaporization cooling factors are as follows:

1. Is enough coolant provided?
2. Has a makeup reservoir been installed?
3. Can vapor vent to atmosphere?
4. Can toxic fumes reach personnel?
5. Is a pressure control valve provided?
6. Is a pressure relief valve provided?
7. Have refrigeration systems been environmentally tested?

Parts

1. Are electrical stress levels of electronic parts consistent with equipment reliability requirements?
2. Has a derating policy been utilized in the circuit design?
3. Are tubes and tube sockets free of discoloration?
4. Have electron tubes been positioned to avoid formation of hot spots?
5. Is the design such that tube anodes are not red?
6. Are power resistors in groups mounted vertically?
7. Are resistors over 5 in. long mounted vertically?
8. Do power resistors radiate to heat sensitive parts? (They should not)

Capacitors

1. Are heat sensitive capacitors near hot parts? (They should not be)
2. Are shields used?
3. Are capacitors free from bulging or leaking?

Transformers and Inductors

1. Do transformers and chokes have clean low resistance thermal joints to heat conducting chassis?
2. Are the transformers and chokes designed to conduct heat from the windings to the chassis?
3. Are thermal joints free of impregnant and paint?
4. Are temperature sensitive inductors near hot parts? (They should not be)
5. Are IF and RF coils protected from electromagnetic radiation?

Semiconductors

1. Are low thermal resistance connections to heat conducting chassis provided for power transistors?
2. Is a constant temperature environment provided?
3. Are transistors dissipating more than 100 milliwatts clamped to a chassis?

4. Do power rectifiers have large fins?
5. Are power rectifiers mounted in a cool spot?
6. Are the fins vertical?

15.9 Thermal survey. During MIL-STD-781B tests a thermal survey is required. The thermal survey is different from the thermal evaluation and reconnaissance previously discussed in that parts temperatures are measured for the purpose of establishing the time to thermal equilibrium (the temperature stabilization time). This is a function of the thermal time constants of the parts, the heat capacities and specific heats of the parts and associated hardware, and the respective thermal resistances. The thermal survey identifies the part or parts which have the longest time to thermal equilibrium and measures this time. Thermal equilibrium is defined as having been reached when the rate of change of part temperature does not exceed 2°C per hour.

Determining which parts to instrument during a thermal survey requires engineering judgement and experience. Candidate parts to be instrumented can be selected based on either large thermal masses or high thermal resistances, or both. Heavy parts such as power transformers, chassis, and large heat sinks cooled by natural means are typical of parts with large masses. Parts with small heat dissipation and passive parts in areas where there is little air circulation are candidates for high thermal resistance configurations. The important point is that all candidate parts should be instrumented (usually with thermocouples) and the tests performed to identify and measure the longest stabilization time in the equipment.

15.10 Methods of measuring pressure and flow rate*.

15.10.1 Pressure drop. Pressure drop is the difference between two absolute pressures. It is analogous to the potential difference (i.e., voltage difference) encountered in electrostatics. Since an absolute pressure is more difficult to measure than the difference between two absolute pressures, difference measurement is preferred. The atmospheric pressure is usually used as the reference. However, when considering the flow of a fluid through a duct, pressure drop refers to the difference between an upstream pressure and a downstream pressure. An example is the pressure drop across an obstruction to the fluid flow.

After the rate of flow of coolant has been determined by the cooling needs of the equipment under consideration, the total pressure drop through the system for the prescribed flow rate must be found. Since fans and blowers are rated on the basis of rate of flow and pressure drop, the knowledge of both these quantities is necessary for the selection of the most economical system. Knowledge of the pressure drop across a given type of obstruction or restriction to the flow permits the velocity to be calculated, and, therefore, the volumetric flow rate.

*The theory portions of this section are applicable to any fluid flowing in a duct.

15.10.1.1 Units of pressure measurement. Atmospheric pressure is commonly measured in pounds per square inch absolute or in inches of mercury. In electronic systems, pressure differences are usually small relative to the absolute atmospheric pressure. For the required accuracy, these small pressure differences are normally measured and expressed in inches of water. In dynamic relationships for a flowing fluid, it is customary and convenient to express pressures in pounds per square foot. The use of these various units of pressure measurement leads to the importance of the following quantities:

(1) Density of water = 62.4 lb./ft.^{3}. The exact density is 62.43 lb./ft.^{3} at 4°C, the temperature of maximum density. The relative density is 0.9991 at 15°C, and 0.9971 at 25°C, taking the mass of water at 4°C as unity. (Reference 24) These variations of density with temperature are negligible.

(2) Specific gravity of water = 1.00. Specific gravity is defined in Reference 24 as "the ratio of the mass of a body to the mass of an equal volume of water at 4°C or other specified temperature."

(3) Specific gravity of mercury = 13.6. Data in Reference 24 for the densiry of mercury show that this value is sufficiently accurate over the temperature range of interest.

(4) Standard atmospheric pressure, the pressure at sea level in the standard atmosphere, is equal to:

29.92 inches of mercury (in. Hg)
407 inches of water (in. H_2O)
33.9 feet of water (ft. H_2O)
14.7 pounds per square inch absolute (psia)
2116 pounds per square foot absolute (psfa)

(5) Density of standard air = 0.0765 lb./ft.^{3}. This is the density at sea level in the standard atmosphere (pressure 29.92 in. Hg, temperature 15°C).

Static, total, and velocity pressures. When a fluid flows in a duct, three types of pressure can be measured: (1) static pressure, (2) total pressure, otherwise known as stagnation pressure, and (3) velocity pressure, known also as dynamic or impact pressure.

In any given system, only two of the foregoing pressures need to be measured, since total pressure equals the sum of static and velocity pressures.

$$\begin{array}{ccccc} \text{(total)} & = & \text{(static)} & + & \text{(velocity)} \\ P_t & = & P_s & + & P_v \end{array} \qquad (15\text{-}1)$$

15.10.1.2 Principles of manometers. In general, a manometer is an instrument which displays a pressure differential based upon the relative pressure difference and the mass of the indicating fluid. A typical U-tube manometer

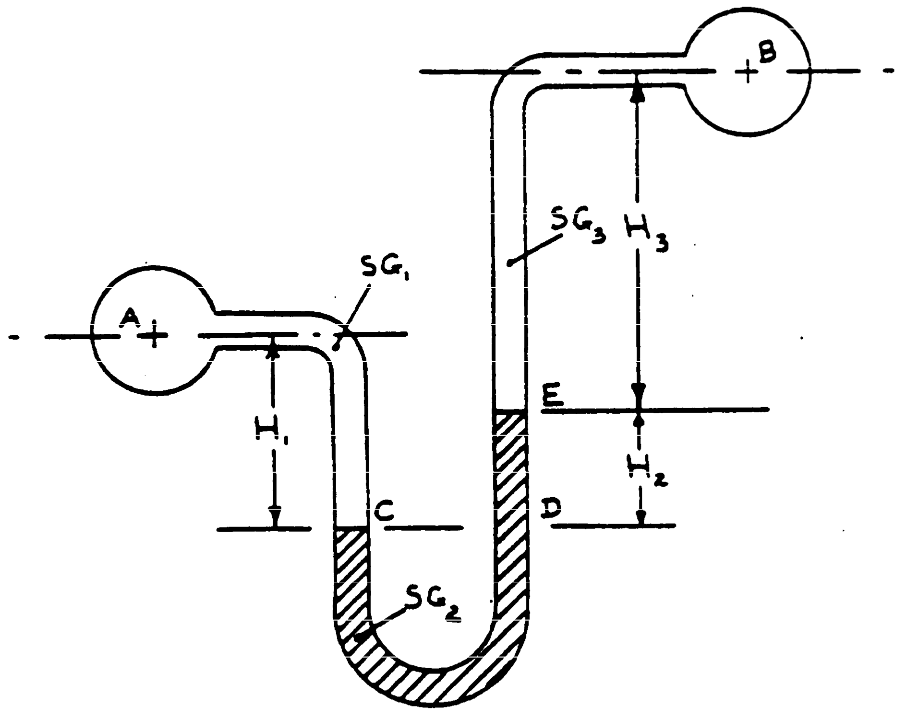


FIGURE 193. Typical U-Tube Manometer

For the illustrated manometer, assume that the pressure at point A is known and that it is required to find the pressure at point B, also the pressure difference between points A and B. The procedure will be explained below. All pressures will be expressed in inches of water.

NOTE: When pressures are expressed as the equivalent height of a column of fluid, it is common practice to use the symbol "H", which donates the height of the column.

Step 1. The pressure difference in inches of water between points A and C is $(H_1)(SG_1)$, where H_1 is the height in inches of the fluid column between A and C and SG_1 is the specific gravity of the fluid. Since point C is lower than point A, the pressure difference is added to the pressure at A, i.e.

$$H_C = H_A + (H_1)(SG_1)$$

Step 2. The pressures at points C and D are identical, since each point is at the level in the same fluid.:

$$H_D = H_C$$

Step 3. The pressure difference in inches of water between points

D and E is (H_2) (SG_2). Since point E is higher than point D, the pressure difference is subtracted from the pressure at D:

$$H_E = H_D - (H_2)\ (SG_2)$$

Step 4. Similarly,

$$H_B = H_E - (H_3)\ (SG_3)$$

Step 5. Combining the above equations:

$$H_B = H_A + (H_1)\ (SG_1) - (H_2)\ (SG_2) - (H_3)\ (SG_3) \qquad (15\text{-}2)$$

and

$$H_A - H_B = -(H_1)\ (SG_1) + (H_2)\ (SG_2) + (H_3)\ (SG_3) \qquad (15\text{-}3)$$

Almost any kind of manometer system may be solved in a like manner.

15.10.1.3 Static pressure measurement. Static pressure is defined as the compressive pressure existing in a fluid and is a measure of its potential energy. It exists in a fluid at rest or in motion and, in the latter case, causes the flow and maintains it against resistance. Some static pressure measuring devices are shown below.

15.10.1.3.1 Piezometer opening.

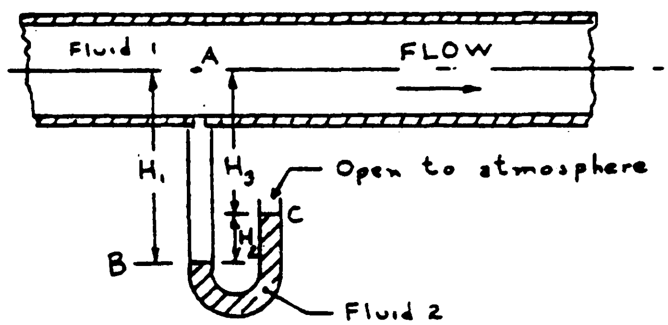


FIGURE 194. Piezometer Opening

In Figure 194 A is the point at which the static pressure is to be measured. Applying the manometer principles explained above, the absolute pressure at point A is given by:

$$H_A\ (abs) = H_C + (H_2)\ (SG_2) - (H_1)\ (SG\ )$$

where H_1 and H_2 are expressed in inches and all pressures are in inches of water. But H_C is the atmospheric pressure at point C. Therefore, relative to that atmospheric pressure

$$H_A = (H_2)\ (SG_2) - (H_1)\ (SG_1)$$

If the ratio of the specific gravity of the manometer fluid to that of the flowing fluid is large, the second term may be ignored. As an example, let the flowing fluid be standard air (density 0.0765 lb./ft.^{3}) and the manometer fluid water (density 62.4 lb./ft.^{3}), and let H_1 = 4 inches and H_2 = 1 inch. Then,

$$SG_2 = 1.0$$
$$SG_1 = 0.0765/62.4 = 0.0012$$
$$H_A = (1 \times 1) - (4 \times 0.0012) = 1 - 0.005$$
$$= 0.995 \text{ inches of water}$$

The error introduced by excluding the effect of column H_1 would be only 0.5%, which is in all probability less than the error in reading the manometer. Therefore, the reading of the water manometer may be taken directly as the pressure at point A, inches of water.

NOTE: Actually, it might be of more interest to express the pressure at A relative to the atmospheric pressure at the same elevation as point A. This pressure is less than that at point C by the amount (H_3) (SG_1), which is the pressure in inches of water exerted by a column of air of height H_3. In that case, assuming standard sea level air outside the duct:

$$H_A = (H_3)\ (SG_1) + (H_2)\ (SG_2) - (H_1)\ (SG_1)$$
$$= (H_2)\ (SG_2) - (H_1 - H_3)\ (SG_1)$$
$$= (H_2)\ (SG_2) - H_2\ (SG_1)$$
$$= (H_2)\ (SG_2)\ (1 - (SG_1/SG_2))$$

Then for the example given the result obtained by ignoring the columns of air is in error by the factor 1 - 0.0012 = 0.999, an error of only 0.1%.

When using pressure taps similar to the type shown in Figure 195, the following precautions should be heeded:

1. The tap should be inserted only where the duct has a smooth inner surface.

2. The diameter of the hole through the duct wall should be small with respect to the size of the duct.

3. The tap opening in the duct should not have burrs on its inner surface; i.e., it should be polished.

4. If a thin walled air duct is to be tapped, provisions should be made so that the hole will have a length of at least one-eighth inch. This can be accomplished by "building up" the thickness of the duct in the vicinity of the tap opening.

15.10.1.4 Static tube (piezometer opening).

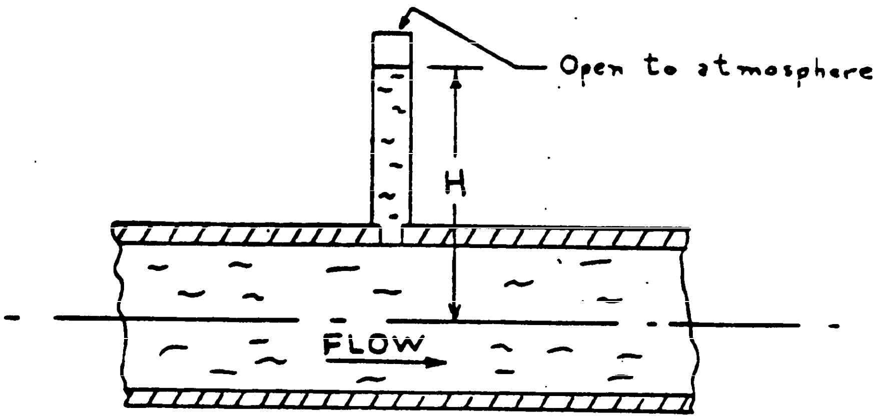


FIGURE 195. Static Tube (Piezometer Opening)

In this device, the static pressure of the fluid causes it to rise to a given level, H, above the center line of the duct so that the height, H, is the average static pressure reading in inches of fluid flowing in the duct, referenced directly to atmospheric pressure.

The limitations of this method are: (a) only a liquid of low volatility should be used, and (b) the static pressure cannot be less than atmospheric, otherwise air will be introduced into the system.

15.10.1.5 Static tube.

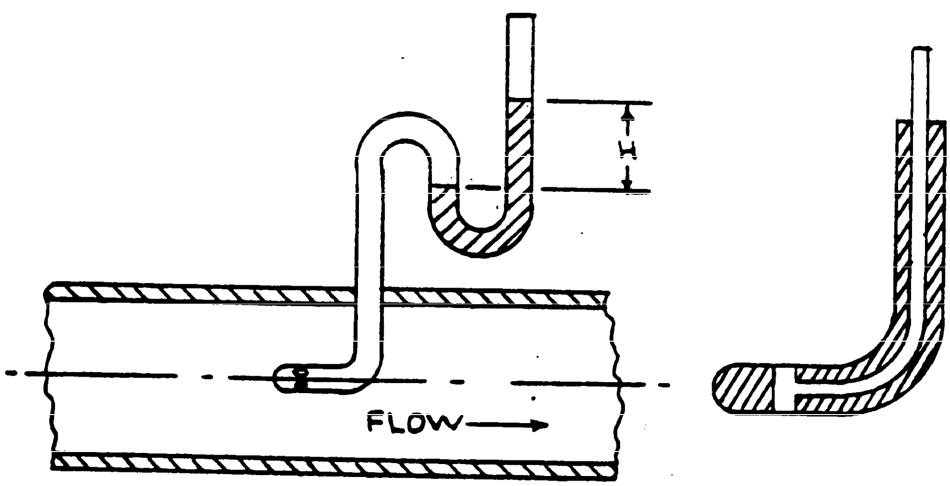


FIGURE 196. Static Tube

The static tube (Figure 196) is an instrument which is independent of the smoothness condition of the inner surface of the duct. It is especially useful in measuring static pressure in a rough-walled duct (e.g., a corrugated steel duct).

The device is a tube directed upstream with its end, or nose, closed and having radial holes in the cylindrical portion downstream from the nose. It is imperative that the axes of the duct and the static tube coincide for accurate readings to be obtained. The pressure indicated is H inches of manometer fluid. A cross-sectional view of the tube appears at the right in Figure 196.

A disadvantage of this instrument is that the indicated pressure reading may not be the true static pressure. This is possible because of the disturbance of the flow both by the cylindrical portion of the tube and by its perpendicular leg. It is recommended, therefore, that the static tube be calibrated against some accurate device.

15.10.1.6 Piezometer ring.

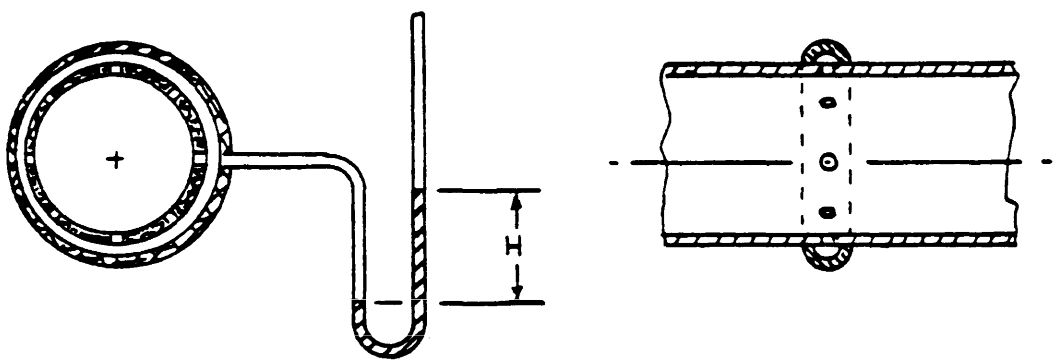


FIGURE 197. Piezometer Ring

This instrument utilizes several piezometer openings connected in parallel (see Figure 197). It is recommended for measuring the static pressure of gases flowing in ducts. The flow of a gas generally does not assume the uniform pattern of liquid flow, and gases have a tendency to spiral through a duct. A measurement of average static pressure is necessary and this is most easily accomplished with the piezometer ring.

15.10.2 Measurement of velocity and total pressures. Velocity pressure is the pressure corresponding to the velocity of the flow and is a measure of the kinetic energy of the fluid.

Total pressure, the sum of static and velocity pressure, is a measure of the total energy of the fluid. Total pressure is generally measured with a simple pitot tube. In Figure 198, the measurement of total pressure in open channel flow of liquid is illustrated.

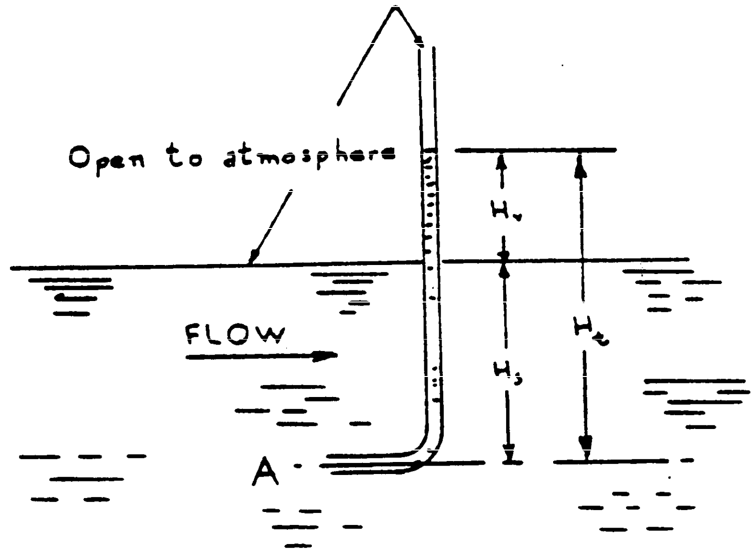


FIGURE 198. Simple Pitot Tube for Open Channel Flow

In Figure 198:

H_v = the velocity pressure at A, inches of fluid

H_s = the static pressure at A, inches of fluid

$H_s + H_v = H_t$, the total pressure at A

The system of Figure 199 is commonly used to measure total pressure of a fluid flowing in a duct, particularly when the fluid is a gas, such as air.

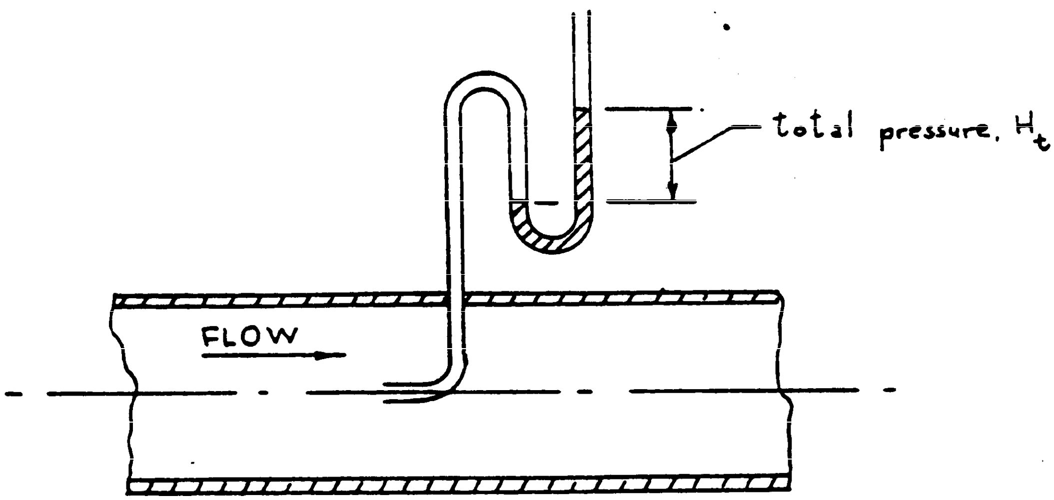


FIGURE 199. Simple Pitot Tube in a Duct

When a measurement of velocity pressure alone is desired, either of two techniques may be used: the first is a combination of the pitot tube and a piezometer opening (see Figure 200), and the second is the pitot-static tube (see Figure 201). Measurement of velocity pressure is fundamental in the determination of the velocity of the flow.

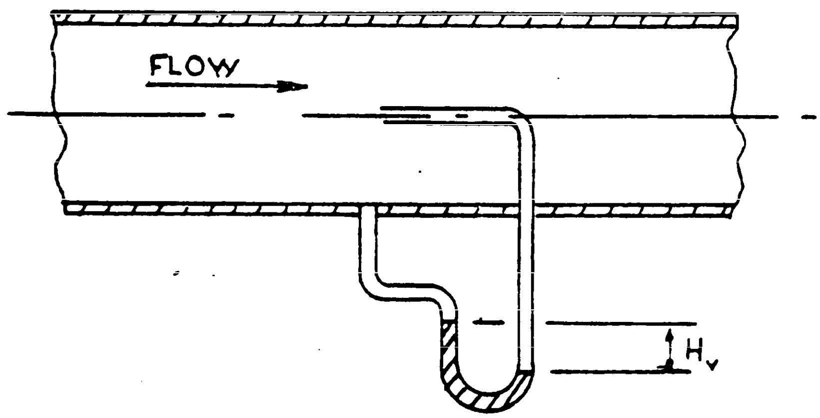


FIGURE 200. Pitot Tube with Piezometer Opening for Measuring Velocity Pressure (H_v)

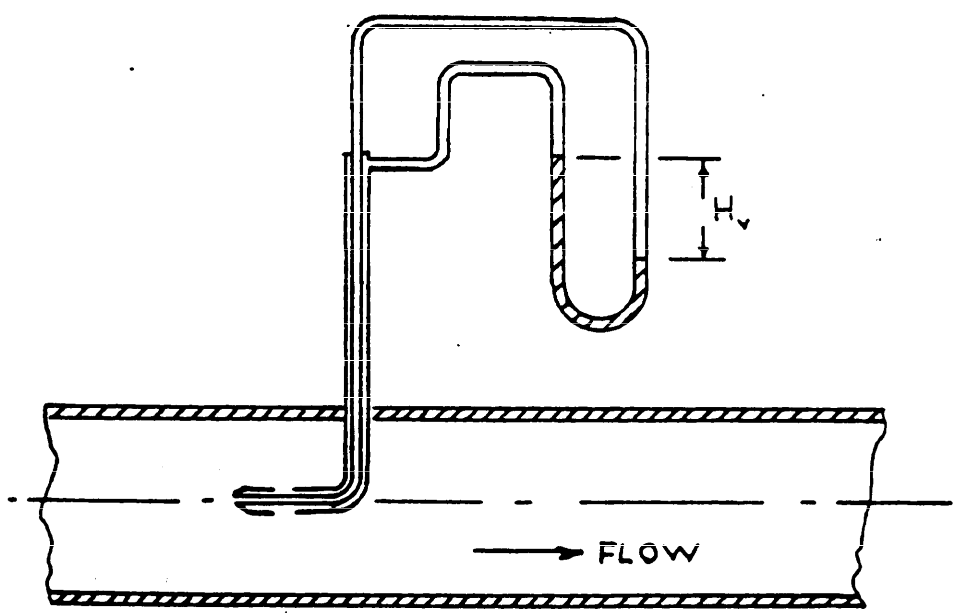


FIGURE 201. Pitot-Static Tube for Measuring Velocity Pressure (H_v)

All of the above instruments should be calibrated. In the preceding sketches of pressure measuring devices, a U-tube manometer was shown to indicate the pressure. However, a different type of manometer may be used. Some of the more common types including the simple U-tube, are shown in Figure 202.

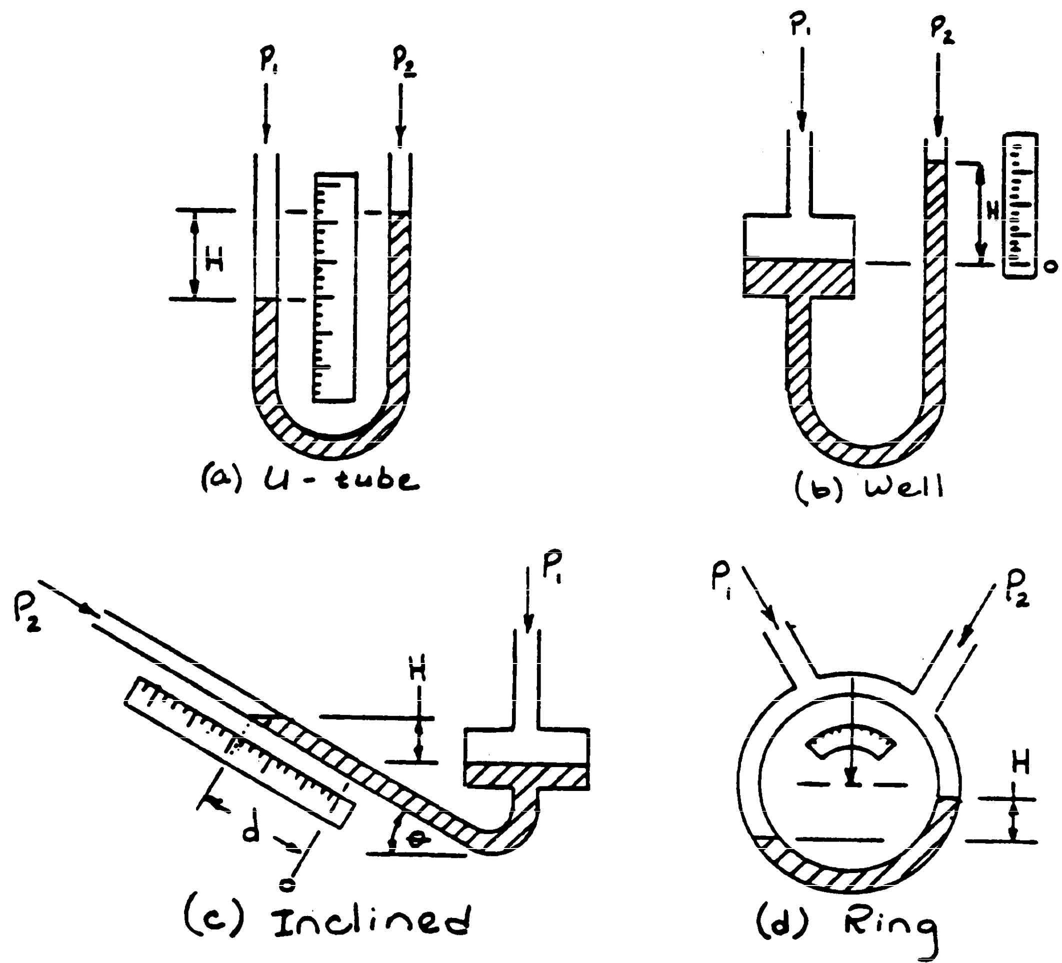


FIGURE 202. Basic Liquid Manometers

NOTE: P_1 = the pressure to be measured

P_2 = a reference pressure where $P_1 > P_2$

$H = P_1 - P_2$ = pressure reading in height of manometer fluid

The manometers in Figure 202 all exhibit the essential principles of the U-tube. However, each type has certain advantages.

The well or cistern manometer is usually set up so that the level of the liquid in the well or reservoir is on the same horizontal plane as the point at which the pressure measurement is desired. This reduces the possible error due to columns of the flowing fluid. The device has the advantage of having a permanently fixed zero point on the scale at the level of the fluid in the well. The error introduced by assuming this zero point to be fixed; i.e., assuming that the level of fluid in the well is constant, is small if the ratio of tube area to well area is small. For instance, if the cross sectional area of the indicating tube is 1% that of the well, then the error introduced is 1%.

The inclined tube manometer is similar to the well type but differs with respect to the indicating tube. This tube is set at an angle, θ, with respect to the horizontal plane and hence, has the ability to magnify the reading according to equation 15-4.

$$d = H/\sin\theta \qquad (15\text{-}4)$$

Where:

H = the pressure in inches of manometer fluid
d = the distance read on the scale
θ = the angle between the inclined tube axis and the horizontal plane

This magnification results in more accurate readings.

The operation of the ring manometer (Figure 202) depends on a rotation of the ring. After a pressure differential is applied, the fluid in the circular tube shifts. After the shift, the ring will rotate until it finds a new equilibrium point. The equilibrium point is determined by the center of gravity of the manometer fluid. This system can be very accurate if properly constructed to eliminate friction in the bearings and resistance to the moving of the connecting tubes. A large range can be covered if an arm and counterweight system is attached. However, the accuracy of the readings decreases as the range increases.

The two most commonly used manometer fluids are mercury and water. Some of the other fluids include kerosene, alcohol, and various oils.

The inclined manometer is recommended for measuring pressures in a duct through which air is flowing, using water as the manometer fluid. It is recommended because the pressure differences, either between two points in the same duct, or, in some cases, with reference to the atmosphere, are generally small and quite easily within the range of most inclined gages.

15.10.3 Velocity and flow rate.

15.10.3.1 Measurement of the velocity of the flow. Velocity can be calculated from the measured velocity pressure.

By equation 15-1, velocity pressure is the difference between the total pressure and the static pressure. Total pressure is the pressure obtained when the flowing fluid is brought to rest without loss of energy.

For an incompressible fluid and with no frictional losses, Bernoulli's equation states that

$$\frac{V_1^2}{2g} + \frac{p_1}{\rho} + z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\rho} = z_2 \qquad (15\text{-}5)$$

Where:

V_1, V_2 = the velocities at points 1 and 2, ft./sec.

p_1, p_2 = the static pressures at points 1 and 2 lb./ft.^{2}

z_1, z_2 = the elevations of points 1 and 2 above a datum level, ft.

ρ = the density of the fluid, lb./ft.^{3}

g = the acceleration due to gravity = 32.2 ft./sec.^{2}

The equation may be written

$$p_1 + \rho V_1^2/2g = p_2 + \rho V_2^2/2g + \rho\,(z_2 - z_1)$$

For low density gas, such as air, the quantity $\rho(z_2 - z_1)$ may be neglected in practical problems. It will therefore be seen that throughout the flow path, the quantity $p_s + \rho V^2/2g$ is a constant, where p_s is the static pressure. When the fluid is brought to rest (V = 0), the static pressure becomes, by definition, the total pressure p_t, i.e.,

$$p_s + \rho V^2/2g = p_t = \text{constant}$$

Hence,

$$p_v = p_t - p_s = \rho V^2/2g$$

Where:

p_v = velocity pressure, lb./ft.^{2}

p_t = total pressure, lb./ft.^{2}

p_s = static pressure, lb./ft.^{2}

V = flow velocity, ft./sec.

Solving for V,

$$V = \sqrt{2g\,p_v/\rho}$$

or

$$V = \sqrt{2g\,H_v} \qquad \begin{matrix}(15\text{-}6)\\(\text{D.E.})\end{matrix}$$

Where:

H_v = velocity head expressed in feet of the flowing fluid.

In all the above, it was assumed that the fluid was incompressible and that there were no energy losses due to friction. As will be discussed in detail later, cooling air flowing through electronic equipment may generally be considered incompressible, but a compressibility factor will be introduced to provide for cases in which compressibility becomes a factor. To provide for the effect of losses and/or instrumentation errors, a calibration coefficient C_c is introduced into equation 15-7.

$$V = C_c \sqrt{2g\ H_v} \qquad \begin{pmatrix}15\text{-}7\\ \text{D.E.}\end{pmatrix}$$

Arrangements for measuring velocity pressure were shown in Figures 200 and 201. Applying manometer principles to these arrangements, it will be found that the true velocity pressure, in inches of water, is:

$$H_v{}' \ (SG_m) - H_v{}' \ (SG)$$

Where:

$H_v{}'$ = reading of the differential manometer, inches

SG_m = specific gravity of the manometer fluid

SG = specific gravity of the flowing fluid

To convert the velocity pressure from inches of water to feet of the flowing fluid, it is necessary to divide by $12(SG_1)$. Therefore:

$$H_v{}' = H_v{}' \ (SG_m - SG)/12(SG)$$

and equation 15-7 becomes

$$V = C_c \sqrt{\frac{2gH_v{}'}{12} \left(\frac{SG_m}{SG} - 1\right)} \qquad \begin{pmatrix}15\text{-}8\\ \text{D.E.}\end{pmatrix}$$

Where (recapping the terminology):

v = flow velocity, ft./sec.

C_c = calibration coefficient (1.0 for a Prandtl pitotstatic tube)

$H_v{}'$ = manometer reading in inches

SG_m = specific gravity of manometer fluid

SG = specific gravity of the flowing fluid

g = 32.2 ft./sec.^{2}

For air the density is given by:

$$\rho = 0.0765 \left(\frac{P_s}{P_o}\right)\left(\frac{288}{t + 273}\right) \qquad \begin{pmatrix}15\text{-}9\\ \text{D.E.}\end{pmatrix}$$

* When the flowing fluid is air, SG_m/SG_1 could be substituted for $(SG_m/SG_1) - 1$ without significant error, but this simplifies calculations very little.

Where:

ρ = density, lb./ft.^{3}

p_s = static pressure

p_o = standard sea level pressure

t = temperature, °C

p_s and p_o may be expressed in any units provided the same units are used for both pressures.

The specific gravity of the air is

$$SG = \rho/62.4 \tag{15-10}$$
(D.E.)

15.10.3.2 Flow rate measurement. The rate of flow of a fluid in a duct, particularly a gas, is most easily and accurately measured by a head meter. A head meter consists of two units: (1) a primary device producing a differential head or pressure difference which varies as the square of the rate of flow, and (2) a secondary device, usually a manometer of some kind, for measuring the differential head. The head meters commonly used are: (1) thin plate, sharp-edged orifice; (2) Venturi tube; (3) flow nozzle or rounded entrance nozzle; and (4) pitot tube.

Pressure loss caused by the pitot tube may be considered to be zero; for the other three devices, a loss of 10 to 90 percent of the differential pressure will occur. The accuracy of all four types of meters will be within two percent if they are properly installed, calibrated, and operated.

The head meter location is very important, as upstream disturbances affect the flow. It is a good general practice to allow 10 diameters of straight duct upstream and 5 diameters downstream, if possible, and to use straighteners (egg crate or a nest of several tubes) several diameters from the meter.

The discussion which follows applies to flow measurement with an orifice, Venturi, or flow nozzle type head meter. Since the principles of measuring flow rate with a pitot tube are different, that device will be discussed separately.

15.10.3.3 Thin-plate, sharp-edged orifice. The orifice meter is the most common flow-metering device. It is relatively inexpensive, easy to construct, and easy to install. A thin-plate concentric orifice is a flat diaphragm with a circular hole in the center. It is usually clamped concentrically between the flanges in a circular duct.

Three sets of pressure taps are recognized by the American Society of Mechanical Engineers for the measurement of differential head:

(a) Flange taps with the centers of the holes 1 inch from the respective faces of the orifice plate.

(b) Vena contracta taps with the upstream hole located one pipe diameter from the upstream face and the downstream hole at the "vena contracta" where the diameter of the jet of air issuing from the orifice is a minimum. This location is taken as 0.5 to 0.8 pipe diameters from the downstream face.

(c) Radius taps, with the upstream hole located one pipe diameter from the upstream face and the downstream hole one-half pipe diameter from the downstream face.

It is recommended that the orifice meter be constructed as shown in Figure 203. This design is presented (in Reference 46) by the Ohio state University Research Foundation, based on Section III of "Gas Measurement Committee Report No. 2", issued by the American Gas Association, May 5, 1935.

The overall orifice plate thickness should be at least 1/8 inch (as shown) and not over 1/4 inch for inside pipe diameters d_1 between 4 inches and 16 inches. For larger ducts, the thickness may be increased. For ducts less than 4 inches inside diameter, the thickness should be at least 1/16 inch. The thickness at the orifice edge should not exceed:

(a) 1/30th of the pipe diameter d_1

(b) 1/8th of the orifice diameter d_2

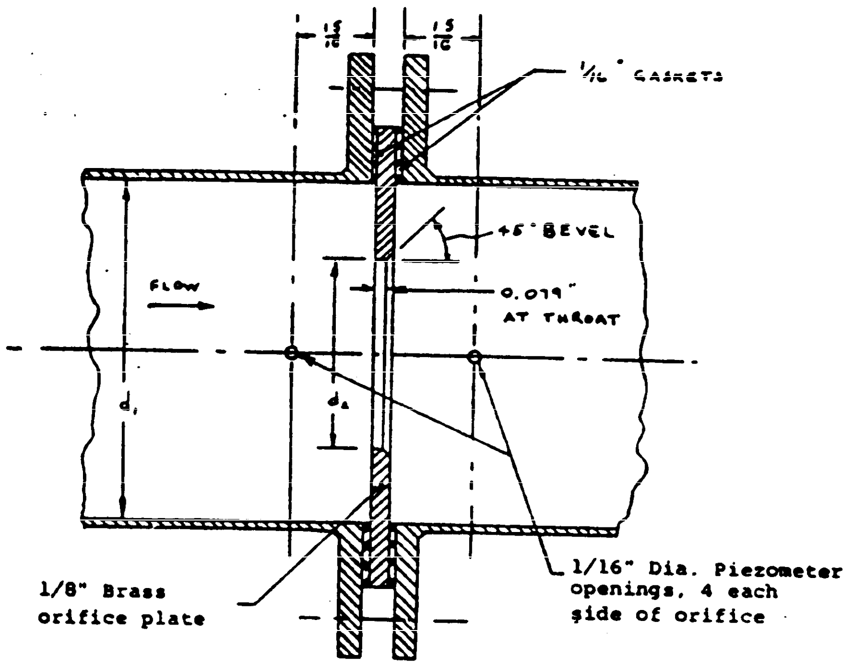
(c) 1/4th of the "dam height" $(d_1 - d_2)/2$

The minimum of these requirements should govern in all cases. It is recommended that whenever practicable the requirement (a) be reduced to 1/50th of the pipe diameter d_1. The dimensions shown in Figure 203 meet these requirements for orifice diameters of 1.000, 1.500, 2.350, and 2.900 inches in pipes of approximately 4 inch diameter. Such dimensions are proposed in Reference 46 to cover the flow range from 0.01 to 0.3 lb./sec. (8 to 235 cubic feet per minute for standard air of density 0.765 lb./ft.^{3}). It is desirable to keep the orifice pressure drop in excess of one inch of water.

The upstream face of the orifice plate should be as flat as can be obtained commercially and must be perpendicular to the axis of the pipe when in position between the orifice flanges. Any plate which, when clamped between the flanges, does not depart from flatness along any diameter by more than 0.01 inch per inch of pipe radius is considered flat. The upstream edge of the orifice should be square and sharp so that it will not appreciably reflect a beam of light when viewed without magnification, and should be maintained in thic condition at all times. Any rounding or leveling of the sharp upstream edge, even slightly, increases the flow rate for a given differential pressure. The plate must be kept clean and free from accumulation of dirt and other extraneous materials.

The four upstream piezometer openings are connected together and to one side of a manometer measuring the pressure drop. Likewise the four downstream openings are connected together and to the other side of the manometer. The upstream openings may also be connected to a manometer measuring the upstream (gage) pressure relative to the atmospheric pressure.

15.10.3.4 Venturi meter. The venturi meter (see Figure 204) differs from the orifice and flow nozzle head meters in that it has a generally smooth inner surface without sharp discontinuities, and hence, gives rise to only a minimum amount of turbulence and pressure drop. The device



NOTE: See appendix for calibration curves

FIGURE 203. Thin Plate, Sharp-Edged Orifice (Ref. 46)

consists of an upstream section the same size as the duct, which has taps for measuring the upstream static pressure; a converging conical section; a cylindrical throat, which also has pressure taps; and a gradually diverging conical section leading to a cylindrical section the same size as the duct. A differential manometer is usually connected between the pressure taps.

The size of a venturi meter is specified by the duct and throat diameters, e.g. a 6" x 4" venturi meter fits a 6 inch diameter duct and has a 4 inch diameter throat. The diverging after-section helps to restore

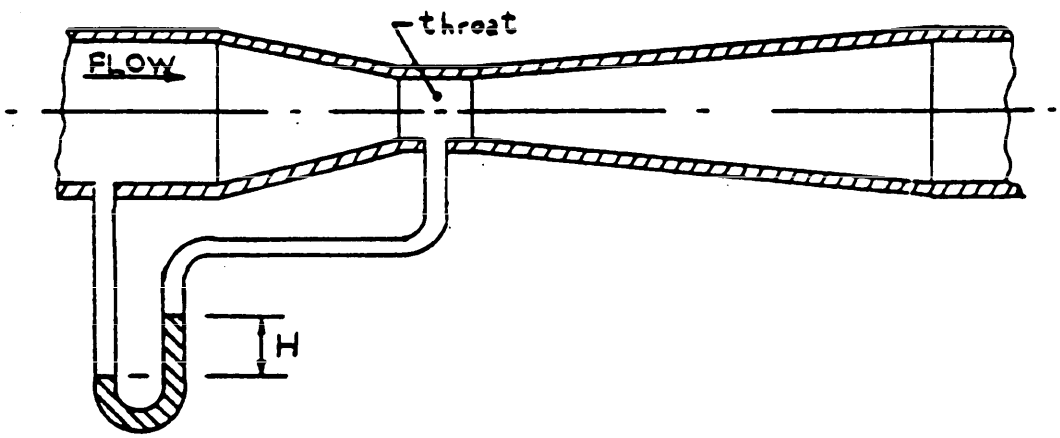


FIGURE 204. Venturi Meter

some of the static pressure that is lost when the flow is constricted, hence reducing the overall pressure drop. In this respect the venturi meter is much superior to the orifice and to the flow nozzle.

The venturi converging section has an included angle of between 25° and 30°. The diverging section should never exceed 7-1/2° included angle.

15.10.3.5 Flow nozzles. Flow nozzles are generally designed to be clamped between the flanges of a duct and usually have rather abrupt curvatures of the converging surfaces. The curvature of the inner surface may be either circular or elliptical. The flow nozzle terminates in a short cylindrical section, which exhausts into the duct. A typical flow nozzle with the conventional pressure taps is shown in Figure 205.

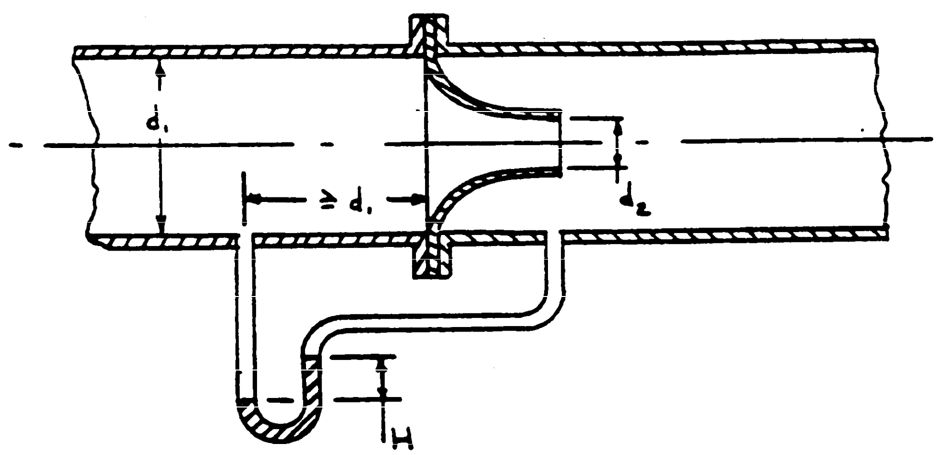


FIGURE 205. Flow Nozzle

15.10.3.6 Recommendation. The orifice meter is recommended for measurement of air flow primarily because of its simple construction and easy installation. Once calibrated, the orifice is more than sufficiently accurate for the systems encountered in electronic cooling. It is recommended that the details of construction of the orifice be in accordance with Figure 203.

15.10.4 Flow rate equations. The orifice, venturi, and flow nozzle head meters have a common feature in that a pressure drop is created by a reduction in area of the flow path, and this pressure drop is measured.

For an incompressible fluid with no frictional losses, and neglecting the effect of the difference in elevation (if any) between points where the static pressure is measured, Bernoulli's equation 15-5 yields

$$\frac{V_1^2}{2g} + \frac{p_1}{\rho} = \frac{V_2^2}{2g} + \frac{p_2}{\rho}$$

or solving for V_2: $$V_2 = \left(V_1^2 + \frac{2g(p_1 - p_2)}{\rho}\right)^{1/2} \qquad (15\text{-}11)$$

Where:

V = velocity, ft./sec.
p = static pressure, lb./ft.^{2}
ρ = density of flowing fluid, lb./ft.^{3}
g = 32.2 ft./sec.^{2}
Subscript 1 refers to a point in the duct ahead of the meter
Subscript 2 refers to a point at the reduced flow path area

By the continuity equation for an incompressible fluid, (Reference 47)

$$Q = V_1A_1 = V_2A_2 \qquad (15\text{-}12)\ (\text{D.E.})$$

Where:

Q = volumetric flow rate, ft.^{3}/sec.
V_1 = velocity in duct, ft./sec.
V_2 = velocity at reduced section. ft./sec.
A_1 = cross sectional area of duct, ft.^{2}
A_2 = area of the reduced section, ft.^{2}

From equation 15-12,

$$V_1 = V_2A_2/A_1 \qquad (15\text{-}13)$$

Substituting equation 15-13 into 15-11 and solving for V_2

$$V_2 = \frac{1}{\sqrt{1 - (A_2/A_1)^2}} \sqrt{\frac{2g\,(p_1 - p_2)}{\rho}} \qquad (15\text{-}14)$$

It is customary to define as the "velocity of approach factor" the quantity

$$M = 1/\sqrt{1 - (A_2/A_1)^2} \tag{15-15}$$

Then, equation 15-14 may be written,

$$V_2 = M\sqrt{2gH}$$

where H is the differential head between points 1 and 2, expressed in feet of the flowing fluid; i.e.,

$$H = (p_1 - p_2)/\rho$$

It follows that:

$$Q = V_2A_2 = MA_2\sqrt{2gH} \tag{15-16}$$

Equation 15-16 yields only the "ideal" flow rate, based on the assumptions of incompressible flow completely filling the flow channel and no losses. The general equation for the actual flow rate through the meter is

$$Q = YMC_dA_2\sqrt{2gH} \tag{15-17}$$

Where:

Q = volumetric flow rate, ft.^{3}/sec.
Y = compressibility factor
M = velocity of approach factor
C_d = coefficient of discharge for the particular meter
A_2 = area of minimum section, ft.^{2}
H = differential head in feet of the flowing fluid, based on fluid density at the upstream tap
g = 32.2 ft./sec.^{2}

The mass flow rate is given by

$$M = \rho_1 Q \tag{15-18}$$

where:

m = mass flow rate, lb./sec.
ρ_1 = fluid density at the upstream tap

15.10.5 <u>Flow rate calculation</u>. For the orifice, venturi, and flow nozzle types of flow meters, volumetric and mass flow rates are calculated by use of equations 15-17 and 15-18, respectively.

15.10.5.1 <u>Velocity of approach factor</u>. In equation 15-17, the velocity of approach factor has already been defined as:

$$M = \frac{1}{\sqrt{1 - (A_2/A_1)^2}}$$

which may also be written:

$$M = \frac{1}{\sqrt{1 - (d_2/d_1)^4}} \qquad \text{(15-19) (D.E.)}$$

Where:

M = the velocity of approach factor
A_1, d_1 = the cross sectional area and diameter, respectively, of the duct
A_2, d_2 = the cross sectional area and diameter, respectively, of the reduced section

In equation 15-15, A_1 and A_2 may be expressed in any units so long as the same units are used for both quantities. Likewise, in equation 15-19, it is necessary only that the same units be used for d_1 and d_2. M is a pure number; i.e., it is dimensionless. Note carefully, however, that A_2 in equation 15-17 must be expressed in square feet, in order to make Q come out in cubic feet per second.

15.10.5.2 Coefficient of discharge. The coefficient of discharge is defined as

$$C_d = \frac{\text{actual flow rate}}{\text{ideal flow rate}} . \qquad \text{(15-20) (D.E.)}$$

The ideal flow rate was given by equation 15-16. The actual flow rate is smaller due to the effect of frictional losses, and in the case of the orifice, meter, due to the "vena contracta" effect described below. C_d is a function of the ratio of the area of the constriction to the area of the duct, the velocity of the flow (more exactly, the Reynolds number), and the type of meter. In equation 15-16, the effect of compressibility is neglected. (The factor Y, discussed later, provides for this effect.)

The coefficient of discharge is usually determined experimentally, although it can be calculated from data presented in most mechanical engineering handbooks and fluid mechanics textbooks. Typical values are listed below:

(a) Orifice meters: As the flow passes through the orifice opening, the flow cross section contracts further and approaches a minimum cross section, approximately 0.62 that of the opening. This phenomenon is called "vena contracta." The approximate C_d for orifices is 0.60.

(b) Venturi: In the venturi, there is not flow contraction as in the orifice, due to the gradual transition between the duct and the throat. C_d for venturis ranges from 0.95 to 0.99.

(c) Flow nozzles: The curved entrance and short cylindrical section of the nozzle prevent contraction. Generally, for a smooth surfaced nozzle, C_d varies from 0.97 to 0.99.

NOTE: The coefficient of discharge listed above are merely representative values and are not to be used for accurate measurements. The C_d for a particular meter can usually be obtained from the manufacturer.

15.10.5.3 Compressibility factor and compressible flow. A normal low-density gas will, of course, be compressed or expanded to different densities as it passes through electronic equipment or through a head meter, so that the flow is not truly incompressible. In many cases, the effect of compressibility on flow rate measurements will be found to be negligible. For air flowing through an orifice, the effects of compressibility will be small if the velocity through the orifice does not exceed 0.4 of the velocity of sound, which is 450 ft./sec. for standard sea level air. (Speed of sound in feet per second = 1120 (t + 273)/288, where t is the temperature in °C.) However, in test work the safest procedure is to evaluate the compressibility factor Y. For air, this is easily accomplished with the chart presented below.

The factor Y is a function of the area ratio A_2/A_1, the pressure ratio p_2/p_1, and the specific heat ratio:

$$k_c = \frac{c_p}{c_v} = \frac{\text{specific heat at constant pressure}}{\text{specific heat at constant temperature}}$$

The theoretical expression for Y is:

$$Y=\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{K_c}}\left(\frac{k_c}{k_c-1}\right)\left(\frac{1-(p_2-p_1)^{(k_c-1)/k_c}}{1-(p_2-p_1)}\right)\left(\frac{1-(A_2/A_1)^2}{1-(A_2/A_1)^2(p_2/p_1)^{2/k_c}}\right)\right]^{1/2} \quad \begin{matrix}(15\text{-}21)\\(\text{D.E.})\end{matrix}$$

The compressibility factor, Y, is given as a function of the pressure, diameter, and specific heat ratios for air in Figure 206. The values of Y for the venturi meter and nozzle have been computed from Figure 204, while the values for sharp-edged orifices have been determined experimentally. The difference arises from the fact that the minimum jet area and the pressures at various points are different for the orifice, when operating at the same pressure drop with the same diameter ratio. (Reference 47)

15.10.5.4 Differential head. The differential head H in equation 15-1 must be expressed in feet of the flowing fluid, based on density at upstream tap. The pressure difference produced by a head meter is usually measured by a differential manometer, as shown in Figures 204 and 205 (and as described in connection with Figure 203). The reading of such a manometer in inches of manometer fluid will be designated H'. We then have the relationship

$$H = \frac{H'}{12}\left(\frac{SG_m}{SG_1} - 1\right)^{*} \quad \begin{matrix}(15\text{-}22)\\(\text{D.E.})\end{matrix}$$

Where:

H = differential head in feet of the flowing fluid based on density at the upstream tap
H' = manometer reading, inches
SG_m = specific gravity of the manometer fluid
SG_1 = specific gravity of the flowing fluid at the upstream tap

*(From pg. 559) As in equation 15-8, SG_m/SG_1 could be substituted for $(SG_m/SG_1)-1$ without significant error, but this simplifies calculations very little.

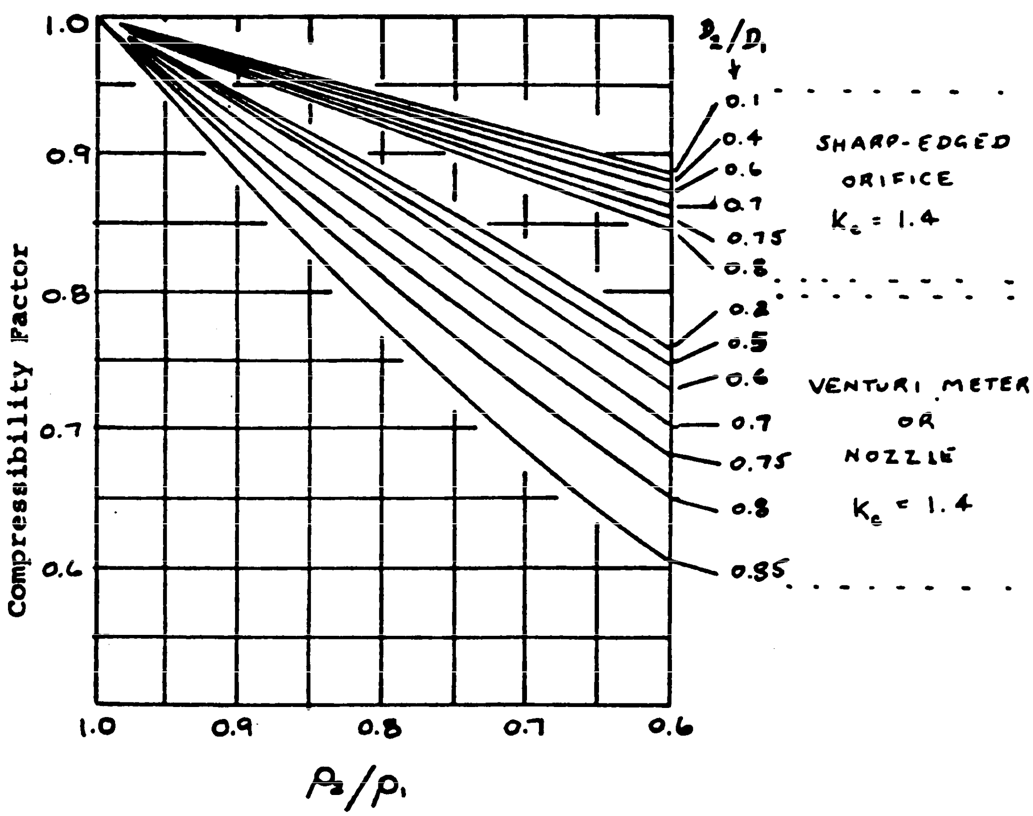


FIGURE 206. Compressibility Factor

The differential head H may also be expressed as

$$H = \frac{P_1 - P_2}{\rho} \tag{15-23}$$

Where:

p_1 = static pressure at upstream tap, lb./ft.^{2}

p_2 = static pressure at downstream tap, lb./ft.^{2}

ρ_1 = density of the flowing fluid at the upstream tap, lb./ft.^{3}

When the flowing fluid is air, density and specific gravity at the upstream tap are given by equations 15-24 and 15-25:

$$\rho_1 = 0.0765 \left(\frac{p_1}{p_0}\right)\left(\frac{288}{t_1 + 273}\right) \quad \text{(15-24) (D.E.)}$$

$$SG_1 = \rho_1/62.4 \quad \text{(15-25) (D.E.)}$$

Where:

ρ_1 = density of air at the upstream tap, lb./ft.^{3}

p_1 = static pressure at the upstream tap

p_0 = standard sea level pressure

t_1 = temperature at upstream tap, °C

SG_1 = specific gravity of air at the upstream tap

p_1 and p_0 must be expressed in the same units. It is often convenient to determine p_1 in inches of mercury, in which case we use p_0 = 29.92 in. Hg.

15.10.5.5 System constant. It is convenient to define the system constant

$$C_s = MC_aA_2\sqrt{2g} = \frac{C_d A_2 \sqrt{2g}}{\sqrt{1 - (A_2/A_1)^2}} \quad \text{(15-26) (D.E.)}$$

Then, from equation 15-17,

$$Q = C_sY\sqrt{H} \quad \text{(15-27) (D.E.)}$$

and

$$m = C_sY\rho_1\sqrt{H} \quad \text{(15-28) (D.E.)}$$

In equations 15-27 and 15-28, we may substitute from equation 15-23,

$$H = \frac{p_1 - p_2}{\rho_1}$$

It is probably more convenient to determine H from equation 15-22:

$$H = \frac{H'}{12}\left(\frac{SG_m}{SG_1} - 1\right)$$

It is suggested, then, that C_s be determined from equation 15-26, H from equation 15-22, Y from Figure 15-23, Q and m from equation 15-27 and 15-28. Recapping the nomenclature used in these equations:

C_d = coefficient of discharge
C_s = system coefficient
A_1 = area of duct, ft.^{2}
A_2 = area of minimum section, ft.^{2}
g = 32.2 ft./sec.^{2}
H' = differential manometer reading, inches
SG_m = specific gravity of manometer fluid
SG_1 = specific gravity of flowing fluid at upstream tap
H = differential head, feet of the flowing fluid at the upstream tap density
Y = compressibility factor
Q = volumetric flow rate, ft.^{3}/sec.
ρ_1 = density of flowing fluid at upstream tap, lb./ft.^{3}
m = mass flow rate, lb./sec.

Alternately, the equations may be combined, yielding

$$Q = \frac{YC_dA_2}{\sqrt{1 - (A_2/A_1)^2}} \sqrt{\frac{2gH'}{12}\left(\frac{SG_m}{SG_1} - 1\right)} \qquad \text{(15-29) (D.E.)}$$

and

$$m = \frac{YC_dA_2\rho_1}{\sqrt{1 - (A_2/A_1)^2}} \sqrt{\frac{2gH'}{12}\left(\frac{SG_m}{SG_1} - 1\right)} \qquad \text{(15-30) (D.E.)}$$

These equations may also be written:

$$Q = \frac{YC_dA_2}{\sqrt{1 - (A_2/A_1)^2}} \sqrt{\frac{2g(p_1 - p_2)}{\rho_1}} \qquad \text{(15-31) (D.E.)}$$

and

$$m = \frac{YC_dA_2}{\sqrt{1 - (A_2/A_1)^2}} \sqrt{2g\, \rho_1\, (p_1 - p_2)} \qquad \text{(15-32) (D.E.)}$$

Where:
p_1 = pressure at upstream tap, lb./ft.^{2}
p_2 = pressure at downstream tap, lb./ft.^{2}

15.10.5.6 Pitot tube as head meter. The three types of head meters discussed above measure the differential static pressure across a constriction in a duct. The pitot tube operates differently in that it measures the total pressure at a point in the duct. Subtracting the static pressure yields the velocity pressure, i.e.,

$$H_v = H_t - H_s \tag{15-33}$$

Where:
- H_v = velocity pressure, or velocity head, measured in feet of the fluid flowing
- H_t = total pressure in feet of the fluid flowing
- H_s = static pressure in feet of the fluid flowing

The velocity pressure, which is equal to the kinetic energy per pound of fluid, is related to the velocity by

$$H_v = V^2/2g$$

or

$$V = \sqrt{2gH_v}$$

Where:
- V = the velocity of flow, ft./sec.
- g = 32.2 ft./sec.^{2}

As previously discussed, H_s may be measured by the use of a static pressure tap (piezometer opening). If a pitot-static tube is employed, H_s is measured at the static pressure openings. In either case, H_v may be measured directly by connecting a manometer between the pitot tube and the static pressure openings. (see Figure 201) It was shown that the velocity may then be calculated from

$$V = C_c\sqrt{\frac{2gH'_v}{12}\left(\frac{SG_m}{SG} - 1\right)}$$

Where:
- H_v' = manometer reading in inches
- SG_m = specific gravity of manometer fluid
- SG = specific gravity of the flowing fluid
- C_c = calibration coefficient (1.0 for a Prandtl pitot-static tube)

Flow velocity varies across a given duct cross section. This is especially true of gases, which tend to spiral down a duct when under moderate or low pressures. The average velocity can be determined by performing a pitot tube traverse, in which velocity pressure is measured at a number of points across the duct and the results are averaged. The

The method to be followed is described in a number of handbooks. National Association of Fan Manufacturers (NAFM) Bulletin No. 110 (pgs. 11 to 27) presents the accepted procedure.

Once the average velocity has been obtained, the following equations may be used:

$$Q = V_{av}A \qquad (15\text{-}34)\ (D.E.)$$

$$m = \rho V_{av}A \qquad (15\text{-}35)\ (D.E.)$$

Where:

Q = volume rate of flow, ft.^{3}/sec.
A = area of duct cross section, ft.^{2}
V_{av} = average flow velocity, ft./sec.
m = weight rate of flow, lb./sec.
ρ = fluid density at the cross section where V_{av} is measured, lb./ft.^{3}

15.10.5.7 Variable-area flow rate meters (Ref. 46 and 47). The variable area type meter provides a direct reading of flow rate. The fluid flows vertically upward through a glass tube which has a small internal taper, with the large end at the top. Inside the tube is a freely suspended "float," shaped like a plumb bob. When no fluid is passing, the float rests on a stop at the bottom end of the tube. With flow, the float rises toward the larger end of the tube, to make a passage for the fluid, and the rise distance is a function of the rate of flow. Readings are made on a scale engraved on the tube for which calibration data is furnished by the manufacturer. Slantwise slots cut in the head of the float cause the float to rotate and to maintain a position in the center of the tube. The basic arrangement of the instrument is shown in Figure 207.

This type of meter permits a more compact test installation, since long pipe sections are not required. For a wide range of flow rates, two meters of different capacity may be installed in parallel and used individually or jointly. To cover the range, meters may be cut in and out of the flow circuit merely by manipulating valves while, when using an orifice meter, the plates must be changed.

15.10.5.8 Anemometers. Anemometers are used for measuring the velocity of flow of a gas. These instruments, which are generally portable, are especially useful in measuring air velocity exiting from a duct system.

Two types of anemometers are in common use: (1) the vane anemometer, and (2) the hot-wire anemometer.

The vane anemometer consists of a series of vanes mounted on an axis which is coupled through a gear train or an electrical generator to some calibrated indicating device. The force of the gas flow causes the vanes to rotate at a speed proportional to the velocity of the flow.

The hot-wire anemometer consists of a short length of fine platinum wire which is heated by an electric current. The resistance to the flow of current through the wire is a function of its temperature. Flow of a gas around the hot wire cools it and thus changes its resistance. By holding either the voltage across the wire or the current through the wire constant, the change in current or voltage, respectively, becomes a function of the velocity of the gas flow across the hot wire.

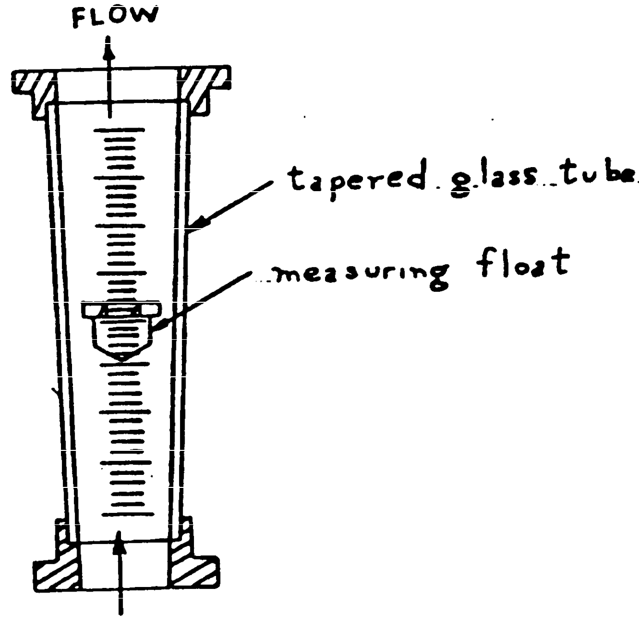


FIGURE 207. Variable Area Flow Meter

The hot wire anemometer, when calibrated, has a reasonable degree of accuracy. However, the vane anemometer, due to its fundamentally mechanical operation, can have a considerable error, because the flow tends to evaporate bearing lubricants, thus changing the friction in the system.

It is recommended that anemometers be calibrated often, since their relatively complex structure is likely to lead to a change in calibration after a short period of usage.

When using an anemometer, it is advisable to make several measurements of the velocity and average the results to calculate flow rate. As when a pitot tube is used, volume and mass rate of flow may then be calculated by equations 15-34 and 15-35. However, if the anemometer is calibrated in feet per minute, the flow rates may be conveniently calculated directly in cubic feet per minute and pounds per minute.

15.10.6 Auxiliary equipment for air flow tests. Meters for measuring air flow cause pressure losses by virtue of their own characteristics, and also because they must often be installed in duct runs of considerable length which offer additional resistance to air flow. Blowers incorporated in electronic equipment for cooling purposes are usually selected to produce static pressure sufficient only to overcome the resistance of the unit at required air flow rates. Adding the resistance of the metering system to that of the unit itself would, in most instances, reduce the blower capacity excessively and would not give a correct indication of blower performance in conjunction with the electronic unit. Therefore, measurement of external air flow through a unit open to the atmosphere or over the heat exchange surfaces of a closed unit is best accomplished by means of auxiliary equipment. The purpose of this equipment is to permit air flow measurement without affecting the performance of the blower installed in the unit.

The basic features of such a set of auxiliary test equipment are shown in Figure 208. Air flow is induced by a centrifugal compressor which, for reasons of economy, is preferably driven by an induction motor at essentially constant speed. Other elements are a duct incorporating a metering section, a flow straightener, a throttling device, and a bleed valve. The metering section has two manometers, one for measurement of static pressure, the other for measurement of differential pressure, and provision for the installation of an orifice plate of suitable size, depending on the range of air flow required. The flow straightener consists of an "egg crate" of flat plates installed in the duct. The plates are typically 1 inch apart, one to two duct diameters long, and made of 0.010 inch brass shim stock. The straightener is necessary to insure accuracy in meter readings by eliminating the spiraling air flow which is characteristic of single-inlet centrifugal blowers. It is recommended that dimension A be at least 10 duct diameters and dimension B at least 5 duct diameters.

The throttle and bleed valves serve to adjust the air flow and pressure level in the duct to the requirements of the electronic unit to be tested. These may be of the damper type or of more elaborate design. In Figure 208, the throttling valve is shown at the blower intake. It is very important to avoid non-uniform, turbulent flow in the air approaching the metering section, and it is suggested that the blower manufacturer be consulted regarding placement and selection of valves.

An auxiliary flow apparatus of the type described should be part of the permanent instrumentation of an electronic equipment development facility, since it is not only useful in thermal evaluation of existing equipment, but can also be utilized to determine required blower characteristics for cooling new equipment under development.

For evaluation purposes, the apparatus can be installed in several ways. The two basic schemes are shown in Figure 209. When the air intake of the electronic unit consists of a single opening to which a duct can be connected, by means of a transition piece from the circular cross section of the duct to the shape of the opening, the apparatus should be installed as shown in Figure 209(A). Since the auxiliary blower is a constant speed device, the bleed and throttle valves must be so adjusted that the static air pressure at some reference location within the unit's air flow passage is the same as measured when the auxiliary system is not attached to the unit and only the unit's blower is used. If this reference pressure is reproduced, the unit's blower is operating under the same conditions. Then, the auxiliary blower serves only to overcome the resistance of the auxiliary system, including that of the metering section. The type of operation of the unit's blower, whether as a discharge or a suction device, and the manner of discharging the cooling air to the atmosphere, whether through a single opening or through multiple openings, have no influence on the installation and operation of the auxiliary system.

When the air discharge from the electronic unit consists of a single opening, the apparatus may be converted readily, as shown in Figure 209(B), so as to be attached to that opening. In that case, the auxiliary blower operates as a suction device. The positions of the bleed and throttling

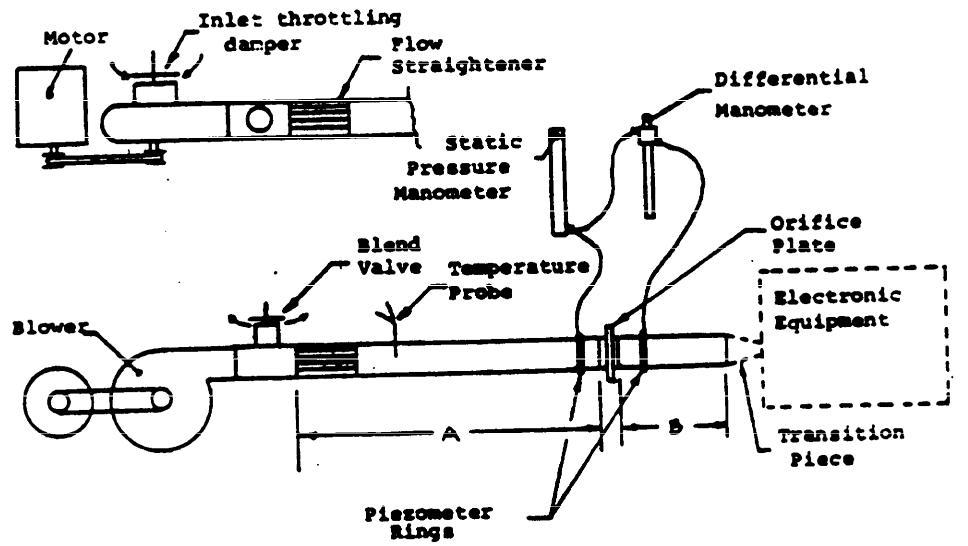


FIGURE 208. Auxiliary Air Flow Test Apparatus (Ref. 46)

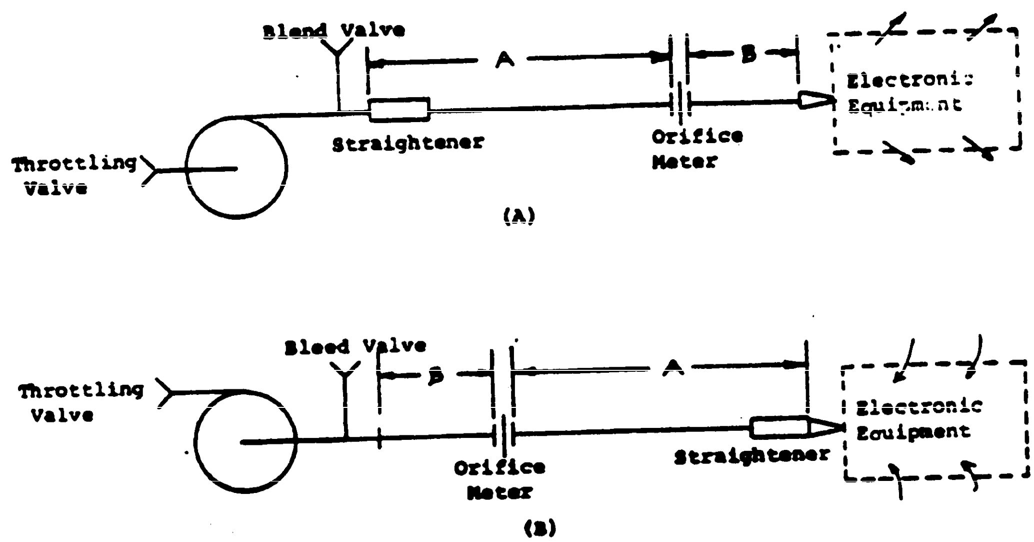


FIGURE 209. Basic Schemes for Application of Auxiliary Air Flow Apparatus (Ref. 46)

valves are interchanged relative to the auxiliary blower and the duct with straightener and metering section is reversed. The operation of the auxiliary blower is controlled as in the previous arrangement by the adjustment of the two valves so as to reproduce a given reference pressure within the electronic unit. While volumetric capacity of the blower is constant, weight flow capacity is reduced since it handles heated air discharged from the unit. Therefore, if a unit happens to have inlet and outlet openings both suitable for connection of the auxiliary system, it is preferable to connect to the inlet opening as shown in Figure 209(A).

With reference to the arrangement shown in Figure 209(A), the reference pressure for the electronic equipment might be the ambient air pressure outside the air intake. In that case, the total pressure at the air intake should be made equal to the reference pressure. This total pressure may be measured with a pitot tube. An alternate precedure would be to install a large plenum chamber between the discharge from the auxiliary apparatus and the intake to the electronic equipment, and to make the pressure in the plenum chamber equal to the reference pressure.

With the arrangement shown in Figure 209(B), the static pressure at the air outlet from the electronic equipment should be made equal to the reference pressure, if that reference is the ambient pressure outside the air outlet (since the velocity pressure at the air outlet will presumably be lost in actual operation of the electronic equipment). Again, an alternative would be to install a large plenum chamber between the electronic equipment outlet and the auxiliary apparatus, and make the pressure in the plenum chamber equal to the reference pressure.

15.10.7 Sample problem. Air (k_c = 1.4) flows through an 8 inch diameter duct. The flow rates, volumetric and weight, are to be found using a 4 inch diameter thin-plate orifice whose coefficient of discharge C_d = 0.67, (see Figure 210). Manometer readings are H_1 = 5.33" H_2O with respect to atmosphere, H' (differential manometer reading) = 6.73" H_2O. The temperature at the upstream tap is 25°C and barometric pressure was measured at 29.42" Hg. Find the volumetric and weight rates of flow.

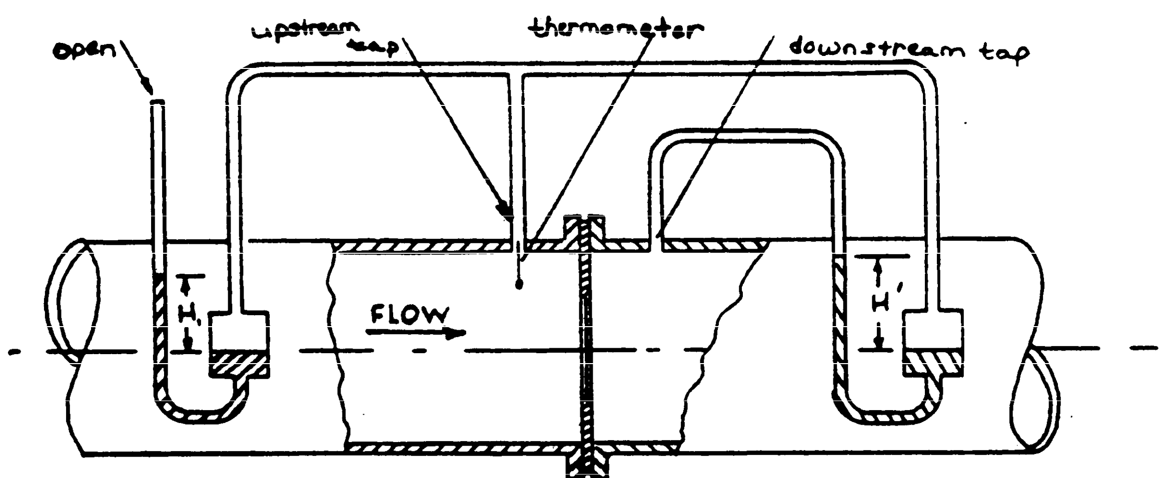


FIGURE 210. Typical Installation for Measuring Flow of Air in a Duct with an Orifice

Solution

(a) Area of orifice opening, A_2:

$$A_2 = \pi d_2^2/4 = \frac{\pi}{4} \left(4 \text{ in.} \times \frac{1 \text{ ft.}}{12 \text{ in.}}\right)^2 = (\pi/4)(4/12)^2$$
$$= 0.0873 \text{ ft.}^2$$

(b) Velocity approach factor, M:

$$M = \frac{1}{\sqrt{1 - (d_2/d_1)^4}} = \frac{1}{\sqrt{1 - (4\text{in.}/8\text{in.})^4}} = \frac{1}{\sqrt{1 - 0.0625}}$$
$$= 1/\sqrt{0.9375} = 1/0.968$$
$$= 1.033$$

(c) System coefficient, C_s:

$$C_s = MC_dA_2\sqrt{2g}$$
$$= 1.033 \times 0.67 \times 0.0873 \text{ ft.}^2 \sqrt{64.4 \text{ ft./sec.}^2}$$
$$= 0.0604 \times 8.04$$
$$= 0.485 \text{ ft.}^{5/2}/\text{sec.}$$

(d) Upstream absolute static pressure, p_1:

Specific gravity of mercury = 13.6. Therefore,

$$p_1 = 29.42 + \frac{H_1}{13.6} = 29.42 + \frac{5.33}{13.6} = 29.42 + 0.39$$
$$= 29.81 \text{ in. Hg}$$

(e) Downstream absolute pressure, p_2:

$$p_2 = p_1 - \frac{H'}{13.6} = 29.81 - \frac{6.73}{13.6} = 29.81 - 0.49$$
$$= 29.32 \text{ in. Hg}$$

(f) Compressibility factor, Y:

$$p_2/p_1 = 29.32 \text{ in. Hg}/29.81 \text{ in. Hg} = 0.983$$
$$d_2/d_1 = 4 \text{ in.}/8\text{in.} = 0.5$$
$$y = 0.99 \text{ (from Fig. 206)}$$

(g) Upstream air density, ρ_1:

$$\rho_1 = 0.0765 \left(\frac{p_1}{p_o}\right)\left(\frac{288}{t_1 + 273}\right)$$

$$= 0.0765\ (29.81\ \text{in. Hg}/29.92\ \text{in. Hg})(288/(25 + 273)$$

$$= 0.0736\ \text{lb./ft.}^3$$

(h) Differential head, H:

$$H = \frac{H'}{12}\left(\frac{SG_m}{SG_1} - 1\right)$$

$$SG_m = 1,\ SG_1 = 0.0736/62.4$$

$$SG_m/SG_1 = 62.4/0.0736 = 847$$

$$H = (6.73/12)(847 - 1)$$

$$= 475 \text{ feet of air}$$

(i) Volumetric rate of flow, Q:

$$Q = C_s Y \sqrt{H}$$

$$= 0.485 \times 0.99 \frac{\text{ft.}^{5/2}}{\text{sec.}} \times \sqrt{475 \text{ ft.}}$$

$$= 0.485 \times 0.99 \times 21.8$$

$= 10.45\ \text{ft.}^3/\text{sec.}$ ANSWER

or, $10.45 \frac{\text{ft.}^3}{\text{sec.}} \times \frac{60 \text{ sec.}}{\text{min.}}$

$= 628\ \text{ft.}^3/\text{min.}$ ANSWER

(j) Weight flow rate, m:

$$m = \rho_1 Q = 0.0736\ \text{lb./ft.}^3 \times 10.45\ \text{ft.}^3/\text{sec.}$$

$= 0.770$ lb./sec. ANSWER

or, $0.0736\ \text{lb./ft.}^3 \times 628\ \text{ft.}^3\ \text{min.}$

$= 46.2$ lb./min ANSWER

16. IMPROVING THE THERMAL PERFORMANCE OF EXISTING ELECTRONIC EQUIPMENT

16.1 General. Existing electronic equipments which have thermal design deficiencies or which are to be used under more severe environmental conditions than they were designed for can often be thermally modified for improved reliability and performance. Typically, such thermal modifications can result in significant improvements. However, a major redesign of the packaging arrangement or cooling system is seldom possible because of economic constraints. Often, a thermal "salvage job" results, but even so the improvements can still be significant.

There are several important factors which must be considered prior to initiating a thermal modification, namely:

(a) How adequate is the existing thermal design?

(b) Can the thermal performance be improved? If so, how much can it be improved and can the modifications be easily implemented?

(c) Is thermal improvement worthwhile?

The following subsections are directed towards resolving these questions.

16.2 Determination of thermal inadequacies. Initially, it is recommended that a thermal reconnaissance be performed on a representative equipment which is suspected to have inadequate thermal performance. This test should be made, if possible, with the equipment in its normal operating environment.

If it is found that selected critical parts are overheating, then a reasonably comprehensive thermal evaluation should next be conducted. It is necessary to obtain sufficient thermal data to establish parts temperatures, thermal resistances in the important heat flow paths, and a heat balance which accounts for all the dissipated heat via the various modes of heat transfer. (Electrical power input = electrical power output plus total heat rejected by all modes of heat transfer.)

Next, the electrical dissipation, ratings, and stress levels of the parts must be established. Sometimes, these data are available from the equipment manufacturer. If not, the schematic wiring diagram should be used as the basis for calculating these data. After the stress levels have been established refer to MIL-HDBK-217 for stress vs. maximum parts temperature data. From this the maximum safe temperatures of the parts can be determined.

Finally, compare the maximum safe parts temperatures with the actual temperatures measured during the thermal evaluation.

If the actual temperatures are significantly greater than the maximum then the equipment is overheating and the thermal design is inadequate. Engineering judgement is required here. A few degrees of overheating is likely insignificant. Usually, 20°C of excess temperature is considered too hot, dependent upon the particular part, its stress level, temperature vs. life sensitivity, and the application. On the other hand, with critical parts, 10°C of excess temperature may be very significant.

16.3 Once it is established that an equipment is thermally inadequate, the problem becomes that of evolving a thermal design which will improve the thermal performance of the equipment while accommodating many major constraints. Constraints typically are: lack of space for larger blowers, fans, heat exchangers, or ducting; the weight cannot change much; blower noise cannot be increased; the quantity and temperature of the coolants supplied to the equipment cannot be increased; blowers or liquid cooling are not permitted; the electrical performance of the equipment cannot be impaired; and economics. Thus, the thermal designer is often faced with a thermal design problem that is much more difficult to resolve than that of the original design or a complete new redesign.

The design procedures presented in this handbook are applicable and it is likely that several alternative designs for the thermal modification will evolve. This leads to tradeoff studies which are next discussed.

16.4 Tradeoff studies of life cycle costs vs. modification costs. There are two bases for justifying the thermal modifications of an equipment:

16.4.1 When the equipment must have improved availability for mission reasons. Vital equipment must exhibit maximum reliability and most any reasonable improvement is worthwhile.

16.4.2 When it can be shown on a life cycle cost basis that it will cost less in the long run if the equipment is thermally improved. The increased reliability results in reduced maintenance costs, logistics, and parts costs, and the increased availability is "fall out." Thus, only equipments which are planned for use for several future years are usually worth modifying. Reference 2 describes in detail the cost analysis of thermally improving three selected shipboard equipments. The modifications were paid for in from five weeks to three years of operation in the Fleet, based on maintenance savings alone, dependent upon the equipment and the quantity modified.

In order to perform the tradeoff study, the following information is necessary:

(a) The number of remaining years of service anticipated for the equipment.

(b) The current maintenance costs, including labor costs, logistics, parts, and support costs, etc., on a per equipment basis.

(c) The current reliability and maintainability.

(d) The predicted improved reliability and maintainability after modification.

(e) The total cost of the contemplated modification, including the costs of R & D, testing, fabrication, installation, retraining personnel, if required, and procurement, on a per equipment basis.

(f) Predicted maintenance costs for the improved equipment and the annual savings on a per equipment basis.

(g) The time required to recover the investment.

(h) Often marginal and break-even cost analyses are desirable.

(i) Such factors as the inflationary costs of increased labor and material and interest on the investment should also be considered.

16.5 Upon modification of the equipment, a complete thermal evaluation should be conducted to validate the predicted thermal improvements.

16.6 Use of indirect fresh water cooling for improving shipboard equipment. One of the advantages of indirect liquid cooling over direct liquid cooling is that it can be applied to the existing equipment without necessity for major redesign.

Some shipboard equipments, which were originally designed for forced air cooling, are operating at temperatures well beyond safe values for reliability. In general, this situation has been caused by installation in confined spaces, along with other equipments of high heat dissipation. Often the space is almost uninhabitable because the air ventilation system is inadequate for the heat load and no space is available for additional air cooling ducts. During the interim period, until more effectively cooled equipments and spaces are provided, it is suggested that the cooling of such equipments be supplemented through the utilization of air to fresh water cooled heat exchangers in the air system or water cooled cold plates attached to the inside of the equipment enclosures. When water is circulated through the cold panel, the original metal enclosure will act as an additional heat exchanger surface. In either case, the internal air should be cooled by the exchanger and recirculated over the equipment. Louvers and openings in the enclosure should be covered so that none of the internal air leaves the system. This will alter the free and forced convection currents inside the equipment and it may become necessary to divert the air flow along new paths to move the heat from the parts to the surface of the enclosure. Increasing the rate of circulation of the air by the addition of new fans, or redirecting the output of existing fans, should create more uniform internal temperatures.

17. THE THERMAL CHARACTERISTICS OF PARTS

17.1 General. Electronic parts are designed for two basic types of cooling; i.e., by natural means at sea level or by special means. Those parts designed for cooling by natural means also fall into two sub-categories, i.e., conduction cooling, or radiation and convection cooling. For example, resistors are designed for free air cooling by convection and radiation; power transistors are designed for conduction through their bases to heat sinks; and high power klystrons may be designed for liquid, vaporization, or forced air cooling. The internal thermal design of a part is influenced by the type of external thermal system into which it is expected to transfer heat. The external surfaces of parts are often designed to minimize the temperature rise across the interface to the external thermal system.

Many of the parts developed a decade or more ago were designed and rated for natural cooling by convection and radiation in free air. These parts were intended to be widely separated in free ambient air with no thermal interaction among them. Unfortunately, the dense packaging of parts in modern equipment results in thermal coupling among the parts by conduction, radiation, and convection. Thus, these parts operate thermally differently from the modes intended. The situation is further complicated by the "ambient air" ratings which should be but are not given in terms of parts surface temperature. Modern parts, such as transistors, are rated in terms of surface temperature at given locations.

The effects of heat on parts failure rates are discussed in paragraph 5.3.2, and Figure 1 (pg. 23) presents the equivalent thermal circuit of a simple electronic part. The determination of the maximum safe parts temperature consistant with the required reliability should be made in accordance with MIL-HDBK-217.

17.2 Thermal characteristics of semiconductor devices.

17.2.1 Semiconductor devices, general. Semiconductor devices are peculiarly sensitive to extremes of temperature and to temperature variations. The operation of these devices depends on the mobility of charge carriers moving through a crystal lattice, which is intrinsically rigid. However, the molecules are in continuous vibration, the amplitude of which increases with temperature. High temperature therefore affects the motion of the charge carriers, and affects the device eletrical parameters. Furthermore, the devices are fragile internally, are built up of materials having different coefficients of thermal expansion, and have very small heat capacities. Temperature variations will therefore result in mechanically stressing the device material. Chemical reactions are possible among the constituent materials, and reaction speed increases exponentially with temperature. Such phenomena as "purple plague" are therefore enhanced by high temperature.

Exposure to extreme low temperature, and temperature cycling over a wide range, may cause fatigue failure and mechanical fracture, or permanent distortions resulting in a change of operating characteristics.

Increasing temperature causes an increase in leakage current and current gain, resulting in increase of power dissipation which in turn raises the temperature. If not checked, this cycle results in "thermal runaway" and a temperature high enough to cause catastrophic failure. Lateral thermal instability has been observed in a number of different power transistors. At a certain critical temperature rise, current density and temperature tend to build up in one small region of the device, resulting in a hot spot. Because current distribution is assumed to be uniform over the entire area of the transistor, the temperature at the hot spot may be very much higher than had been expected. These hot spot temperatures produce localized alloying and diffusion resulting in early failure of the device. It is believed that most secondary breakdowns observed on transistors results from the development of a hot spot. Since the destructive temperature of semiconductors is low compared to that of many electrical devices, and heat capacity is small, thermal runaway can result in rapid destruction.

In general, the operating characteristics of solid state devices vary with internal temperature. Some characteristics increase or decrease monotonically with temperature; some exhibit maxima or minima. In general, there are no apparent relations among the temperature characteristic functions, so that devices matched at one temperature may not match at higher or lower temperatures. Precautions must be taken in solid state circuitry design to provide temperature compensation and to prevent thermal runaway.

Failure of a device may be catastrophic, or it may consist of circuit function degradations, either temporary or permanent. The thermal time constants of solid state devices are short, and periodic load variations can cause signal distortion due to changes in characteristics with temperature. Such an effect is observed at frequencies under 2 KHZ. It is desirable to maintain the temperature of solid state devices at a relatively constant level.

Manufacturers of solid state devices provide useful thermal data, including curves of operating parameters vs. temperature, maximum and minimum storage temperatures, maximum junction operating temperature, and one or more pertinent thermal resistances. As with all data pertaining to solid state devices, unless specially selected preium parts are specified, deviation from the mean reported values is large, 20% or more.

The stated maximum juction operating temperature must be derated by circuit designers with reference to failure rate vs. temperature data, so that the desired reliability is achieved. Since equipment and system reliability are functions of parts failure rates, design junction temperatures can be determined only through careful reliability design. A common design error is to compute worst case semiconductor junction temperatures, and to assume that the thermal design is adequate if the manufacturer's maximum operating junction temperature is not exceeded. While the device may function under such conditions, its reliability, or life, will generally be so low as to be unacceptable. Maximum allowable semiconductor junction temperatures are meaningless unless related to required system reliability. This normally will require considerable derating of the manufacturers data. For example, silicon semiconductors are normally rated by manufacturer's at 200°C maximum junction temperature, but to achieve the required equipment reliability, junction temperatures may have to be limited to 150°C or even 125°C. In addition to the requirements of system reliability, maximum junction temperature derating is advisable to provide some margin of analytical error, to allow for non-uniform heating ("hot spot" development) without catastrophic failure, and to allow for system electrical transients.

Since thermal design of circuits employing semiconductors is necessarily related to the semiconductor junction temperature and since the junction does not in itself interface directly with the environment, some functional parameter must be used to relate junction temperature to the environment. Manufacturers supply a value of internal thermal resistance, in °C/watt. This resistance preferably is from the device junction to some part of the device mounting, typically the case, the base substrate, or the mounting stud of power devices. Since these points can be thermally related to the environment, a complete thermal circuit from junction to environment is available for analysis. Some devices (particularly small signal devices) are rated in terms of junction-to-air thermal resistance. This value is determined by deriving the junction temperature while the device is operated when suspended in free air. This is a poor rating technique, since it includes not only the device internal resistance, but also the effects of external natural convection, radiation, and lead conduction. When devices with this type of rating are contemplated for a design, it is often possible to obtain junction-to-case thermal resistance values by contacting the manufacturer. Otherwise the junction-to-case thermal resistance must be estimated, computed, or determined by test, unless dissipations are low enough that junction-to-air thermal resistance may be used with confidence. Occasionally, a derating curve similar to Figure 211 is provided in lieu of a value of thermal resistance. The derating curve generally specifies a maximum power (p_{max}) at some base temperature of the case, T_B (usually 25°C), and a maximum case temperature. Junction-to-case thermal resistance R_{jc}, may be determined by

$$R_{jc} = \frac{T_{max} - T_{Base}}{p_{max}}$$

For example, if, as shown in Figure 211, p_{max} = 5 watts, T_B = 25°C, and T_M = 175°C, then

$$R_{jc} = \frac{175 - 25}{5} = \frac{150}{5} = 30°C/watt$$

Some examples of thermal resistance and heat transfer calculations for solid state devices have been given in chapter 8.

17.2.2 Low-power transistors. Low power transistors are frequently overlooked or ignored in equipment thermal analysis because of their low dissipation, occasionally leading to disastrou results. The stabilization temperature of any component is a function not only of its dissipated power, but also of the thermal resistance to the environment. Unless a suitably low resistance path is provided, excessive temperature differentials

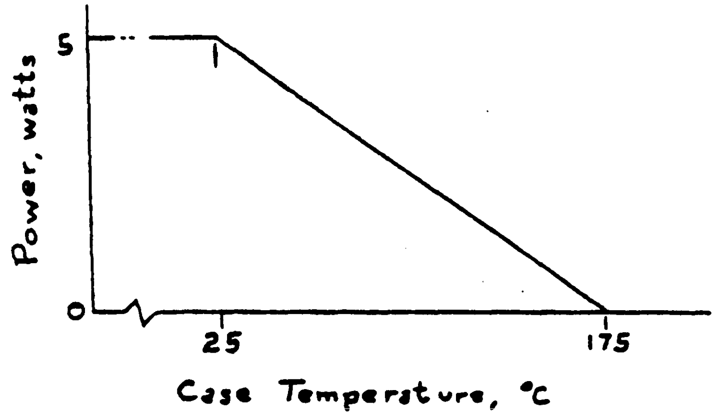


FIGURE 211. Allowable Semiconductor Case Temperature vs. Power Dissipation

will be required to drive the heat through the thermal resistance, even though the heat may be in terms of milliwatts.

17.2.2.1 Thermal rating methods. Low power transistors are rated thermally by a variety of methods, the most common of which are discussed below.

Junction-to-case thermal resistance (R_{j-c}). This is the preferred method of thermal rating, since it allows computation of a continuous thermal resistance path from the junction to the environment. Two precautions are necessary when using this type of rating. First, for many-lead devices (10 or more leads), the thermal resistance through the leads and circuitry and substrate may be low enough to be significant in relation to the thermal resistance to the case and mounting structure. Normally, however, the lead resistance is several times larger than the resistance through the case, and may be safely ignored. Secondly, the outer case temperature may vary widely, and the point on the case to which R_{j-c} is referenced should be known. Usually this reference point is the base of the case. Since it is usually more convenient to attach thermocouples to the top of the case in instrumental tests, this temperature variation from the base to the top may lead to erroneous conclusions unless accounted for.

Junction-to-air thermal resistance (R_{j-A}). This is a common method of rating low power transistors. It is applicable only when components are widely spaced, so that interactive effects are minimized, and when heat transfer to the surrounding air by convection is the primary heat transfer path. In using this parameter to determine a junction temperature, the air temperature must be evaluated carefully. The temperature of the air local to a given transistor will rarely be as low as the "ambient" air temperature of the system, due to enclosure effects and effects of neighboring components. This parameter further assumes no impediment to the generation of natural convection cooling air circulation, and no radiant heat load on the component.

Maximum component power level (P_{max}). This rating method was once widely used for low power transistors, and is still fairly common, although it is generally being replaced by more definitive rating methods. It is the maximum power level which may be dissipated by the device under a given operating environment (usually air temperature or case temperature = 25°C) to maintain a given maximum junction temperature (usually 175°C or 200°C). If the operating environment condition and the maximum junction temperature are given, it may be converted to R_{j-c} or R_{j-A}. Thus, for example, if a component is rated at 200 milliwatts in 25°C air for a 175°C maximum junction temperature, then $R_{j-A} = \frac{175 - 25}{0.2} = 75°C/watt$. Similarly, if a component is rated at 1/2 watt at a case temperature of 25°C for a 200°C maximum junction temperature, then $R_{j-c} = \frac{200 - 25}{0.5} = 350°C/watt$. Unfortunately, the operating environment and/or maximum junction temperature are not always explicitly given, particuarly for older components. Unless these parameters are known, the internal thermal resistance, and subsequent computation of junction temperature, cannot be determined. The designers only alternatives are (1) to consult with the manufacturer in hopes of obtaining the required data; (2) to obtain the data by test, which may be expensive, and misleading because of the device manufacturing tolerances; (3) assume data, which can be dangerous (very generous safety factors should be used); or (4) to use a different device for which data are available. Do not assume an equivalent internal thermal resistance of a device based solely on its external physical resemblance to another device for which a value is given. Extreme variations exist in internal thermal resistances of different devices packaged in the same physical configuration (e.g., a TO-5 package).

Derating curve. This is the type curve shown in Figure 211 which may be converted into R_{j-c} or R_{j-A}.

17.2.2.2 Heat transfer mechanisms. Early design of transistorized equipment relied upon natural convection from the case, with minor contributions from radiation and lead conduction, to remove the heat without excessive temperature rise. This is one of the reasons for the junction-to-air thermal rating method. This cooling method was adequate with low dissipations per component, and with widely spaced circuitry. As equipment was reduced in size, circuitry became more and more closely spaced, and the devices themselves became smaller, with increased unit heat concentration. The limitations of natural convection cooling by way of the case became apparent, as equipments began experiencing failures due to failure of low power transistors, either catastrophically, or by variation of their parameters with high temperature. Natural convection cooling of low power devices should be recognized as a high thermal resistance path, suitable only for dissipations of 500 milliwatts or less (dependent upon device), and requiring free circulation of air around the components. This last limitation also makes natural convection cooling somewhat dependent on exterior characteristics, since changes to the surrounding structure or neighboring modules could change the air circulation pattern around the component, thereby changing its equilibrium temperature.

Heat transfer through the leads of low power transistors is also a high resistance path. For example, assume a low power transistor with three leads, each 0.25 inches long. Transistor leads are normally 0.017 inches diameter (total area = 227 x 10^{-6} sq. in.). If the leads are copper (k = 10 (watts/sq. in.)/(°C/in.)), then the external lead thermal resistance, R_L is

$$R_L = \frac{L}{kA} = \frac{.25}{(10)(3)(227)(10^{-6})} = 36.7°C/watt.$$

This is the lead external thermal resistance only. Internal to the case, the lead thermal resistance may be orders of magnitude higher due to the fine wire leads used for internal connections. The leads terminate in printed circuitry on a circuit board, which is usually significantly higher in temperature than the air. Consequently, overall lead resistance is usually very high, and this thermal path normally may be ignored in thermal circuits.

Thermal conduction from the case to the substrate is the preferred heat transfer method, if the thermal resistance can be made sufficiently low, since this path is controlled within the module to which the transistor belongs. The heat transfer area is generally large (the full base area of the transistor), and the thermal path relatively short. The thermal effectivenss of this path is subject to the mounting conditions.

17.2.2.3 Low power transistor mounting. Low power transistors were originally (and often still are) mounted by soldering their leads into a circuit board. Thermally, this is acceptable if natural convection air cooling is acceptable. Base conduction is minimized, since it is necessary to provide an air gap between the transistor base and the circuit board to allow for thermal expansion effects and avoid mechanical stressing of the leads.

The air gap may be avoided by using transistor mounting pads, by stress relief bends in the leads, or by a combination of the two. Transistor mounting pads are thin devices molded or machined to the base size of the transistor package, with holes or slots for the transistor leads. Some pads are semi-resilient plastic materials, which offer sufficient mechanical compliance to compensate for thermal expansion without excessive lead stress. While their thermal conductivity is low, it is much better than that of an equivalent air gap, and the overall thermal resistance through the base may be reduced to an acceptable value. Other pads are of high conductivity (metallic), with insulating layers (anodize) if necessary for electrical isolation. These pads provide excellent thermal paths. Transistor leads may contain stress relief bends for expansion effects, and a spring device to maintain base contact pressure is often used.

Base conduction heat transfer may be enhanced by leaving as much copper as possible on the circuit board beneath the transistor. This is particularly true for low conductivity boards, such as glass-epoxy laminates. The copper acts to diffuse the heat over a large area to avoid excessive temperature rise. A continuous wide copper trace to the next thermal level (e.g., the board side guides and chassis) is desirable. All interface resistances can be minimized by the use of thermally conductive compounds. (see chapter 8)

17.2.2.4 Other conduction paths. Where natural convection cooling and/or base conduction cooling are not adequate to transfer the heat of a low power transistor, a simple conductive strap will often provide the necessary low thermal resistance path. For example, the 2N3414 transistor of Figure 212 is rated by the manufacturer at 360 milliwatts. The 2N3402 transistor of Figure 213 is rated at 900 milliwatts (Reference 127). This is basically the same transistor with the addition of a metallic conduction clip. Clips should be designed to grip the transistor body firmly, or filler materials should be used at assembly. Care must be taken not to stress transistor leads as a result of attaching the clip to a thermal heat sink.

17.2.2.5 Convectors. A variety of devices is available, to fit any standard transistor package outline, to reduce the thermal resistance to air, whether by natural or forced convection. These generally consist of a series of fingers protruding into the airstream, projected form a base which may be attached to the transistor case either by spring action or by mechanical mounting. Surfaces may be insulated (usually by anodize on aluminum) if required. In natural convection processes, the primary effect is one of increasing the convection area. In forced air applications, the reduction of thermal resistance is due to both the increase of area and the creation of local turbulence to reduce the air boundary layer thickness. Manufacturers offer thermal resistance data for these devices for both natural and forced convection applications.

17.2.3 Power transistors.

17.2.3.1 Ratings. Manufacturers of power transistors recognize that thermal deficiencies are a prime potential cause of device failure, and they design power transistors to provide low resistance thermal paths internally from the junction to the case structure. The base is relatively thick, to provide for the diffusion of the heat across the base area, and to incorporate thermal capacity to provide some protection against thermal transients. Because of the emphasis on the low thermal resistance internal paths, ratings are usually given in terms of thermal resistance, collector junction-to-case (R_{j-c}), in °C/watt. Maximum power ratins are sometimes given, but usually in terms of a specific case temperature. Then,

$$R_{j-c} = \frac{T_j - T_c}{p}$$

where R_{j-c} = junction-to-case thermal resistance, °C/watt
T_j = maximum junction temperature, °C
T_c = stated case temperature, °C
p = maximum rated power, watts

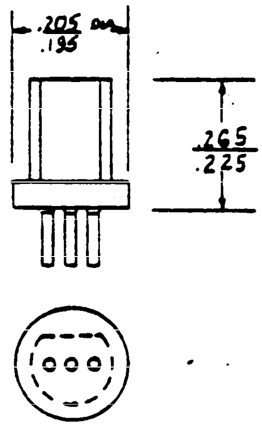


FIGURE 212. 2N 3414 Transistor Outline

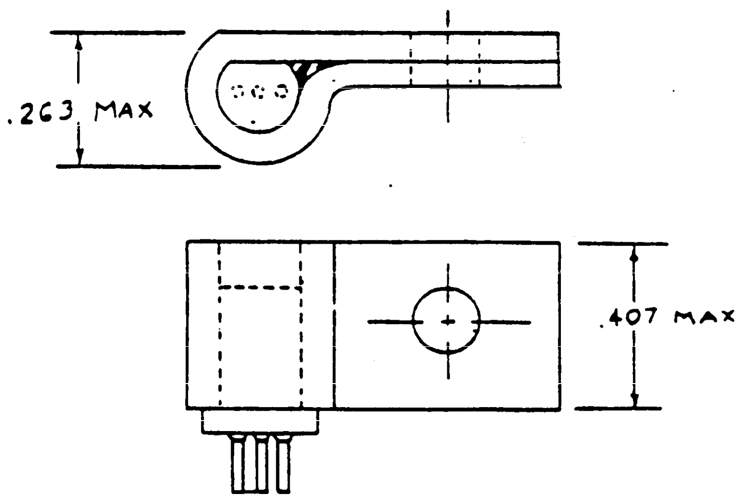


FIGURE 213. 2N3402 Transistor Outline

Because of the heavy base or stud used in power transistors, temperature variation in the base is likely to be minimal. Variations of 5-10°C are possible at device power levels on the order of 50 watts dissipation. Lower power levels will, of course, yield lower variations. In temperature tests by use of thermocouples, insertion of the thermocouple is a small hole drilled into the base or stud is recommended.

17.2.3.2 Heat transfer mechanisms, mounting. The heat transfer mechanism for power transistors is almost invariably conduction through the base. (The term "base" in this context includes the threaded studs of stud mounted transistors.) Consequently, the mounting surface must be flat and smooth to minimize the interface resistance. The use of thermal conductive compounds at the interface is commonplace, and highly recommended. The heat sink to which the transistor is mounted should be of high conductivity (metallic) and sufficient thickness to assure rapid diffusion of heat. In analysis, assume all heat flow by conduction through the base. Convection and radiation from the case will be small in comparison. (Possible exceptions include forced direct liquid cooling or boiling heat transfer, where appreciable heat transfer from the case into the cooling medium may occur.)

17.2.3.3 Convection cooling. Forced air convection cooling of power transistors is accomplished with metallic heat sinks having extended surfaces (fins or pins) to increase the heat transfer area into the airstream and to develop local turbulence. Standard commercial sinks are available, drilled for any of the common transistor package and lead configurations. Manufacturers provide data in terms of thermal resistance vs. airflow rate.

17.2.4 Integrated circuits. The junction-to-case or junction-to-air thermal resistance concept which originated with transistors is generally applied as well to integrated circuits (IC's). I.C.'s, however, contain several semiconductor junctions within the case. The manufacturer will generally provide worst case thermal resistance values or maximum power ratins. The internal heat path is primarily directly from the clip into the base. Similar to transistors, the lead thermal resistance is high, due to the relatively fine wires used for connections internally. Conduction from the base should be emphsized in external heat flow paths. Resilient mounting pads, spring loaded clips to insure base contact, and thermal compounds at the interfaces are recommended, similar to transistor applications.

17.2.5 Medium-and-large-scale integrated circuits (MSI and LSI). For medium and large scale integrated circuits, the junction-to-case thermal resistance concept becomes nebulous, since there may be hundreds or even thousands of junctions on a single large chip. Since the circuits are almost always digital, the dissipation per junction is generally small, but the total device dissipation can be significant. These devices essentially become modules in themselves. Usually, a maximum allowable substrate temperature can be specified by the manufacturer for reliable operation, and the maximum dissipation can be derived by the manufacturer, the user, or both. The thermal problem then becomes one of providing a sufficiently low external thermal path or paths to maintain the substrate at or below the maximum allowable temperature at the specified power level. Intimate thermal contact is required between the substrate and the mounting surface. This necessitates a smooth machined surface for mounting, and usually the use of interface compounds. Once the heat is transferred into the device mounting, any of the heat transfer methods described in this handbook may be used to transfer it into the path to the ultimate sink. The mounting structure must have a high thermal conductivity normal to the mounting surface to insure a relatively uniform temperature of the device substrate. Since MSI and LSI devices contain so much circuitry, and represent a considerable expense, it is customary to provide ultra-reliable cooling systems. Forced air and liquid cooled heat sinks with temperature controlled coolants are common. In any case, close liaison between the user and the manufacturer of the device is essential for reliable application.

17.2.6 Rectifiers and diodes. Small diodes and rectifiers are generally similar in thermal characteristics to low power transistors. Equivalent thermal rating methods are commonly used. In the very small physical sizes, heat transfer through the leads into the substrate may be the predominant cooling method, and short lead lengths will help provide adequate cooling. With medium size units, a peculiarity associated with diodes and rectifiers is the relatively high voltages applied. These high voltages necessitate heavier insulators (mica or beryllia), which add thermal resistance.

17.2.7 Microwave devices. Manufacturers of microwave devices, like those of high power transistors, place a high degree of emphasis on thermal design, since these devices are usually extremely temperature sensitive. Internal thermal resistance values are minimized, to the extent that diamond mounts are used for some devices. (Within the normal temperature range encountered for electronics, Class II diamonds have a lower thermal resistance than silver.) Comparable thermal design efforts must be followed external to the device to ensure the transfer of heat into the ultimate sink without excessive temperature differentials.

Ceramics, particularly beryllia, are extensively used in packaging microwave devices, because of the favorable combination of high thermal conductivity and electrical insulation properties. The ceramics can be metallized directly to provide leads in hermetically sealed cases without the need for glass seals, thereby improving strength and reliability. Since ceramics are brittle materials, the thermal design of devices using them should include an analysis of temperature expansion effects relative to the mounting, to avoid excessive mechanical stress buildup.

17.3 Thermal characteristics of electron tubes.

17.3.1 Modes of heat transfer in electron tubes. The transfer of heat within an electron tube is a complicated process. A high temperature emitting surface is necessary to maintain electronic emission. Heater temperatures range from 1000°C, and cathodes operate in the neighborhood of 750°C. To reduce heater power to a minimum, tube structures are designed so that the thermal resistance from the heater and cathode to the envelope and external leads is as great as possible. However, tubes must also have short leads from the internal elements to provide electrically low inductances and low resistances into their external circuits. These leads conduct heat from the cathode or filament and a compromise between these two incompatible requirements results.

Most of the heat dissipated in an electron tube appears at the plate. Not only is the plate heated by its normally dissipated energy, but much of the heat originating at the filament, cathode, control grid, and screen grid is transmitted by radiation through the vacuum into the plate. The remainder of the heat produced by tube elements other than the plate is radiated into the tube envelope and/or conducted into the tube pins along the tube element leads.

Plate temperatures in electron tubes, other than transmitting types, range from 350°C to as high as 600°C. Most of the energy dissipated by a plate is transmitted through the vacuum by radiation and is transmitted through or absorbed by the glass envelope dependent upon the configurations and relative temperatures. Due to its transmission characteristics, electron tube glass usually begins to be a poor transmitter of infrared radiation at 2.5 microns. Thus, it is semi-absorbent radiation wise. For example, most tube glass is essentially opaque to radiation from sources near 350°C and only 3% of the energy radiated from a plate at this temperature may be transmitted directly through the average glass envelope. The remaining 97% of the heat radiated from a plate at 350°C can be absorbed by the glass. (see Figure 214) The glass is heated and reradiates part of this energy at a lower temperature level and convects or conducts the remainder to the environment. Some heat from the plate (5 to 10%) is conducted along the plate lead through the tube pins. Conversely, when plates operate between 400 and 500°C a larger increment of the radiated energy may be directly transmitted through the glass. If plates operate at temperatures of the order of 750 to 850°C (cherry red), as in tantalum element transmitting tubes, a significant portion of the radiation from the plates passes directly through the glass, due to the shorter wave length of the radiation spectra. High temperatures are required for the removal of the heat from the plates of such tubes, since the emissivity of bright metal plates is relatively low. However, the glass in this type of tube is heated less than in a carbon plate tube. Conversely, because a larger portion of the radiation passes directly through glass, parts adjacent to tubes with high plate temperatures will be absorbing more of the rejected heat.

The total radiation which will be transmitted through a given glass envelope is influenced by several variables including:

(a) The emissivity of the plate. Plates range from bright metal to carburized and blackened metal and their emissivities will vary with temperatures.

(b) The temperature of the plate and the temperature of the glass.

(c) The angle of incidence of the radiation with respect to the glass.

(d) The transmittance characteristics of the glass. This varies with the type and mix of the glass.

(e) The thickness of the glass.

(f) The temperature and emissivity of the objects or the media surrounding the glass.

The above modes of heat rejection from within an electron tube result in a concentration of heat in the glass envelope adjacent to the plate and to some extent at the base of the tube. If a tube is mounted vertically and operated in free air, a small hot spot, due to conduction through the leads, will appear at the base and the envelope will have a definite hot spot at approximately two-thirds its height, opposite the plate, due to radiation through the vacuum. Glass is a relatively poor heat conductor and temperature gradiants will appear in the envelope adjacent to the upper and lower edges of the plate structure. It is desirable to cool tubes in a manner that will reduce such gradients in the envelope. Large temperature differences can cause severe mechanical strains which lead to envelope breakage.

The feasibility of removing heat through the leads and pins of miniature and octal tubes and into their sockets has been investigated. In general, less than 10% of the heat can be removed through the leads of miniature tubes such as the 5AQ5, when operated at full rating. Because of their long lead lengths, octal tubes and flying lead subminiature tubes have a relatively high thermal resistance through their leads and significant heat rejection does not occur along the path unless the tubes are operated at reduced ratings and/or the thermal resistance from the plate and envelope to the sink is high.

17.3.2 The effects of overheating and temperature limiting factors.

17.3.2.1 Electron tube glass (general). The upper temperatures of electron tubes are sometimes limited by the physical characteristics of the glass bulbs. Overheating of the glass can ultimately lead to serious malfunctions. Receiving type tube bulbs are usually made of soda-lime-silicate glasses.

17.3.2.2 Electrical properties of electron tube glass. Electrical conductivity in glasses is ionic, that is, the current is carried by the movement of ions as in electrolytes rather than by free electrons as in the case of metals. The sodium ion is responsible for by far the greatest part of this effect. As a consequence, the conductivity rises sharply with temperature and with the amount of sodium in the glass.

Resistivity and other dielectric properties are subject to considerable variation with the thermal history of the glass. The volume resistivity is temperature sensitive and can vary as much as five decades between 50 and 250°C. Conduction and electrolysis can occur readily in glass at high temperatures.

Electron tube glass failures can also be associated with high voltage stresses of long duration and at high temperatures, which represent a condition of thermal instability. The dielectric losses occurring under the action of the voltage stress raise the temperature of the dielectric. This higher

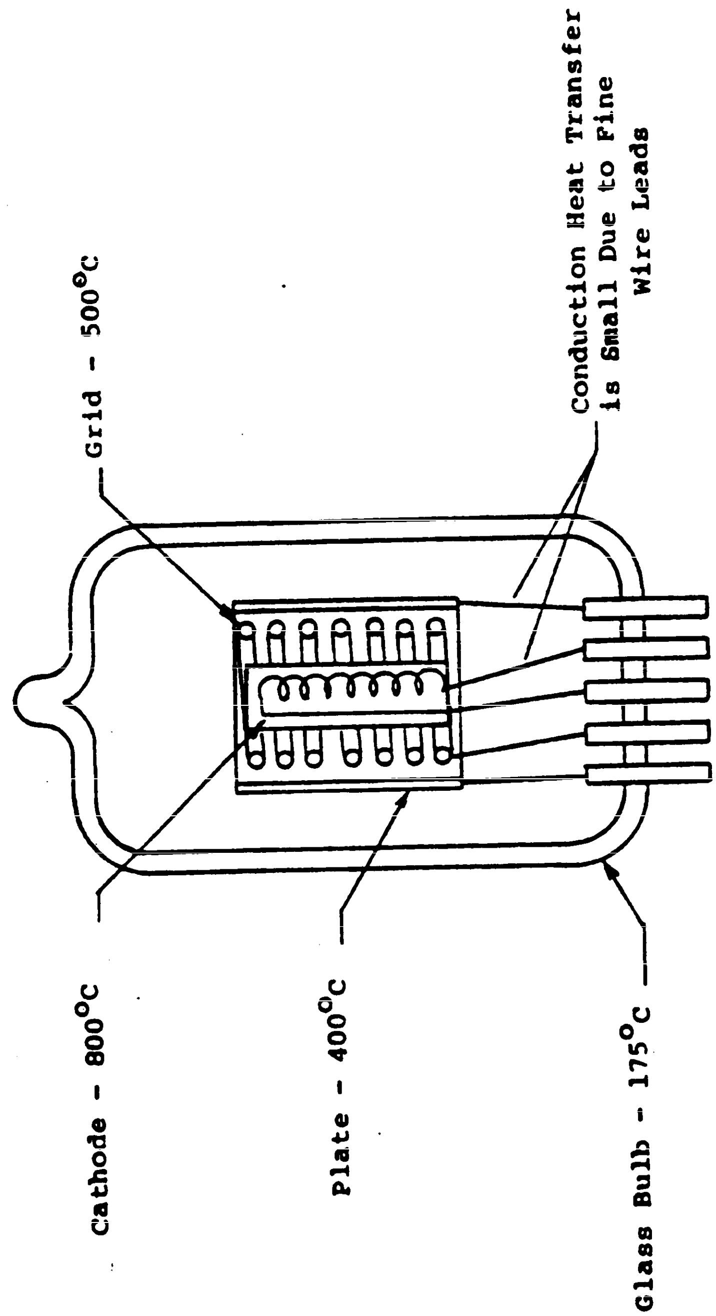


FIGURE 214. Heat Transfer in an Electron Tube

temperature increases the losses and consequently raises the temperature still more. At some voltage, or after some period of time, this action becomes unstable; the current, usually in some restricted area, rises to high values because of the lowered resistivity, until the dielectric fuses or vaporizes, which leaves the characteristic puncture. For this type of failure, it has been found that the breakdown voltage drops with temperature as a function of the resistivity. This is applicable to high voltage rectifier tubes and radar modulators, etc.

The glasses with the best electrical characteristics are seldom used in electron tubes, due to undesriable chemical or mechanical properties. Electron tube glasses must be rugged and easily annealed for shaping and sealing; they must not contaminate the vacuum; they must exhibit thermal expansion compatible with the metal leads; and they must form a good hermetic seal at all joints including leads. In many instances the bulb glass differs from the glass at the seal containing the metal leads. This technique permits the utilization of "high temperature" glasses opposite the plates at the hot spots and the necessary glasses at the seal.

17.3.2.3 Mechanical properties. The mechanical properties of electron tube glasses also warrant special consideration. Glass is generally brittle and breaks easily. It is also susceptible to thermal shock. Its tensile strength is low and fractures occur after long periods under sustained loads considerably below its breaking strength. Further, small scratches can grow in time, leading to cracks and subsequent failure. The glass envelopes of electron tubes frequently are under considerable stress due to their internal vacuum and the stresses formed during fabrication or processing. Unfortunately, tempered glasses cannot be utilized for electron tubes. Thus, any electron tube cooling device must nost impose thermal or mechanical stresses on the fragile glass bulbs.

17.3.2.4 Thermal properties. Transmissions of radiations of various frequencies by glass is of practical importance in many electronic applications. When heat is dissipated within an electron tube, the amount absorbed by the envelope will limit the operating level because of the resultant temperature rise.

Because the operating spread of plate temperatures falls within the infrared absorption or transmission range of vacuum tube glasses, it is sometimes possible for a bulb to be at a normal temperature when the internal elements are overheating, and conversely, it is also possible for the glass to be hotter than normal while the internal elements are at normal operating temperatures. It has been found that the plate temperature is the best index of the true thermal condition of a tube. Thus, a well cooled tube may, under certain conditions, exhibit a higher glass temperature than a poorly cooled tube. The glass temperature should be as low as possible, but not at the expense of higher internal element temperatures.

In general, plate and bulb temperature of a glass electron tube at a constant power dissipation will increase or decrease in unison, but in varying degrees, dependent upon the temperatures, emissivities, and transmittances of the surfaces the radiating plate "sees." As the cooling is improved, the plate temperature will tend to "level off," while the bulb temperature continues to drop. However, the plate temperature "plateau" is seldom reached with tube shields, and then only when the unit heat dissipation and concentration are low. The "plateau" condition can be achieved, however, with adequate forced air and liquid cooling.

Chapter 8 discusses radiation in further detail. Calculation of the temperature rise of a glass envelope due to the absorption of heat from a radiating hot anode or filament can be accomplished from these data provided the thermal emissivities and pertinent temperatures of the sources are known. These quantities, unfortunately, are not always available. One authority recommends a limiting unit heat dissipation of as high as 5 watts/sq. in. for a glass thickness of 1/16 in. in quiet air.

In general, the thermal conductivity of glass is relatively low, being in the neighborhood of .03 watts/(in.^{2})(°C)/in. for electron tube glasses. Even though the thermal conductivity is low, the temperature gradients through the walls of electron tube bulbs are small. This is due to the relatively large glass surface areas with respect to the total dissipation. These reasonable unit heat dissipations together with the thin glass walls produce small temperature gradients from inside to outside of receiving type electron tube bulbs would be of the order of 2 to 4°C. Measurements generally agree with these calculations.

While the gradients through electron tube glasses are small, the gradients along the outer surface of the envelopes due to conduction can be very large (as much as 100 to 150°C), if a non-uniform method of cooling is used. Such high temperature gradients can produce severe stresses in the glass and ultimately lead to glass breakage. It is, therefore, necessary that electron tube bulbs be cooled in a manner which results in a uniform distribution of heat on the surface of the bulb. In general, the hot spot temperatures should not exceed the average temperatures by more than 35 to 40°C.

17.3.2.5 The effects of overheating glass envelopes. Glass temperatures in excess of 250°C will usually produce significant ionic conduction and electrolysis between tube leads. Softening and vacuum sucking of glass can be initiated at slightly higher temperatures. Operation of tube glasses in this range of temperatures also results in severe tensile stresses in the bulbs. Generally, temperatures above 250°C will cause the above effects to occur simultaneously so that the bulb will soon fail. Hot spot temperatures of those tubes utilizing conventional glass bulbs should not be greater than 175°C for reliable service. "Hard" glass premium tubes may be operated with hot spot bulb temperatures as great as 250°C but for reliable service less than 200°C is recommended.

17.3.3 Electron tube elements.

17.3.3.1 General. The release of gas and the migration of getter and cathode materials is greatly accelerated when tube elements exceed certain temperatures. Any of these deleterious effects will cause short tube life and malfunctions of circuitry.

17.3.3.2 Processing limitations. Receiving type electron tubes are usually outgassed with the internal elements heated to approximately 500°C and the glass heated to 250 and 300°C. Should these temperatures be exceeded during later service, the immediate release of excessive gas occurs, the getter is unable to absorb the gas, and a gassy tube will result.

17.3.3.3 Getters. Electron tube getters will usually start to release small amounts of their absorbed gas when the glass bulbs are operated at temperatures greater than 200°C. Further, the gas may later condense on the grids when a tube is cooled or minute particles of the getter material may migrate to the grids. Under these conditions, grid emission can be accelerated by high temperature operation.

17.3.3.4 Grids. Excessive grid temperature can cause several serious effects. Since the control grid is physically located closely and adjacent to the cathode or filament, its temperature is normally high, being between 200 and 400°C due to heat exchange by radiation. Relatively small increases in temperature will thus place this grid at a temperature level which can produce malfunctioning. Grids are prone to become emitters at increased temperatures, since they will usually be somewhat contaminated with emitting material that has migrated from the cathode or filament. Actually, there are approximately eight different types of grid current, most of which are greatly influenced by the internal element temperatures. This includes positive grid current and secondary emission grid currents. Any of these effects can produce tube and equipment failures. Pulse circuit applications, especially, must be analyzed thoroughly for grid overheating. The thermal inertia of grids is small and their peak temperatures may easily be excessive during each pulse, even though their average temperature is reasonable.

Screen and supressor grids are temperature sensitive similar to control grids. Due to the appreciable power dissipation of most screen grids, the release of gas due to operation at temperatures exceeding the outgassing temperature is probably their most common weakness. Fortunately, high grid temperatures are usually associated with high plate temperatures and plate temperature measurement will normally provide an indication of general element overheating.

17.3.3.5 Cathodes. Cathode temperature is a function of the heater voltage and the temperatures of the surrounding tube elements, the bulb, and the environment. As a rule cathode temperatures in equipment operating over a narrow temperature range fluctuate mostly due to variations in heater voltage. The present specification limitations of 6.3 volt heaters are usually 5.7 to 7.0 volts (or equivalent) and temperature can vary significantly over a wide range of environmental temperature. Findings to date indicate that a miniature triode, such as a 604, operating from -60 to +250°C ambient would have a change in cathode temperature of from 65 to 75°C. This exceeds the temperature variation resulting from the above limitation of heater voltage. Consequently, even with perfect filament voltage regulation, the temperature of the cathode would vary beyond normal limits. When the effects of the range of environmental temperatures are superimposed on those due to the variation of heater voltage, the resultant performance and life may be unsatisfactory.

Further, if a tube is overheated, the small quantities of gas ions which are usually present, or which may be released, will bombard the cathode and dislodge cathode coating material. Also, since the hot cathode has an affinity for gas, ions arriving at the cathode combine with the free barium to reduce and impair the emission. Thus, excessive tube temperatures can also lead to cathode malfunctioning.

17.3.3.6 Plates. It has been established that most of the heat in a tube ultimately arrives at the plate and that plate temperature is a reliable index of the thermal condition of a given tube. Since the plate in a electron tube is radiation cooled, the temperature of the plate is related to the net temperature of its surroundings. Thus, the life of a given tube is related directly to the temperature of its elements, as mentioned earlier, not necessarily to the temperature of its enclosure (the bulb).

In general, when plates are overheated, gas is released from the plate and the other elements are also overheated by the resulting heat exchange. Consequently, it is always desirable to minimize plate temperature.

17.3.3.7 The effect of tube temperature on life. The primary effect of excessive temperature on tube life is a slow deterioration of characteristics. The life if drastically reduced through decreased emission, evaporation of getter, and emitting materials, increased inter-electrode leakage, gas leakage, glass failure, insulation failures, and grid loading, all of which are temperature sensitive and accelerate rapidly with increasing temperatures.

In the order of thermal importance, the release of gas, electrolysis in the glass, getter migration, grid emission, glass failures, interelectrode leakage, contamination, grid loading, and loss of emission are all influenced by the temperatures of tubes. These temperature sensitive effects can only be alleviated through the proper electrical and thermal operation of tubes.

It can be concluded that electron tubes must be cooled by removing the heat from the envelopes. Only a small portion of the heat can normally be removed through the pins or leads at the base. The cooling of a tube is the most important consideration in its mounting.

17.3.4 Electron tube ratings. In general, the hot spot envelope temperature of receiving type tubes should be less than 175°C. Several organizations currently recommend maximum envelope temperatures in the neighborhood of 100°C for optimum life and reliability. It is definitely known that hot spot bulb temperatures ranging from 200 to 250°C will reduce tube life and cause accelerated deterioration of transconductance. The above values are considered to be only "bench marks," because glass temperature does not necessarily indicate the degree of overheating which may exist in a given instance. Further, electron tube ratings are actually compromises. It is recommended that the plate temperature be monitored in applications wherein the envelope temperature exceeds 175°C. The reduction of element temperatures can improve electron tube reliability more than any other single factor.

When tubes must be operated at high environmental temperatures or under conditions which will result in abnormal plate and envelope temperatures, derating is in order.

Typical envelope temperatures for sea level and at 23°C ambient temperature conditions are presented in Table XXXV as an indication of the temperatures obtained with a single tube in free air. Table XXXVI presents bulb temperatures at various "ambient" temperatures.

Figure 215 presents the relative temperatures of various tube types. Note that maximum ratings, excessive electron tube temperatures can be obtained even in free air. When tubes are used in equipment, these free air ambient conditions seldom exist. It is strongly recommended that the plate and bulb temperatures of each tube used in an equipment under design or development be measured to make certain that safe operating temperatures are achieved.

CAUTION: Extremely effective cooling can reduce the plate and envelope temperatures to levels which are below the maximum rated temperatures. This is an excellent practice but is should not be used to increase the internal element dissipation of tubes beyond their normal rated values. Excessive dissipation is hazardous and overcooling should not be used in order to exceed the maximum rated power level of any tube.

17.3.5 Cooling techniques for electronic tubes. Chapters 8, 9, 10, and 11discuss natural methods, forced air, liquid, and vaporization cooling of electron tubes in detail. The tube manufacturers should always be consulted for thermal data, particularly when special tubes such as transmitting or microwave tubes are to be cooled.

17.4 Magnetic core devices.

17.4.1 Transformers, inductors, and reactors are among the most temperature sensitive electronic parts. This is not commonly realized, but examination of the temperature vs. failure rate curves of MIL-HDBK-217 will reveal that power transformers, for example, have one of the steepest curve slopes of any part. Thus, only a few degrees of excess transformer temperature can result in a very drastic reduction in life.

17.4.2 The details of heat transfer within conventional magnetic core devices are discussed in chapter 8. In brief, the primary thermal failure mode is insulation and conductor failure. Insulation does not fail by immediate breakdown upon arrival at some critical temperature, but by gradual deterioration with time.

Utlimately, a short circuit occurs and subsequent "cremation" results. With class A insulation, for example, experience indicates that the insulation life is halved for each 10 to 12°C increase in temperature throughout the practical operating temperature range. With liquid cooling (oil immersion) the life is halved for each 7 to 10°C increase in temperature.

The heat generated in reactors is produced in the magnetic core and in the conductors. The primary mode of heat transfer within conventional

TABLE XXXV.

TYPICAL HOT SPOT BULB TEMPERATURES IN FREE AIR AT 23°C

Tube Type	Bulb Size	Percent Maximum Plate Dissipation				
		20	40	60	80	100
5814	T-6½	77°C	100°C	118°C	113°C	146°C
6135	T-5½	64°C	82°C	98°C	113°C	125°C
6AH6	T-5½	88°C	103°C	116°C	126°C	132°C
5U4G	ST-16	105°C	116°C	127°C	138°C	149°C
5687	T-6½	123°C	140°C	155°C	155°C	183°C
6L6(metal)		-	-	-	138°C	167°C
6005	T-5½	-	-	175°C	200°C	225°C
805	Special	-	-	-	210°C	230°C
5D22	Special	-	-	-	183°C	200°C
6L6GBY	T-12	67°C	100°C	145°C	175°C	213°C
6L6G		-	-	-	144°C	170°C

TABLE XXXVI.

APPROXIMATE BULB TEMPERATURES AT VARIOUS AMBIENT TEMPERATURES
(General values - not including correction for shapes)

Ambient Temp. (Sea level pressure)	Unit Heat Dissipation in Watts per sq in				
	1.0	2.0	3.0	4.0	5.0
23°C	100°C	170°C	230°C	280°C	310°C
160°C	220°C	260°C	300°C	340°C	370°C
250°C	310°C	350°C	390°C	420°C	450°C

iron core reactors is conduction. Due to the necessity for turn to turn and layer to layer electrical insulation, the thermal resistance between internal hot spots and the surface is large, and high winding temperatures can be obtained. If surface temperature ratins are not available, the internal temperatures of reactors should be determined with imbedded thermocouples.

17.4.3 Limiting insulation temperature. The life at the limiting temperature for any one class of insulation may vary widely according to the quality of the material used, the construction techniques, the mechanical strains imposed on the insulation, and the kind of service to which it is applied. From the results of experience with equipment in service and from laboratory tests on various insulating materials, limiting insulation temperatures (called hottest spot temperatures) have

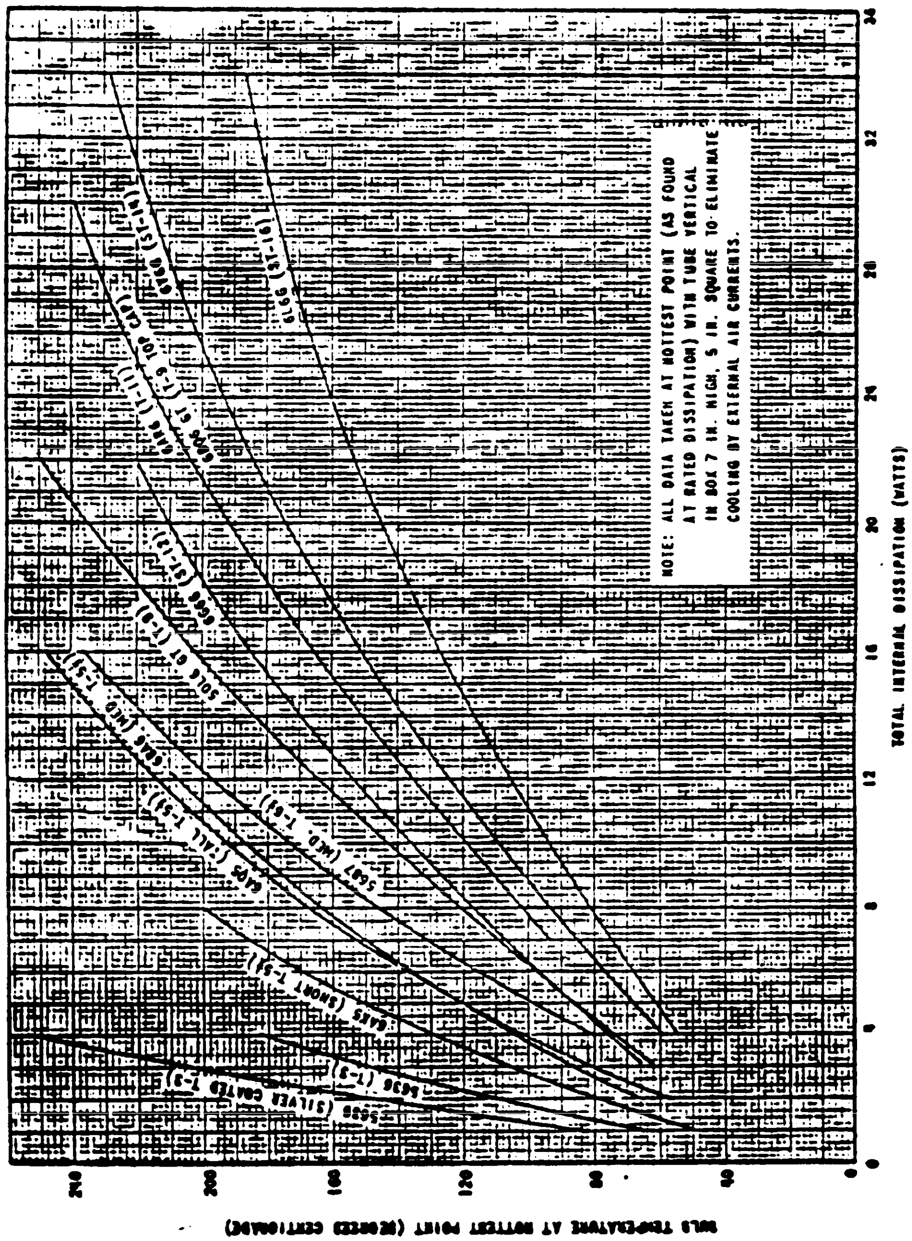


FIGURE 215. Relative Temperatures of Typical Tubes

been assigned by the IEEE. They are of primary importance and useful as a point of reference or "bench mark" in selecting the practical values of observable temperature rise. "Hottest spot" temperature values are not directly applicable for use in rating since the "observable" temperature, that is, the temperature which is directly measurable in practical tests, is less than these peak temperatures by an amount which may be widely different for various types and sizes of reactors. This is due to the inaccessibility of the hottest spot, non-uniformity of cooling, the thermal conductivity, and thickness of the insulation, the form of winding, the rate of heat flow, and the relative locations of the "hottest spot" and the cooled surfaces. Therefore, temperature difference allowances are included in the "hottest spot" (peak) ratings.

17.4.3.1 Class 0 reactors. Class 0 insulation consists of cotton, silk, paper, and similar organic materials when neither impregnated nor immersed in a liquid dielectric. The maximum peak temperature for class 0 insulation is 90°C. The temperature difference allowance between the "hottest spot" and the temperature measuring devices is approximately 5°C and the maximum indicated temperature is therefore limited to 85°C.

17.4.3.2 Class A reactors. Class A insulation, as defined by IEEE, consists of (1) cotton, silk, paper, and similar organic materials when impregnated or immersed in a liquid dielectric; (2) molded and laminated materials with cellulose filler, phenolic resins and similar resins; (3) films and sheets of cellulose acetate and other cellulose derivatives of similar properties; and (4) varnishes (enamels) as applied to conductors.

The usual maximum peak temperature for class A insulation is 105°C. The temperature difference allowance between the "hottest spot" and the temperature sensing element is approximately 5°C and maximum indicated temperature is therefore limited to 100°C.

17.4.3.3 Class B reactors. Class B insulation consists of mica, asbestos, Mylar (or equivalent), fiberglass, and similar inorganic materials in built-up form with organic binding substances. Composite magnet wire insulation consisting of fiberglass (or equivalent) layers covering polyvinyl acetal or polyamide films are included in this class.

The maximum peak temperature for class B insulation is 130°C. The temperature difference allowance between the "hottest spot" and the temperature sensing element is approximately 10°C and the maximum indicated temperature is therefore limited to 120°C.

17.4.3.4 Class H reactors. Class H insulation consists of (1) mica, asbestos, fiberglass (or equivalent) and similar inorganic materials in built-up form with binding substances composed of silicone compounds or materials with equivalent properties; (2) Teflon, silicone compounds or materials with similar properties.

The usual maximum peak temperature for class H insulation is 250 to 275°C. The temperature difference allowance between the "hottest spot" and the temperature difference allowance between the "hottest spot" and the temperature sensing element is of the order of 20°C and the maximum indicated temperature is therefore limited to approximately 230°C. For long life and improved reliability it is recommended that the maximum indicated temperature be reduced to 200°C.

17.4.3.5 Class C reactors. Class C insulation consists entirely of mica, porcelain, glass, quartz, and similar inorganic compounds. No upper temperature limits have been selected for this class of insulation. It is anticipated that the limit will be in the neighborhood of 260°C because of the electrolysis which can occur in glasslike materials at temperatures exceeding this level. (See chapter 18)

17.5 The thermal characteristics of resistors.

17.5.1 General. Almost all resistors have been designed for cooling by natural means. Information related to their ratings when liquid cooled is limited. In general, resistors are provided with lugs for free space mounting and are thus cooled by conduction through the mountings, together with whatever radiation and convection is present. The lead length and temperature of the points of attachment therefore can greatly influence the operating temperature of a resistor in a given location. One of the factors which limits the maximum temperature of many types of resistors is oxidation. If the protective surface material, which is usually an enamel or varnish, is damaged and exposed to the atmosphere, oxidation of the resistance material occurs rapidly, and the resistor is destroyed. Hermetic sealing and interting will tend to overcome this difficulty.

MIL-HDBK-217 discusses the thermal ratings of all commonly used resistors in considerable detail and provides stress analysis data. Forced air, liquid cooling, and vaporization cooling data for resistors is very limited, especially the latter. For these applications, it is recommended that empirical measurements be made and the internal thermal resistance established to make sure that the internal resistance element does not overheat.

17.6 The thermal characteristics of capacitors.

17.6.1 General. Capacitors are not normally considered to be heat sources, with the exception of electrolytic capacitors having high leakage currents and capacitors with relatively high loss factors in radio frequency circuits in transmitters. The surface temperatures of capacitors are usually those of the thermal environment. In general, the leakage resistance of capacitors decreases with temeprature, so that their usable maximum temperature is determined by the permissible circuit losses and their survival temperatures.

Excellent thermal data on capacitors are presented in MIL-HDBK-217. Only data on high temperature dielectrics are presented here.

17.6.2 Glass dielectric capacitors. Glass capacitors are rated for service at 200°C maximum.

17.6.3 Mica Dielectric capacitors. Mica dielectric capacitors are limited to peak temperatures of the order of 120°C when they have plastic cases. Mica is an excellent high temperature dielectric. Mica capacitors with metal cases, or without cases, can be used at elevated temperatures.

17.6.4 Vitreous enamel capacitors. These capacitors are rated for 200°C service.

17.6.5 Barium titanate dielectric capacitors. Capacitors with high K ceramic dielectrics have upper temperature limits of the order of 85°C.

17.6.6 Electrolytic capacitors. High quality conventional electrolytic capacitors are rated to 85°C maximum ambient temperature. Tantalum electrolytic capacitors, dependent upon the type, are rated at 125, 150, 175, and 200°C maximum ambient temperatures.

17.6.7 Variable capacitors. Almost all variable capacitors, with the exception of the barium titantate dielectric type, use dielectric materials capable of service at 200°C.

17.7 Special parts.

17.7.1 Forced air cooled parts. Forced air cooled power transformers are classified for the purpose of this handbook as special parts. Both direct and indirect forced air cooled transformers are available. The directly cooled transformers are uncased with open windings having internal air passages. These transformers usually have an appreciable air pressure drop through the open windings. The thermal efficiency is high compared to conventional transformers as a result of the high Reynolds numbers achieved over the conductors. Indirect forced air cooled transformers are usually internally liquid cooled with external forced air cooled fins. With either type of transformer the manufacturers cooling ratings and requirements must be strictly adhered to.

Forced air cooled high power transistors and SCR's are provided with integral finned heat sinks, similar to external anode forced air cooled transmitting tubes. Again, the device manufacturers have carefully designed the finned heat sink (heat exchangers) for optimum cooling. Thus, the manufacturers thermal ratings and requirements must be met in any thermal design using these devices.

See chapter 9 for detailed forced air cooling techniques.

17.7.2 Liquid cooled parts. Many high powered parts are designed for indirect forced liquid cooling. These include transmitting and microwave tubes, high power semiconductor devices, and other parts such as lasers. Most of these parts include an integral liquid heat exchanger or "jacket." The cooling requirements are peculiar to each individual part and the manufacturers cooling requirements must be met.

Direct liquid cooled parts include transformers and reactors. Usually the liquid system is sealed and integral to the part which is designed for external cooling by some other method.

See chapter 10 for detailed liquid cooling.

17.7.3 Vaporization cooled parts. High powered electron devices such as transmitting tubes, klystrons, and magnetrons are often designed for indirect vaporization cooling. The entire cooling system must be carefully designed to make sure that the device boiler never runs dry. This especially requires that the condensation system must have sufficient capacity to provide an adequate supply of condensate under peak electrical and thermal conditions.

Direct vaporization cooled parts are currently limited to transformers and reactors. Conductor current densities as high as 120,000 amps/sq. in. with heat fluxes as great as 50 watts/sq. in. are being successfully cooled using freon 113 (or equal). (Reference 126) These cooling techniques have permitted the reduction in size and weight of super power airborne 400 HZ transformers to about 0.25 lbs./KVA.

17.7.4 Ferrite devices.

17.7.4.1 Microwave ferrites. Microwave ferrites, especially those used in phased array antenna systems, offer unique thermal problems. These devices typically are bonded in slab form to wave guides or power output devices. Microwave energy is absorbed in the slabs and converted to heat.

In thermal designs associated with ferrites it is often initially necessary to measure the thermal conductivity k of the ferrite material. These materials are usually sintered and the thermal conductivity is a function of sintering pressure and the particular materials used. Because of the range and variety of these materials, the manufacturers have not always determined the thermal conductivity. (This also applies to ferrite memory cores.) The ferrite dissipation can be determined by (a) measuring the losses at low microwave power levels (with uniform temperature) and (b) extrapolating the dissipation to the desired power level. From these data and from the geometry desired, the unit volume heat concentration q/V can be established. K and q/V may be fitted to an exponential function of temperature as:

$$\frac{q/V}{K} = ae^{-bt}$$

where a and b are constants and t = the ferrite temperature.

Substituting in the one-dimensional Fourier conduction equation (see chapter 8) yields:

$$\frac{d^2t}{dx^2} = -ae^{-bt}$$

where x = the distance from the plane of zero heat flow (usually the inner exposed surface of the slab). At the surface bonded to the wave guide, x = L. Subject to the boundary conditions of $t = t_o$ and $\frac{dt}{dx} = 0$ at $x = o$, the solution becomes:

$$\cos^2\left(-x\sqrt{\frac{ab}{2}e^{-bt_0}}\right) = e^{-b(t_0 - t)}$$

It is possible to utilize this equation in several ways. One may assume various values of t_0 and solve for t as a function of x. Interpolation may then be used to determine t at any given x and t_0. Alternatively, if thermally acceptable values of t_0 and t_L are assumed, the required slab thickness x = L may be computed. (For practical solutions, the value of the "angle" must be less than - $\frac{\pi}{2}$, and will be nearer zero.)

An example of a solution, using typical values of the constants a and b, is given in Figure 216. From it, the temperature at any distance, x, may be determined for a given temperature t_0 at x = 0.

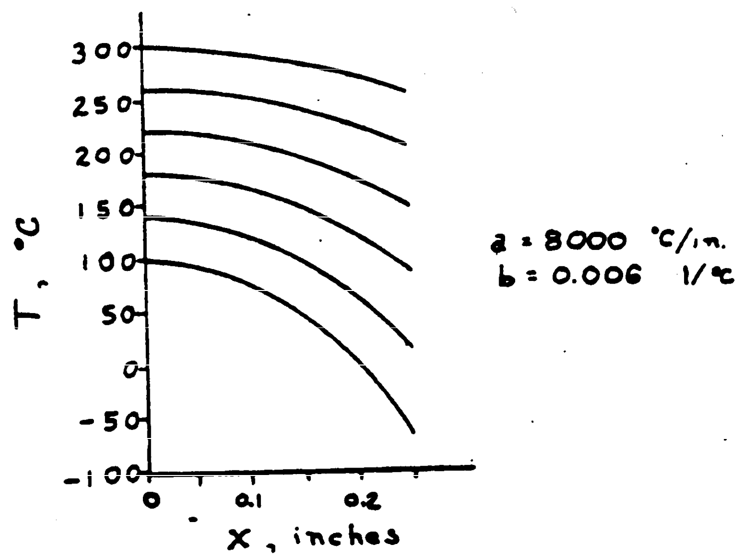


FIGURE 216. Temperature Distribution in Waveguide Ferrite Slabs

This plot may be used to evaluate the total dissipation Q, by approximating the integral

$$Q = A\int_{X=0}^{X=L} q\,dx$$

by the finite difference summation

$$A = A\Delta x \sum_{0}^{n} qn$$

Where:

Q = total ferrite slab dissipation, watts

A = slab cross sectional area

$L = n\Delta x$

17.7.4.2 Ferrite memory cores. Ferrite memory cores used in computers require special thermal consideration. While the dissipation of each core is usually small (75 to 200 milliwatts), such large quantities of cores are used on a core plane that the total heat dissipation can be significant. Core sizes range from 0.125" O.D. x 0.08" I.D. x 0.063" thick to 0.030" O.D. x 0.019" I.D. x 0.005" thick for representative cylindrical cores. The wires threaded through the cores are small and have large currents for their wire size, and have significant heat dissipations. For example, No. 30 copper wires will have half select currents of the order of 0.6 amps and "read" currents of the order of 1.1 amps with duty cycles of 50%. One of the thermal design problems is to determine if the cores are heating the wires or vice versa. Dependent on the configuration and electrical design, the heat flow can be in either direction. Even though only a single core in a 20K core plane, for example, is fully switched at a particular moment, all of the wires and cores along the x and y wires have half select currents with significant heating. With the high access speeds in modern core planes, bursts of heat are produced rapidly at various locations in the core plane as a function of the kind of problem the computer is processing. Thus, heat can be uniformly produced throughout a core plane or concentrated into a few cores. The thermal design must consider the concentrated situation.

Because most core manufacturers do not supply these characteristics, it usually is necessary to measure the core thermal conductivity and heat dissipation. Microcalorimeters can be constructed, checked out, and used to measure the dissipation of a dozen or so cores which are operating normally, i.e., connected to core driver computer circuits, etc. The thermal conductivity of the cores must be measured using the actual cores. Because the cores are sintered the densities (and thermal conductivities) vary among the different sizes. Thus, the thermal conductivity will vary from core size to core size and the thermal conductivity is not subject to scaling.

The heat distributions in free air associated with a core and its wires can be mapped by normally operating a single core with its complement of wires in a transparent plastic case (to avoid stray air current effects) and probing the core and wires with a null type thermocouple attached to a micromanipulator. A null type thermocouple can be made by winding a small non inductive resistance heater wire around the tip of the thermocouple probe and powering the heater winding with D.C. from an adjustable low voltage power supply. The current in the heater is adjusted until the thermocouple, when placed in contact with the core or wires, does not produce a deflection in the galvanometer of the temperature measuring bridge connected to the thermocouple. This means that the thermocouple at the end of the probe is at the same temperature as that of the location of interest on the core or wires and that no heat is exchanged between the core or wires and the thermocouple. Thus, the temperature distribution on the core and the wires can be mapped using a null technique that does not perturb the temperature distribution. (This technique is also excellent for mapping the temperature distribution of uncased integrated circuits or chips.)

Next, the maximum and minimum operating temperatures of the cores must be determined. These temperatures are usually established by the electrical and magnetic requirements, rather than the reliability requirements. The copper wire and ferrite cores can be exposed to high temperatures with no ill effect on life, but the electrical tolerances in computers converge in the core memories and the controlling temperatures are those at which the cores switch unsatisfactorily. The temperature limits can be established by operating a few cores in a temperature controlled oven while operating the cores normally. If the cores are permitted to stabilize passively at the oven temperature and then addressed with a single read-write pulse so that the temperature rise is only a fraction of a degree, examination of the electrical waveshapes will determine if that temperature is satisfactory. Varying the oven temperature will establish the range of satisfactory operation. In several typical instances, the cores could only be operated between 75 and 90°F, either active or passive. This limited the minimum coolant temperature to 75°F and the allowable core temperature rise to 15°F. Also, situations have been encountered wherein the cores were so temperature sensitive that their temperatures could more accurately be determined by their electrical performance than by conventional temperature measuring techniques. The point is that the thermal designs for cooling memory cores must be accurate and exact.

Forced air, liquid, and vaporization cooling can be used to cool memory core planes. Forced air is the more desirable technique since maintenance is more readily accomplished. With forced air cooling, it is recommended that the air flow be parallel to and across the core planes. Each surface of a core plane should be fully exposed to a sheet of cool air moving over the surface of the cores. Typical values of h_c are 0.1 watts/(in.^{2}-°C) with air velocities of 75 ft./min. Figure 217 presents the thermal analog for a single 0.03 x 0.019 x 0.005 core.

In Figure 217:

R_w = the thermal resistance of the wire to the air (without cores), °C/watt

R_c = thermal resistance of core to the air, °C/watt

R_{cw} = the thermal resistance of the core to the wire, °C/watt

t_c = core temperature rise, °C

t_w = wire temperature rise, °C

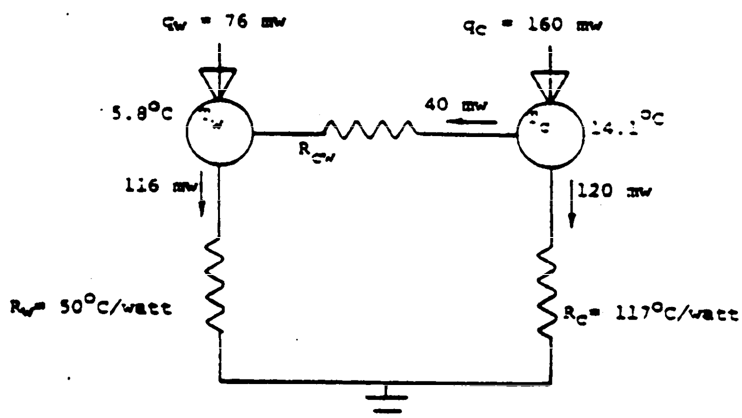
q_c = core dissipation, watts

q_w = wire dissipation, watts

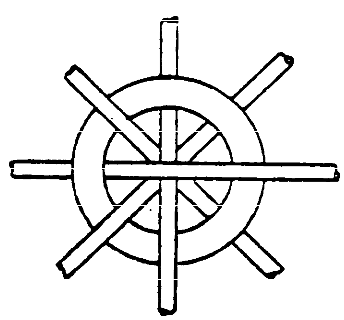
The 116 milliwatts is the total heat rejected to the air by the wire (76 + 40 mw) and 120 milliwatts is rejected by the core directly to the air (160-40 mw).

Liquid cooling is an effective method of core cooling. However, the maintenance problems can be difficult. It is important to note that the core switching can be seriously impaired by a viscous coolant or by plastic encapsulation. The cores change dimensions a few microinches when switched from one magnetic state to another due to magnetostriction. If this movement is restrained or damped the electrical characteristics become unsatisfactory.

Vaporization cooling has been used in memories with very high unit heat concentrations with extreme temperature sensitivity. In one instance, the unit heat concentration was 50 watts/cu. in. and the core temperature had to be maintained between 70 and 80°F either active or passive. This meant that only a maximum of 10°F temperature rise was permissible at 50 watts/cu. in. Freon 11 (or equivalent) with a boiling point of 74.7°F at one atmosphere pressure was successfully used.



Thermal Circuit



Enlarged View

FIGURE 217. Ferrite Core and Wires

18. DESIGN OF ELECTRONIC EQUIPMENT FOR OPERATION AT 150 to 350°C ENVIRONMENTAL TEMPERATURE

18.1 General. High temperature electronic equipment was required by the DOD and special high temperature parts were developed during the mid and late 1950's. These parts were also to be highly resistant to nuclear environments. The requirements for high temperature equipments have eased in recent years. However, special situations may arise wherein high temperature thermal design techniques will be of interest. It is important to note that in this temperature regime the active devices are electron tubes (high temperature semiconductors are almost nonexistant) and that various high temperature equipments have been successfully designed, built, and operated.

18.2 Theoretical considerations.

18.2.1 General. Data on thermal properties of most materials used in electronic parts at temperatures above 300°C are relatively meager. Little information has been published on heat transfer in equipments in the thermal environment of interest. It is however, possible to derive some general conclusions from the available information.

Materials commonly used in conventional equipment alter their physical properties so that they either become unusable or perform quite differently at high temperature. Thus, the materials and configurations of high temperature, and especially ultra high temperature, electronic equipment are quite different from conventional types. The present state-of-the-art is such that there are no well established forms and arrangements. The relative effectiveness of the different modes of heat transfer in electronic equipment shifts radically with increasing temperature as described in the following paragraphs.

18.2.2 Conduction.

18.2.2.1 Thermal conductivity. In general, the materials suitable for use in high temperature electronic equipment have higher heat conductivities than those materials commonly used for low temperature work. Figure 218 is based on rather meager data from a variety of sources. Data on ceramics vary considerably due to differences in composition and curing.

There is little information on the temperature coefficient of thermal conductivity, except for gases and some metals. Solids, generally, have rather low temperature coefficients. For most metals, the coefficient is negative, but aluminum shows an increase in conductivity with a temperature of 0.064% per °C. Crystalline materials generally have a negative coefficient, and amorphous materials (glasses) a positive coefficient. Figure 218 includes a few low temperature materials for comparison. There seems to be roughly an order of magnitude difference in thermal conductivity between low and high temperature electric insulating materials.

Due to the relatively high thermal conductivity of high temperature insulating materials, the thermal resistance could be significantly lower in high temperature equipments than in their present conventional counterparts.

The following examples illustrates some aspects of high temperature conductive heat transfer.

Example 1: Electrical conductors of equal electrical resistance.

100°C	350°C
Copper wire (#16 gauge)	Aluminum wire (#6 gauge)
$A = 0.002$ in.^{2}	$A = 0.0206$ in.^{2}
$L = 1.0$ inch	$L = 1.0$ inch
$R_{elec} = 44.0 \times 10^{-6}$ ohm	$R_{elec} = 44.0 \times 10^{-6}$ ohm
$R_{th} = 52.7$ C/watt	$R_{th} = 7.05$ C/watt

Example 2: Electrical insulation. $A = 1.0$ in.^{2} $L = 0.050$ in.

50 to 100°C	100 to 250°C	250 to 500°C
Varnished Cambric	Teflon	Alumina Ceramic Enamel
11.6°C/watt	7.81°C/watt	0.238°C/watt

Note that the thermal conductivity of the ceramic is about 50 times better than that of conventional insulation.

Example 3: Air, $A = 1.0$ in.^{2} $L = 0.125$ in.

100°C	250°C	500°C
156.°C/watt	117.°C/watt	34.7°C/watt

Note that the thermal conductivity of air increases significantly with temperature.

18.2.2.2 Thermal contact resistance in joints. Conductive heat paths almost always contain series resistances in the form of surface contacts. Most such surfaces are metal to metal, but in high temperature electronic equipments it is also quite probable that ceramic to metal contacts may be encountered.

Certain general conclusions have been reached.

a. Conductivity increases with pressure, at a decreasing rate, and this effect increases with decreasing hardness of the material.

b. Conductivity increases with the mean interface temperature. This increase is linear for aluminum and stainless steel up to 200°C.

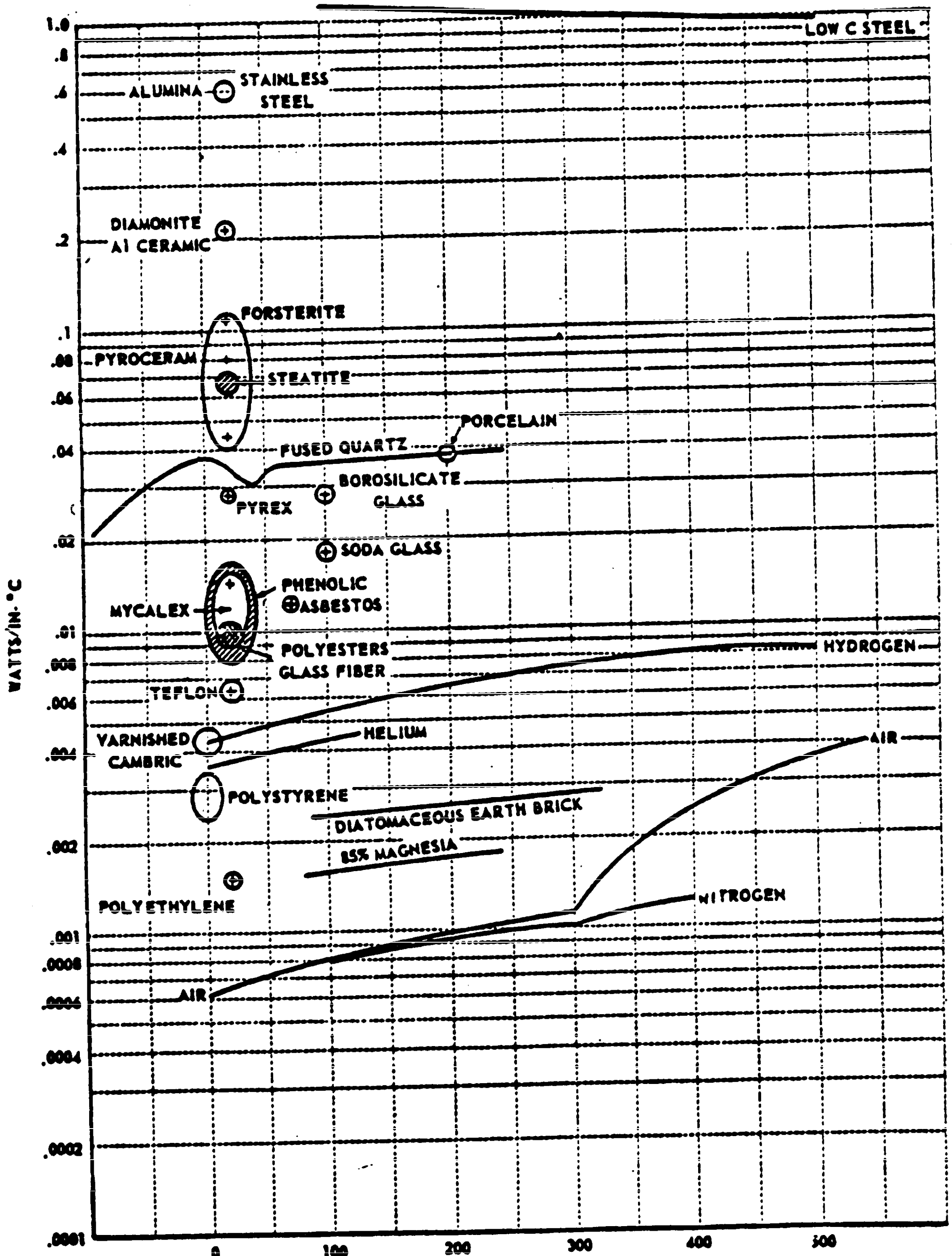


FIGURE 218. Thermal Conductivities of Materials

c. Conductivity increases as surface roughness decreases.

d. Surface flatness is more important than roughness. Warpage due to uneven heat flow decreases conductivity.

e. Conductivity tends to increase slowly during long heating periods, but returns to the original value when cooled.

f. When the surfaces are separated and then reassembled, the conductivity values are reproducible within the error of measurements.

g. Shims of soft material between hard surfaces increase the joint conductivity if the shim material is a good heat conductor. Sheet insulating materials such as asbestos greatly decrease the conductivity of metal to metal joints.

h. A mathematical model of the contact is suggested and illustrated in Figure 219. The contact materials are represented by a network of resistances which are linear with temperature, and these two networks are connected by parallel nonlinear resistances representing radiation and convection through the intervening air film.

The heat flow across the contact can be represented by an equation of the form

$$q = C \left(T_1^{\,n} - T_2^{\,n}\right)$$

where:

q = heat flow rate
C = a constant
T_1, T_2 = temperature of the metal surfaces

The exponent n depends on the material and surface condition and ranges from 1.6 to 3.

Figure 220 indicates the conductivity values attainable with aluminum. It is easy to design metal to metal joints of satisfactorily good thermal conductivity, but it is important to so design them that heat flow is reasonably uniform to prevent transverse temperature gradients and the consequent warpage.

18.2.3 Convection.

18.2.3.1 General. Convective heat transfer involves the mass transport of material, which absorbs heat at a hot surface and then either gives up this heat at a cooler surface or is removed to some thermally remote location, taking the heat with it. Convection thus involves the dynamics of fluids. Table XXXVII shows the general effect of increased operating temperature on free convection for several coolants.

TABLE XXXVII. Natural Convection Coefficient vs. Temperature

	Laminar Flow	Turbulent Flow
Air	$\frac{h(10°C)}{h(371°C)} = 1.26$	$\frac{h(10°C)}{h(371°C)} = 1.72$
Water	$\frac{h(10°C)}{h(300°C)} = 0.253$	$\frac{h(10°C)}{h(300°C)} = 0.156$
Light Oil	$\frac{h(10°C)}{h(300°C)} = 0.31$	$\frac{h(10°C)}{h(300°C)} = 0.207$

Increasing temperature decreases the heat transfer rate with air, but increases it very markedly with liquids. Liquid water and oil would develop high pressures and are hardly suitable for use in electronic equipment at 300°C but their behavior is typical of liquid as opposed to gaseous coolants.

As an illustration of the change in free convective heat transfer, the unit heat dissipation from the side wall of a vertical cylinder one inch high was computed for different free stream temperatures and temperature differences with standard air as the coolant. These results are plotted in Figure 221. The unit heat dissipation will decrease slowly as the height of the cylinder increases. The decrease in convective heat loss with increasing ambient temperature is evident.

Lighter gases are more effective for convective cooling, due chiefly to their higher specific heats. From the free convection equation for streamline flow at 0°C, the coefficient of convection for air and helium show:

$$\frac{h(He)}{h(Air)} = \frac{0.00746}{0.00377} = 1.98$$

For turbulent flow:

$$\frac{h(He)}{h(Air)} = 1.4$$

It is suggested that the use of helium in hermetically sealed equipment be explored. Hydrogen would probably be more effective thermally, but undesirable because of its explosive hazard. Both of these gases diffuse through glass, and some ceramics.

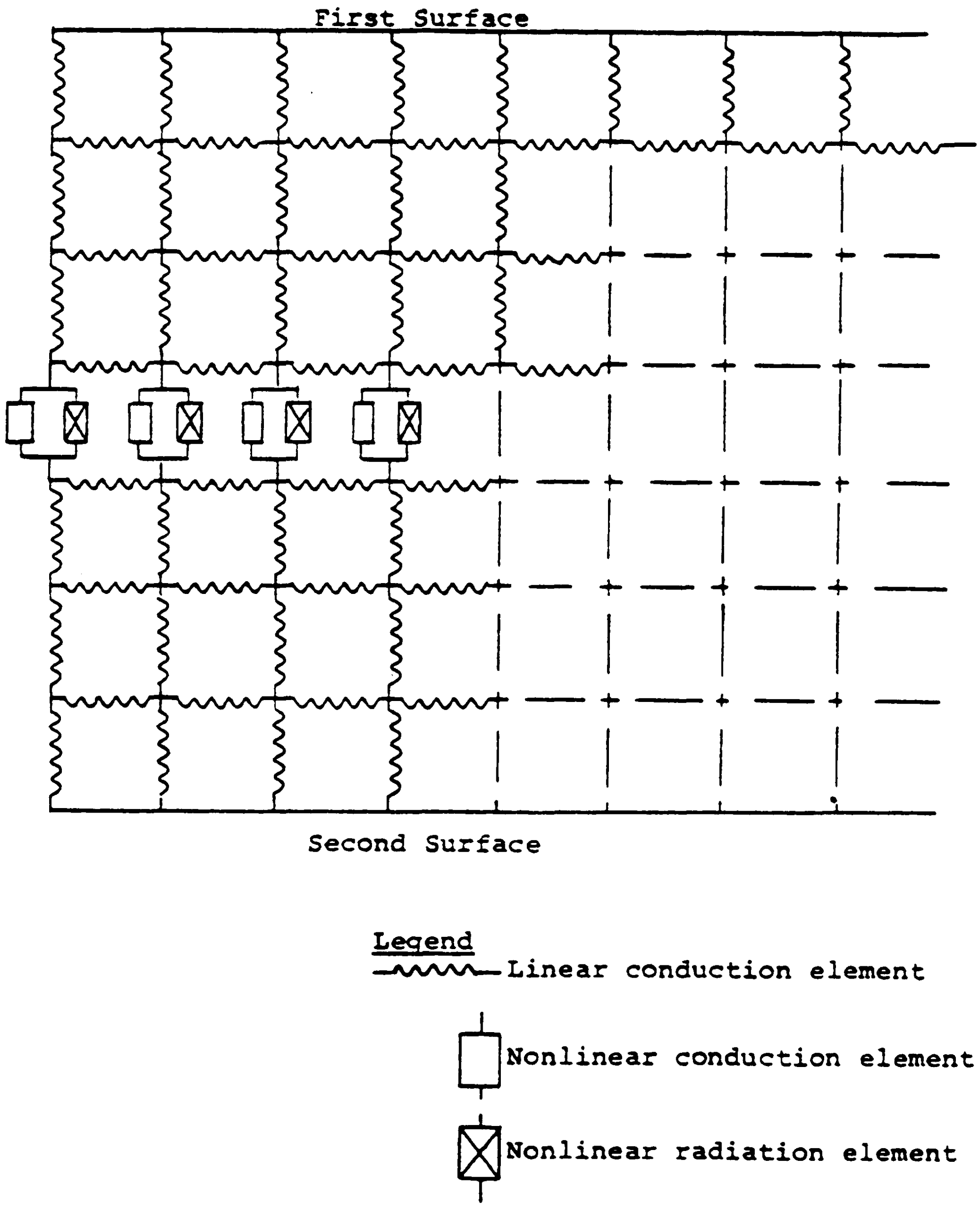


FIGURE 219. Equivalent Circuit for Contact Resistance

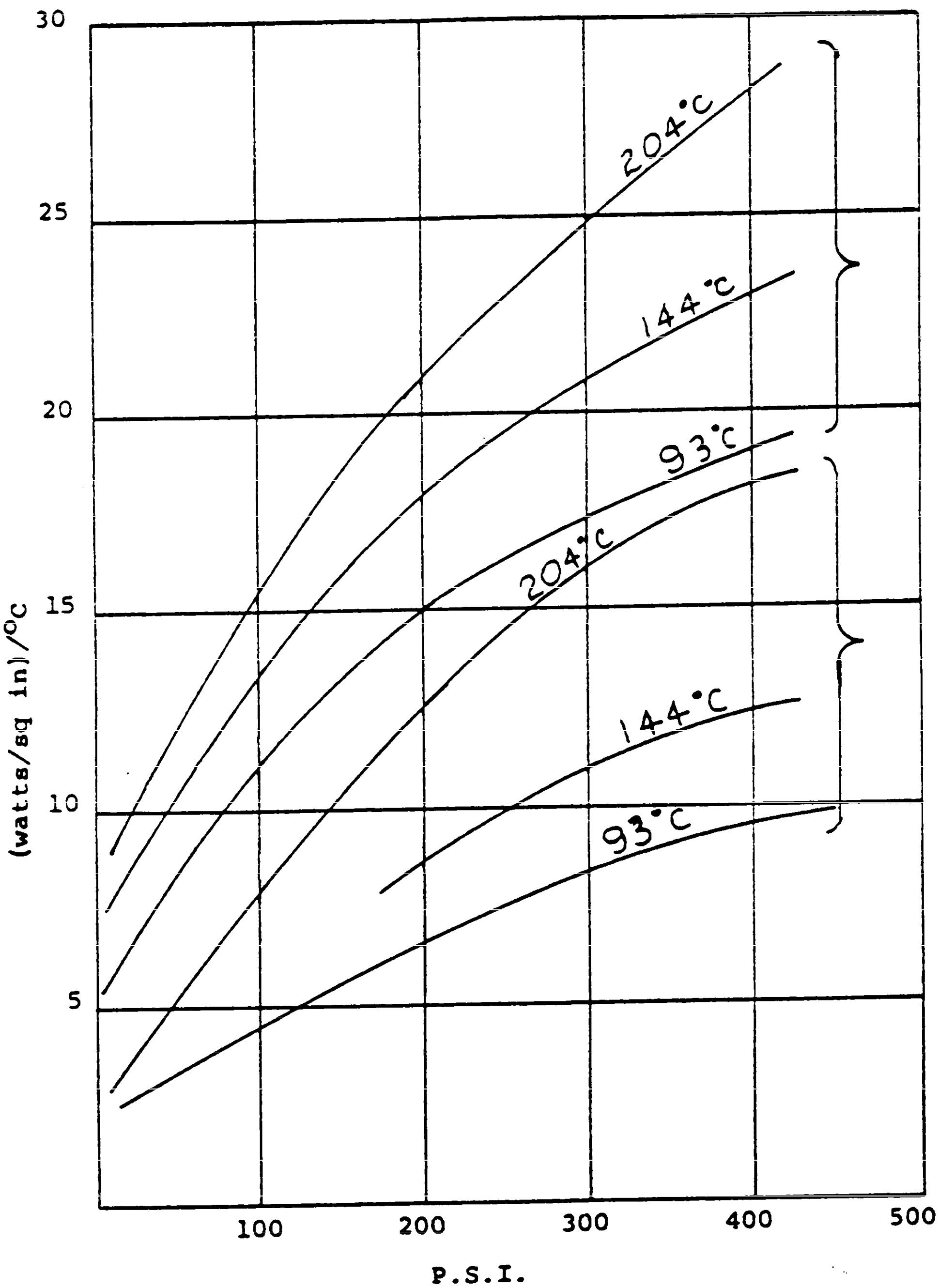


FIGURE 220. Contact Conductance vs. Temperature and Pressure (For Aluminum)

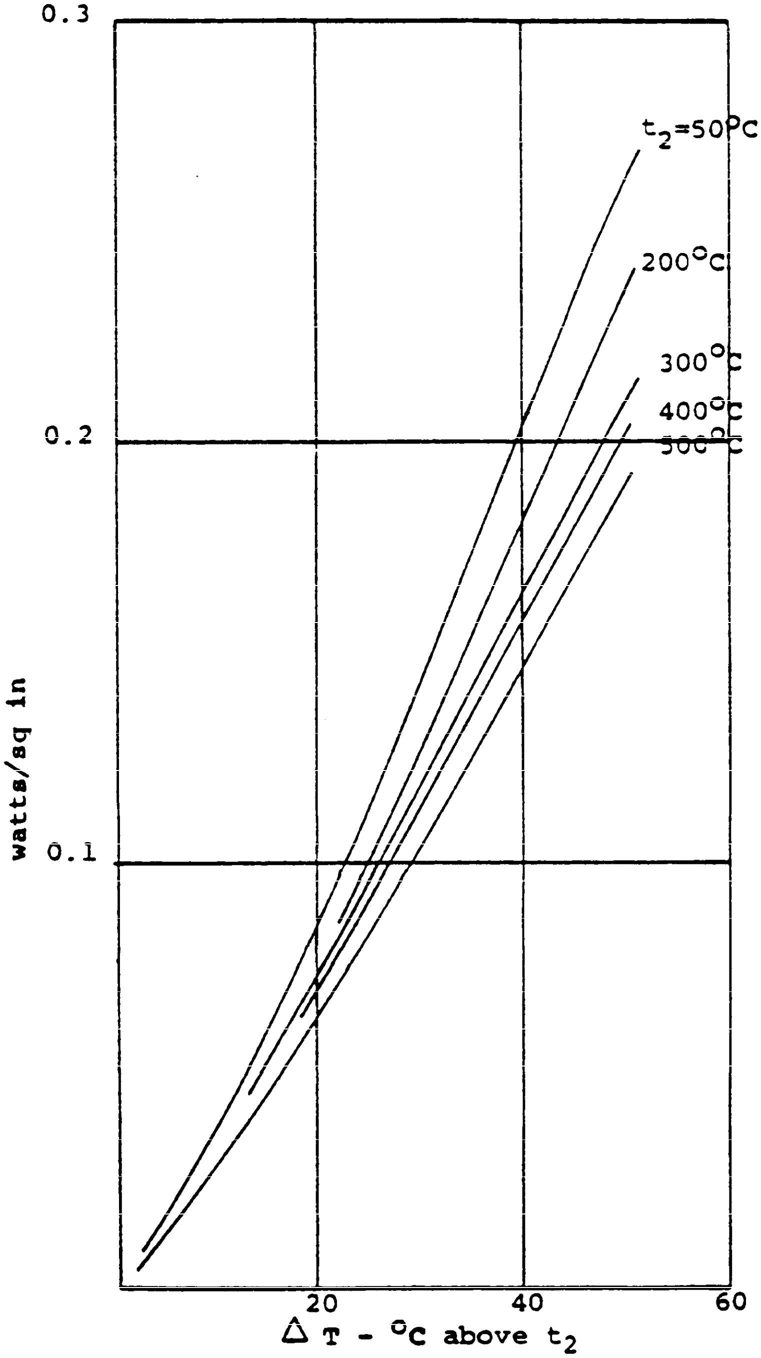


FIGURE 221. Free Convection Heat Loss vs. Temperature

18.2.3.2 Forced convection. The general equation for forced convection is:

$$h_c = k\ f\ (Pr,\ Re)$$

where Re = Reynolds number.

The functional relation is such that h increases with Pr and Re but at less than a linear rate. When the coolant is a gas, the specific heat increases slowly and the density decreases rapidly, so as to keep a constant mass flow rate. This increase in velocity offsets the increase in viscosity and the net effect is an increase in heat transfer rate with temperature.

For air, $\frac{h(10°C)}{h(371°C)} = 0.876$

When the coolant is a liquid, the Reynolds number for constant mass flow rate increases with temperature because the viscosity decreases. The Prandtl number decreases, however. The net effect depends chiefly on the variation of thermal conductivity with temperature. Three examples are given for liquid coolants flowing through tubes.

For water $\frac{h(10°C)}{h(300°C)} = 0.3$

For mercury $\frac{h(100°F)}{h(600°F)} = 1.03$

For Na-K eutectic $\frac{h(200°F)}{h(1300°F)} = 1.22$

It is interesting to compare helium and air for forced convection. At 0°C, assuming the same velocity for both,

$$\frac{h(He)}{h(Air)} = 2.3$$

Assuming the same mass flow rate, the comparison is even more favorable.

$$\frac{h(He)}{h(Air)} = 5.34$$

The specific heat of helium is roughly one-half that of hydrogen and five times that of air. Therefore, the required flow rates of these light gases might differ enough to permit a reduction in turbulence with a saving in blower power.

Hydrogen has been used for some years in cooling electric machinery and ample data are available. Thermal data on helium are scarce.

18.2.4 Radiation.

18.2.4.1 General. Since radiation increases as the difference in the fourth power of the absolute temperatures, radiation is an important mode in high temperature electronic equipment.

18.2.4.2 Radiation characteristics of materials. Figure 222, compiled from various references, shows the effect of temperature on normal total emissivity for several materials of interest in high temperature electronic equipment. Emissivity is very much dependent on surface nature and condition so that a good deal of caution should be used in calculating radiant heat transfer. Data from different references do not always agree.

Glasses and ceramics, including oxidized metals, show a considerable decrease in emissivity as temperature increases, which partially neutralizes the fourth power law advantage. Bright metallic surfaces show a slight increase in emissivity with temperature. Crystalline materials, including metals, act like gray bodies and do not exhibit selective absorption with wave length. Liquids, including glasses, and gases generally, do exhibit selective absorption. Gases with symmetrical molecules such as hydrogen (H_2), oxygen (O_2), nitrogen (N_2), and helium (He) are transparent to thermal radiation. Gases with unsymmetrical molecules like ammonia (NH_3), and carbon dioxide (CO_2) exhibit selective absorption over certain frequency bands. Metallic oxides also exhibit selective absorption.

Since it is manifestly impossible to analyze radiant heat transfer problems over a spectral range, averages are taken by means of the measured total normal emissivities.

The absorption characteristics of gases indicates an attractive possibility for cooling. McAdams (Reference 26) lists the following potentially useful gases:

Water vapor
Carbon dioxide
Sulfur dioxide (corrosive)
Ammonia
Hydrogen chloride (corrosive)

18.2.4.3. Thermal radiation in high temperature electronic equipment. An electronic equipment in general consists of a system of numerous small heat sources and small nonheat producing parts quite closely packed in an enclosure. Some of the parts are mounted on substantial metal surfaces and are connected together by metallic paths. There is thus, the possibility, at least, of considerable heat transfer by conduction. All of the heat removed from the parts by radiation must be absorbed by some body. The absorbing body must then transfer this heat to some ultimate sink and much of this transfer will be by conduction, so that radiative and conductive paths will generally occur in series. There will also be parallel combinations, of course.

If radiant energy is allowed to "bounce around" inside an enclosure, black body conditions will develop, that is, all surfaces inside the enclosure will tend to approach the same temperature. There is no profit in merely circulating the radiant energy among the sources. Since radiative heat transfer becomes potentially more useful as temperatures increase, it is important to provide radiative heat paths as direct as possible from sources to sink. Evidently is is desirable to surround each source completely with sink surfaces.

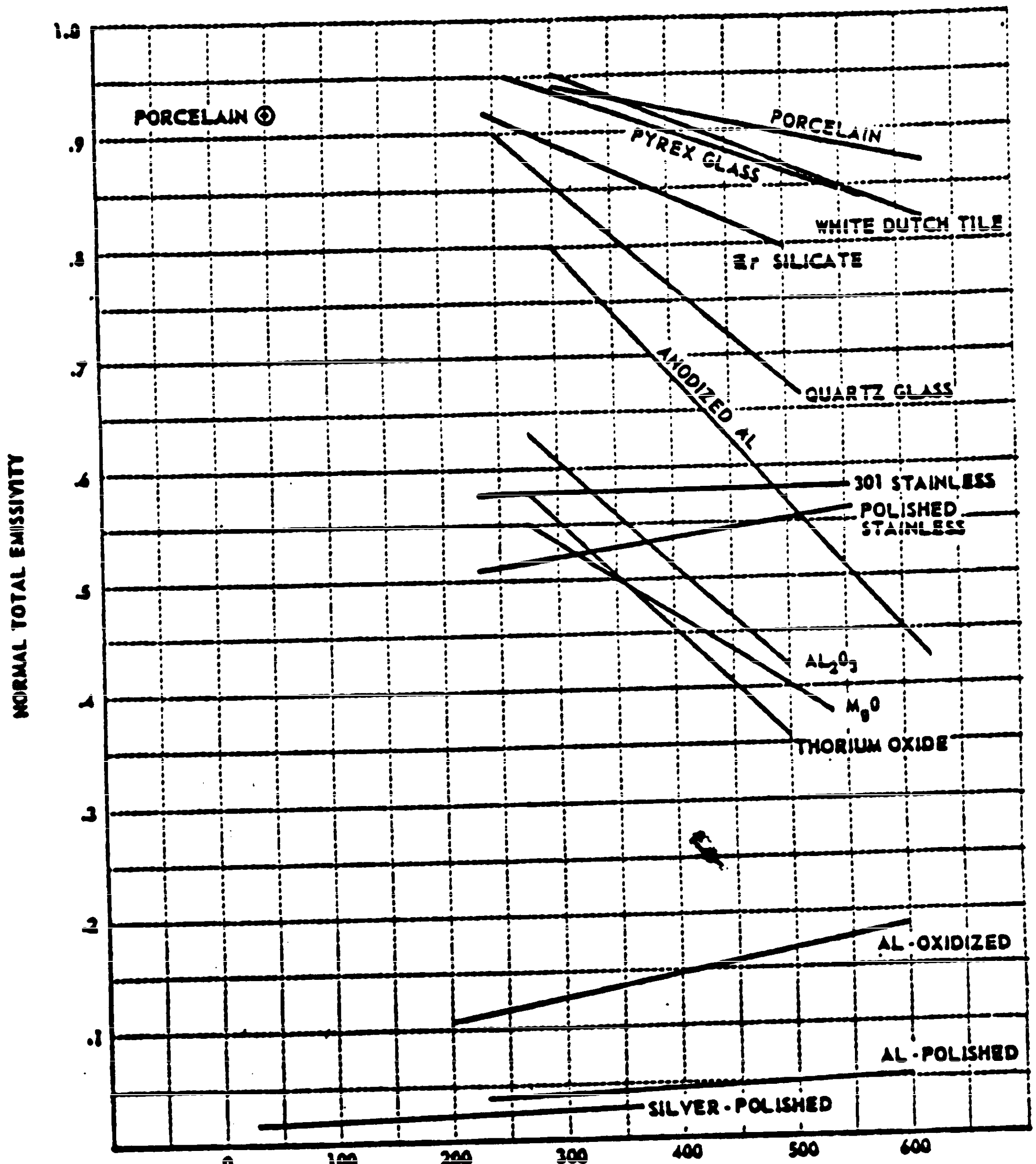


FIGURE 222. Emissivities of Materials

18.2.4.4 Examples of radiant heat transfer. Figure 223 shows radiant heat transfer (unit heat dissipation) versus temperature difference for different surface temperatures, assuming black body conditions. This plot is comparable to Figure 221 for free convective cooling.

Since the materials used in high temperature electronic parts have emissivities which decrease with increasing temperature, the results shown in Figure 223 are optimistic. If the sink surface is totally absorbent all points on these curves will drop by a factor equal to the emissivity of the source.

The relative effectiveness of radiation and free convection is shown by the following example. A heat source in the form of a three-inch cube is suspended in air. The surface is considered to have 100% emissivity. Heat loss is calculated at several ambient temperatures for a temperature difference of 100°C. Table XXXVIII shows the results.

TABLE XXXVIII.

Surface Temp.	Air Temp.	Total heat dissipation	%by free convection	%by radiation
140°C	40°C	66. watts	41.0	59.0
200	100	88.	30.1	69.9
300	200	143.	18.8	81.2
400	300	220.	10.9	89.1

18.3 High temperature Materials.

18.3.1 Liquid coolants. Insulating liquids are very useful for direct liquid cooling by spray or immersion. Two types useful at 200°C and over are silicones and fluorocarbons. When kept free from contact with oxygen, these fluids will operate for 500 hours at 250°C. Their dielectric strength is high and their dielectric constant sufficiently low to avoid serious parasitic capacitances. These characteristics are maintained at high temperature.
Submerged arcing in silicone liberates hydrogen and methane which are readily reabsorbed in small quantities. More severe arcing forms a gelatinous material which occludes carbon particles and lowers the dielectric strength. Moisture and oxygen are harmful, as with all liquid dielectrics, and these fluids must be protected from atmospheric contact. The sealed containers must allow for expansion as temperature rises, and "breathing" during temperature cycling should be prevented.

The manufacturers of these fluids have published extensive thermal and electrical data, a few which are given here.

Monsanto OS-45 (or equal) performs well as a coolant from -60 to 200°C.

Monsanto Aroclor 1221 (or equal) boils at 275°C and has a flash point of 180°C. It is a reasonably good coolant.

Dow-Corning 200 (or equal) freezes at -76°C, boils at 192°C, and has a flash point of 160°C. It compares with sodium as a coolant.

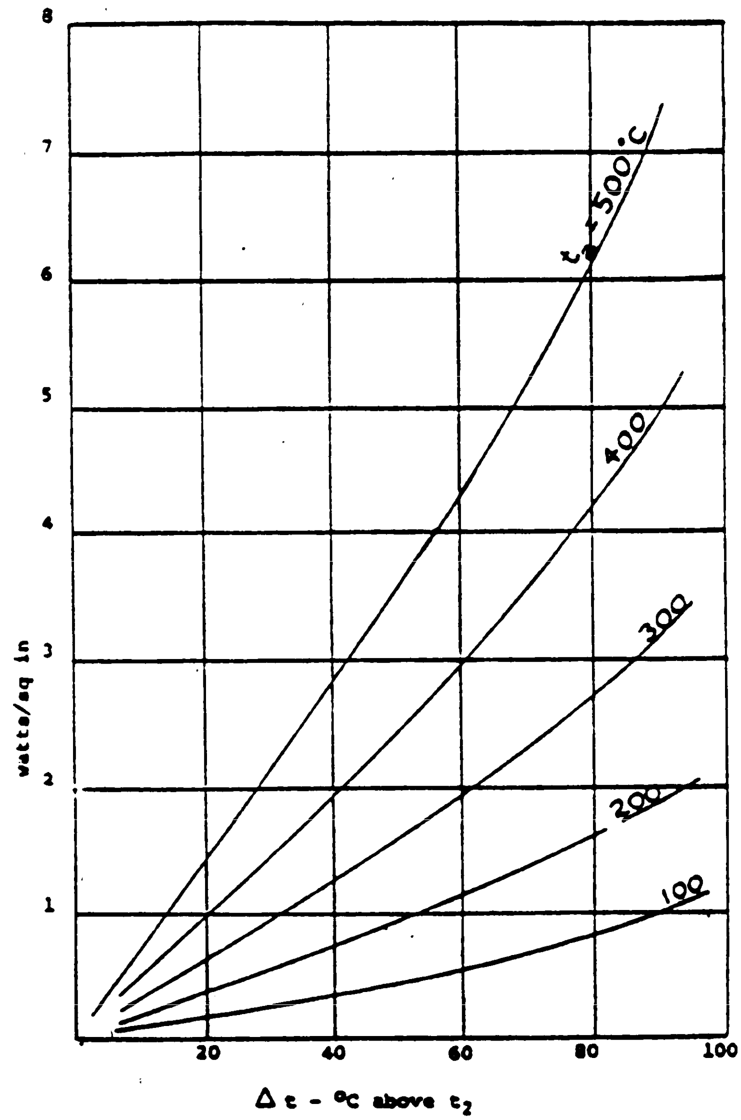


FIGURE 223. Radiation Heat Loss

18.3.2 Evaporative coolants. Dichlorobenzene (Dowtherm) (or equal) is a dielectric and satisfactory refrigerant, recommended by the manufacturer for use up to 260°C. It boils at 180°C at atmospheric pressure and requires 80 psia at 260°C. Evaporation at 150°C and condensation at 205°C yields 84% of the Carnot cycle. It appears to be a suitable direct spray evaporative coolant up to at least 260°C, without requiring excessive pressures. It should not be used in contact with aluminum. It is toxic.

18.3.3 Magnetic materials. Ferromagnetic materials are essential for obtaining high flux densities in reasonably small space. The pertinent characteristics are permeability, loss factors, and durability. The durability of all ferromagnetics is satisfactory up to at least 500°C and does not present a problem.

Permeability varies with temperature and drops very sharply at a certain point known as Curie temperature. The Curie point thus in effect determines the maximum operating temperature. Table XXXIX gives the Curie temperatures for common magnetic materials.

TABLE XXXIX. Curie Temperatures

Material	Curie Temperature
Iron	770°C
Nickel	357
Cobalt	1120
Silicone irons	756 to 640
Nickel irons	760 to 560
Permalloy	570 to 460
Mumetal	450
Supermalloy	380
Perminvar	715 to 537
High carbon steel	770
Tungsten steel	760
Chromium steel	745
Cobalt steel	804 to 890
Alnico II	800
Alnico III	725

The saturation flux density of all magnetic materials decreases with increasing temperature. For iron, it decreases from 100% at 20°C to 85% at 500°C. Cobalt shows less change. Alloys constituted of two and more phases show erratic variations. Aging at temperature usually decreases permeability and increases hysteresis loss.

Hysteresis loss decreases with increasing temperature because of decreased crystalline forces. Eddy current loss decreases because of increased resistivity. The core loss picture at high temperature is thus favorable and the Curie point is the important characteristic.

Table XL lists the maximum operating temperatures for common magnetic materials.

TABLE XL. Magnetic Material Operating Temperatures

Material	Maximum Operating Temperature
Ferrites	250°C
Silicone steel	500°C
Cobalt steel	above 500°C
Nickel steel	below 500°C

Cobalt is undesirable in a radiation environment because of the long half-life of the isotope Co. 60.

For permanent magnets Alnico 5 and Alnico 6 (or equal) are usable to 500°C, and with 20% energy loss to 700°C.

Silicone steel shows a decrease of 10% in core loss and 28% in permeability between 30°C and 224°C, with no permanent change in properties during long soaking at high temperature.

18.3.4 Dielectric materials. All insulating material is dielectric. The term "dielectric material" as used herein means material of high dielectric constant for use in capacitors. Electrical insulation means material of high breakdown strength used to prevent electric contact. A high dielectric constant is frequently undesirable in insulation since it causes high parasitic capacitance.

In general, both dielectric constant and conductivity increase with temperature. The RC product is the best single index of capacitor quality. A good capacitor must have an RC product greater than 5 or 10 megohm-microfarads, depending on the application. Most dielectrics give RC products of less than one at temperatures around 500°C. Ferroelectric or "high K" materials generally show large and variable changes in dielectric constant with temperature, with corresponding changes in capacitance. The dielectric constant always shows a large decrease at some high temperature.

Natural mica swells and exfoliates when heated but synthetic mica does not. Molded mica capacitors of standard type were found to be operable up to 250°C, at least for "one shot" or short time operation. Mixed tantalates titanates, and niobates can be compounded which will make satisfactory capacitors for 250°C operation. Oxides of zirconium, magnesium, and aluminum are all usable at 500°C, and extreme purity is not necessary. Special samples of high temperature mica capacitors have been operated at well over 500°C for extended times with little or no change in capacitance and leakage. They served as components in an audio amplifier at constant frequency, and temporary changes with temperature were not measured.

18.3.5 Electrical insulation. Silicone rubber and silicone insulating oils are satisfactory up to 250°C. Silicone impregnated glass fiber has been used successfully in transformers operating at 250°C winding temperature. Insulation tests were made at 300°C and 2,000 hour life was achieved at 250°C. Silicone impregnated asbestos has been found satisfactory for a 250°C winding temperature.

Teflon (or equal) is satisfactory up to 250°C. Teflon releases poisonous fluorine gas at around 270°C, softens, and cuts easily.

Silicone bonded, integrated phlogopite mica sheet coil insulation and phosphate bonded, recombined muscovite mica for spools has been found satisfactory for 500°C ambient operation with an estimated hot spot temperature of 600°C.

Various ceramic wire insulations are suitable for 500°C but tend to brittleness. The best practice is to apply the wet ceramic to the wire during winding or bending and cure it when in its final form. One supplier now offers wire with an uncured coating which permits winding and bending. Curing is effected after the wire is in place. Wire insulated with high temperature glass has been wound and formed while hot enough to soften the glass.

Stripping the insulation from high temperature wire is an annoying problem. Anodized aluminum is stripped by dipping in caustic. Cured ceramic coatings would require more vigorous treatment. The uncured coatings offer no difficulty.

18.3.6 Electrical conductors. All metals suitable for electrical wire increase in resistance, decrease in strength, and expand with increasing temperature. Copper shows a permanent increase in resistance but silver and aluminum return to the original value when cooled. These physical changes must be considered in the design of high temperature equipment.

The best commercial conductors are silver, copper, and aluminum. Copper oxidizes readily and the oxide spalls off so rapidly that it cannot be used in the presence of oxygen above 300°C. Nickel plate is a satisfactory protection and nickel-clad copper wire is satisfactory up to 350°C. Silver plate is unsatisfactory because it diffuses into the copper and causes a progressive increase in resistance. Silver and aluminum are useful up to 500°C since their oxide layers are very adherent. It has been found that coin silver (10% copper) develops a heavy black oxide coating above 500°C but continues to function satisfactorily. Pure silver shows much less oxidation. Aluminum forms a very thin, tough, adherent oxide coat even at room temperature. A heavier layer and surface passivity is developed by anodizing, and anodized aluminum has been used in experimental 500°C transformers. The space factor is high because of the very thin insulating layer.

Several companies offer nickel-clad copper and anodized aluminum wire in commercial sizes. Silver wire is also a standard commercial item.

In some cases, stainless steel may be used for conductors because of its strength and durability. Titanium, Kovar, and nickel alloys such as Innonel may be found as terminals for parts.

The ease of joining metals is an important factor. Silver solder can be used with silver and copper alloys, but may result in a weak joint with steel. Spot welding is possible but tricky with high conductivity metals, very easy and satisfactory with steel and nickel. Aluminum can be spot welded if the surface oxide is controlled. Titanium is very difficult to weld and braze, and should not be used for wiring equipment. In general, all joining methods are much more difficult and expensive in high temperature work than the conventional soft solder on copper technique. Mechanical joining means such as clamping, crimping, cold welding, and ultrasonic welding are therefore, attractive possibilities.

18.3.7 Semiconductors. Germanium is useless above 100°C. Silicone can be used to 200°C. De-rated silicone rectifiers have been operated at 250°C. Several materials are being investigated for high temperature use, but 250°C seems to be the current ceiling. Thermistors are available for 400°C operation. Silicone carbide rectifiers have been operated at 760°C. Tellurium rectifiers have been operated experimentally at 400°C.

18.3.8 Ferrites. Ferrites in current use have a Curie point of 300°C. Nickel ferrites have a Curie point of 600°C.

18.3.9 Springs. The elastic properties of metals deteriorates at high temperature. Phosphor-bronze is useless after exposure to 500°C. Resistance to fatigue failure in general decreases with increasing temperature. Nickel alloys such as Inconel appear to be suitable for high temperature, electrically conducting springs.

18.3.10 Lubricants. Ordinary lubricants either evaporate, decompose, or burn at less than 200°C. Graphite and various refractory flours have been tried but very little has been published as yet.

18.4 Thermal design of high temperature parts. It has been shown that the effectiveness of radiation and conduction in general increases, and convection decreases with increasing temperature. With regard to individual parts, radiation and convection usually involve the outer surface only, but conduction occurs throughout the volume of the part. Radiation involves the material, color, and surface finish. Convection involves the shape, size, and proportions of the body.

Certain general requirements can be stated more or less categorically.

1. Heat producing parts should have external surfaces of high emissivity to take advantage of radiation cooling, and should have a low thermal resistance to a heat sink. Ordinarily, heat flows from points inside the part through the wall or case to radiating and mounting surfaces. Since the mounting contact is included, the mounting surface should provide a low contact resistance. The conductive heat flow paths should be of short length and large area. The base and case materials should have high thermal conductivities at operating temperatures. The mounting surface should have a smooth mechanical finish, should be mechanically stiff to prevent warping and to assure intimate contact, and should be designed for pressure contact or thermal bonding to the supporting structure which, in turn, should be designed as a heat flow path to the sink.

2. Non heat producing, temperature sensitive parts should have outer surfaces of low absorptivity to prevent absorption of radiation. Since such parts will absorb some heat, they should also have a low resistance to a heat sink.

3. Some components, a relay for example, contain both heat sources and temperature sensitive elements. Good thermal design must provide for heat transfer and still protect the temperature sensitive elements.

4. Structural parts should, in general, have high absorptivity surfaces and high thermal conductance.

5. All materials deteriorate faster at increasing temperature. A rough rule-of-thumb is that the rate of chemical reaction doubles for every 10°C increase in temperature. Materials must be chosen to resist corrosion, oxidation, and any other deteriorative action at the operating temperature. Copper, for example, oxidizes rapidly above about 350°C when exposed to oxygen, and the black oxide spalls off as it forms. No organic material can survive much more than 200°C. A large part of the effort required to develop satisfactory high temperature parts is devoted to a search for satisfactory materials.

6. Mechanical strength and elastic properties deteriorates with increasing temperature. The larger cross sections required for mechanical strength improve the conductance of the heat flow paths. Springs present a serious high temperature problem. Fatigue failure is aggravated by increasing temperature.

7. The permeability of materials to gases varies with temperature. This must be considered when hermetic sealing is contemplated.

8. Gas pressure at constant volume varies directly with temperature. This also is important in hermetically sealed parts.

18.4.2 Examples of specific parts.

18.4.2.1 Electron tubes. High temperature tubes have been developed with three general types of configurations:

(1) Stacked Ceramic - These tubes consist of a stack of cylindrical spacers separating the disc-like elements. One or both ends of the stack is often an anode surface. The terminals are tabs or wires protruding from the cylindrical surface. The cylindrical outer surface is chiefly ceramic (alumina) and the ends are either alumina or anode metal (titanium, nickel, etc.).

(2) Ceramic Envelope - These tubes have a ceramic base with pin terminals of conventional form and the elements are enclosed in a ceramic cylindrical envelope.

(3) Metal Shell - These tubes have a pin-type base and the elements are enclosed in a metal envelope which may be the anode.

In the stacked ceramic type, heat will flow from the elements to the envelope chiefly by conduction, since the radiation path is oblique and the receiving surface is small. It is expected that the envelope will become a nearly constant temperature surface, since alumina is a fairly good thermal conductor and the structure is thermally homogeneous.

Cathode surface temperatures of the order of 750°C are required. In order to minimize the required heater power, the cathode should be thermally isolated. Such isolation is difficult to achieve in the stacked configuration because of conduction through the short cylindrical rings which form the tube envelope.

The ceramic envelope-type will act more like a conventional glass tube. The envelope will receive heat chiefly by radiation. Hot spots are expected at the base and at a zone of the envelope. No data have yet been found on the transparency of ceramic envelopes to infrared, but it is probably very slight.

Metal envelope tubes will probably develop a fairly uniform envelope temperature due to the high thermal conductivity. The kind of metal and its surface finish will determine the heat flow. Metals, in general, are not transparent to infrared; they either absorb or reflect. The sum of the reflectivity and emissivity is unity. It seems desirable to prevent the development of excessive temperature at the grids.

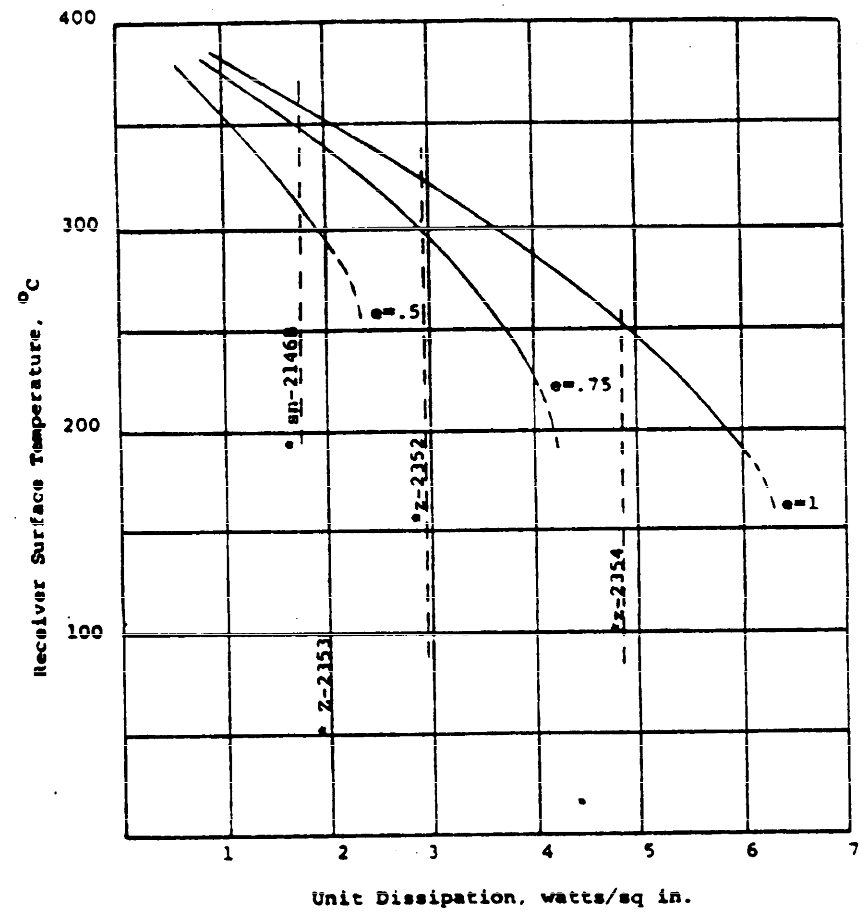
The thermal isolation of the cathode in envelope-type tubes is fairly good, but the large ceramic base will cause some thermal coupling from cathode to other elements. This coupling should be minimized.

In high temperature tubes the cathode temperature will probably be about the same as in conventional tubes, but the grid and plate temperatures will be substantially higher. Grid primary emission is supressed by proper choice of material. Radiative heat loss from the cathode will be less than in conventional tubes because of the smaller temperature difference. Thermal conduction will be greater because of the heavier leads required and the higher conductivity of ceramic materials.

Grids should have highly reflecting surfaces and should have low thermal conductance to cathode and plate. Plates should have a high thermal conductance to a sink. Short plate leads having a large cross sectional area (low thermal resistance) are recommended.

18.4.2.1.1 Radiative heat transfer from high temperature tubes. The unit heat dissipation from a constant temperature surface was calculated as a function of receiving surface temperature, for three values of emissivity, the emissivity of radiating, and receiving surfaces being the same. The results are plotted in Figure 224. The unit heat dissipations of several high temperature vacuum tubes are indicated to illustrate the cooling problem of these tubes. Figure 224 shows that the unit heat dissipation increases with decreasing temperature, but at a decreasing rate, and approaches a constant value which depends only on emissivity and radiating surface temperature.

18.4.2.1.2 Tube sockets. Sockets for high temperature parts must obviously use insulation which has satisfactory resistance at the temperature involved. If spring contacts are used, the spring material must retain its temper and strength at this temperature.



*Unit heat dissipations of some typical ceramic tube types

FIGURE 224. Radiation Heat Loss From 400°C Surface for Different Emissivities and Receiver Surface Temperatures

Many high temperature tubes do not have pin terminals, so the conventional type of socket is not applicable. Tube cooling will be chiefly by radiation and conduction because natural convection in air is relatively ineffective at high temperature. Tubes should be surrounded by an absorbent and/or conduction shield to receive the heat from the tube and conduct it to a heat sink, thus minimizing the thermal resistance to the sink by conduction and preventing radiation from the tube heating neighboring components.

No. 20 aluminum wire will conduct 0.004 watt per inch of length per °C temperature difference. Assuming five leads to a triode, each 1 in. long, and a Δt of 50°C, 1 watt could be carried by the leads. With #16 wire, this is more than doubled to 2.5 watts. A rough idea of the radiant heat transfer from a tube is obtained by the method given by Giedt (Reference 9). Consider a tube 0.75 in. diameter by 1.25 in. separation, assuming emissivity of 0.42. If the surface temperatures are 550°C for the tube and 525°C for the shield, 2.5 watts will flow from the tube to shield, 0.75 watts per in.^{2} if tube surface. Radiation, at the rather small temperature difference of 25°C, has roughly the same effectiveness as conduction through the leads. If the tube temperature rises to 575°C, the radiant heat loss increases to 1.57 watts per in.^{2}.

Tube sockets should be so designed as to surround the tube as nearly as possible with a heat absorbing structure of good thermal conductivity which is thermally bonded to a sink. This structure should be highly reflecting on the outside. The leads to the tube terminals should be large and substantial for good heat conduction. The purposes of a tube mount are:

a) To furnish mechanical support for the tube.

b) To furnish reasonably convenient means of making electrical contact to the tube terminals and to other points of the circuit.

c) To absorb or conduct the heat of the tube to a heat sink.

The possibility of potting tubes in sand or ceramics, as is done for high temperature transformers should be considered. The capsule walls should be heavy aluminum for good thermal conductivity and a ceramic insert would be required to carry the leads. The potting material should have good heat conductivity and, of course, poor electrical conductivity.

18.4.2.3 Terminals. In conventional low temperature equipment parts, terminals are universally designed for soft soldering. High melting solder, melting at 310°C, can be used up to 260°C (Reference 27). Above 260°C, brazing, welding, or a mechanical pressure-type of connection must be used. Terminals must be designed for the type of connection to be used, with due regard to the lowered strength and fatigue resistance of metals at high temperature.

Metal must be selected which can be easily joined to the hook-up wiring. For example, titanium is not suitable for terminals because of difficulty in welding or brazing.

FIGURE 178. Typical Spring-Type Thermal Links

Terminal spacing must be selected with due regard to decreased dielectric strength at high temperature.

Friction or spring-type terminals of plug-in parts must have a suitable high temperature spring material.

18.4.2.4 Hardware. Hardware must be designed with regard to oxidation, reduced strength, and creep at high temperature. The effects of temperature cycling over wide ranges are particularly troublesome. Screws which have been tightened at room temperature frequently break due to expansion stresses when heated. Expansion may also cause looseness with resulting increased vibration.

So far, the electronics industry has been so busy with the more conspicuous problems of high temperature operation that little attention has been given to special high temperature hardware. Screws, washers, terminal strips, grommets, and many other small items are just as necessary at high as at low temperatures, and their particular problems must receive attention.

18.4.2.5 Fuses. The operation of a fuse depends on a transient unbalance between the Joule heating in the fuse wire and rate of heat loss. The equation for transient temperature rise may be written as:

$$\Delta t = I^2 R_{elec} R_{th} - e^{-\frac{\tau}{Wc_p R_{th}}} \qquad (18\text{-}2)$$

Where:

Δt = temperature rise
I = fuse current
R_{elec} = electrical resistance
R_{th} = total effective thermal resistance
Wc_p = heat capacity of fuse
τ = time

Solving for the time required to cause a given temperature rise,

$$\tau = Wc_p R_{th} \ln \frac{1}{I^2 R_{elec} R_{th} - \Delta T} \qquad (18\text{-}3)$$

The general nature of this rather involved expression is illustrated in Figure 225. Occasionally, a fuse is designed to provide a time lag of a second or two at, perhaps, twice-rated current, but most fuses in electronic equipment are intended to blow in a fraction of a second. For very short blow time, equation 18-3 becomes approximately,

$$\tau \approx \frac{Wc_p \Delta t}{I^2 R_{elec}} \qquad (18\text{-}4)$$

The time to blow is thus, directly proportional to the heat capacity of the fuse and to the difference between the melting point and the fuse initial temperature, and inversely proportional to the heating effect. The thermal resistance is of secondary importance.

Certain requirements of fuse design are apparent from this rough analysis.

This steady-state fuse temperature, which will be higher than the ambient, must obviously be lower than the melting point.

The required temperature rise to blow at a given current becomes less as the ambient temperature rises. This speeds up the fuse action. This effect could be counteracted if the electrical resistance decreased with increasing temperature. A negative resistance temperature characteristic could thus be used to make fuse performance independent of ambient temperature.

A fast fuse should have a low thermal capacity.

The design of a fuse to perform satisfactorily over a wide range of ambient temperatures can be accomplished.

18.4.3 Parts thermal data. Figure 226 presents surface temperature rise vs. environmental temperature for a high temperature pyrolytic carbon resistor.

Figure 227 shows the surface temperature rise vs. environmental temperature for a high temperature external anode tube at full rating.

Figure 228 presents the surface temperature rise of the Mycalex (or equal) base (used as the printed circuit base) in a high temperature amplifier.

Figure 229 gives the surface temperature rise of a 3 watt high temperature wire wound resistor dissipating 1.1 watts.

18.5 Mounting, housing, connecting.

18.5.1 200°C Ambient-probable forms and arrangements. Most electronic parts for operation in the range of 100°C to 200°C will be similar in form to low temperature parts, but slightly larger and heavier. There will be increasing use of the high temperature tubes. These are made in several configurations, most of which require quite different heat transfer arrangements than conventional glass envelope tubes. Stacked ceramic and external anode tubes, for example, cannot be fitted with conductive tube shields. Some ceramic tubes have terminal tabs projecting from the cylindrical surface and parallel flow forced air would not be efficient because shields cannot be used to develop the necessary turbulence in narrow passages. Crossflow cooling would be indicated.

Circuitry may be somewhat more complicated because of the necessary temperature compensation over the wide operating range.

Hook-up wiring and structural parts will be of somewhat larger size. Since wiring cannot be laced with ordinary twine, there may be increasing use of manufactured multiconductor cable and of bare rigid wiring.

The fact that soft solder cannot be used at 200°C will affect the arrangement of parts. Since removal and replacement is much more difficult with hard solder, there may be increasing use of replaceable subassemblies, perhaps plugged into sockets. These might be printed circuits or potted units.

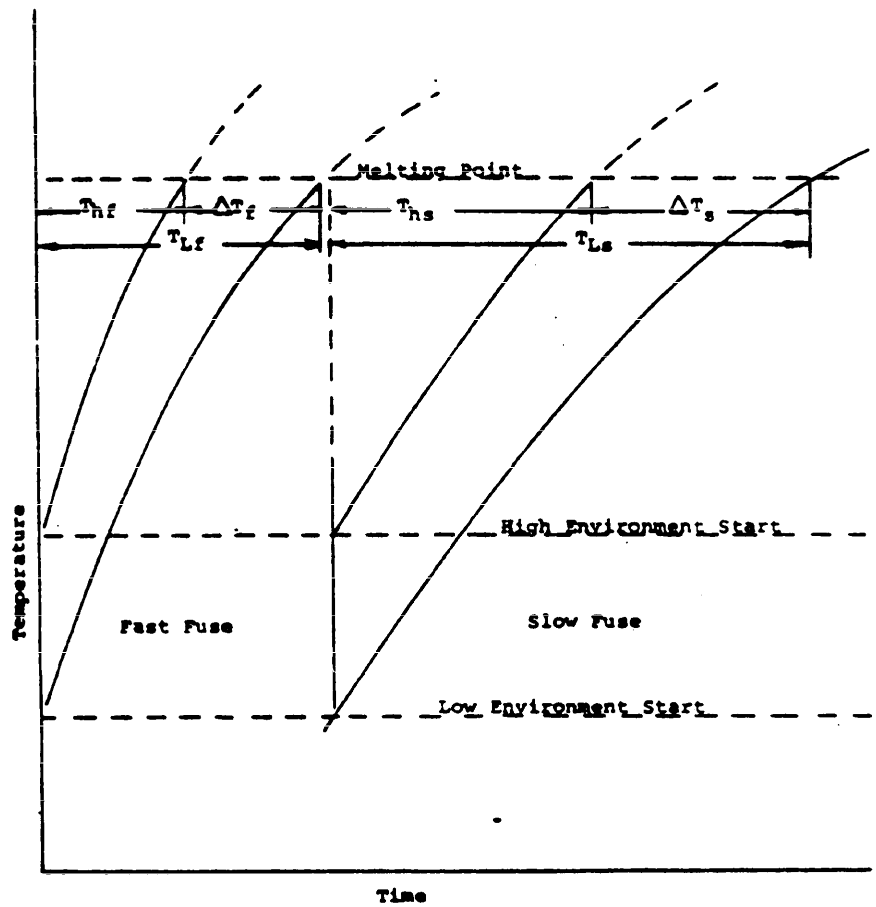


FIGURE 225. Fuse Characteristics

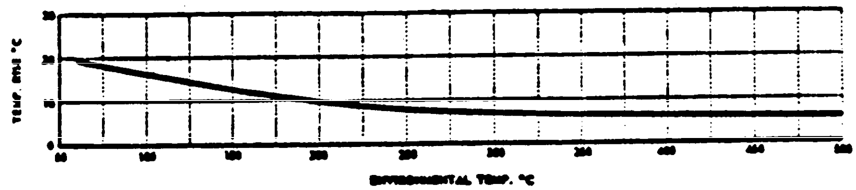


FIGURE 226. Surface Temperature Rise of Pyrolytic Carbon Resistor Dissipating .25 Watts

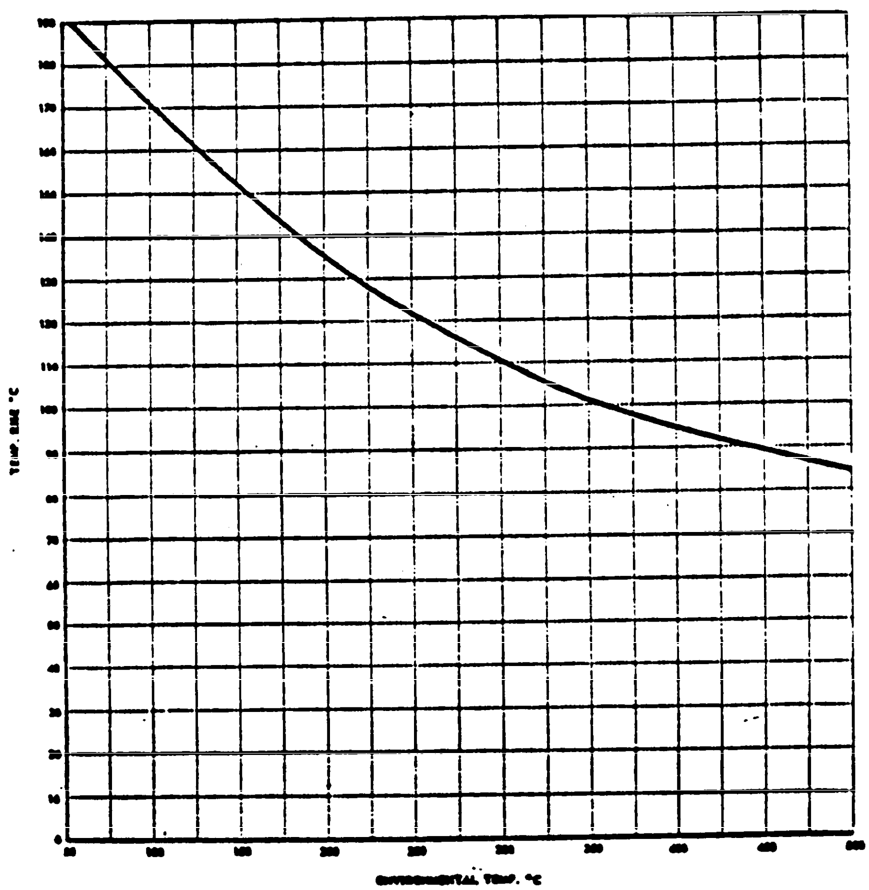


FIGURE 227. Surface Temperature Rise of External Anode Tube

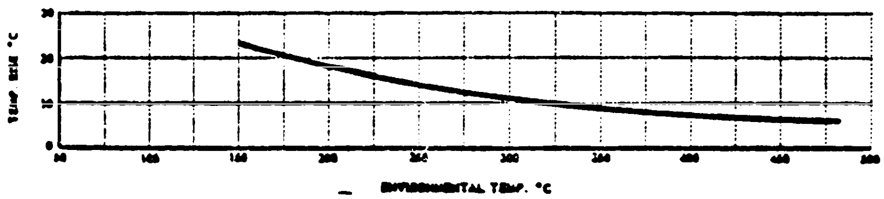


FIGURE 228. Temperature Rise of Mycalex* Base
*or equivalent

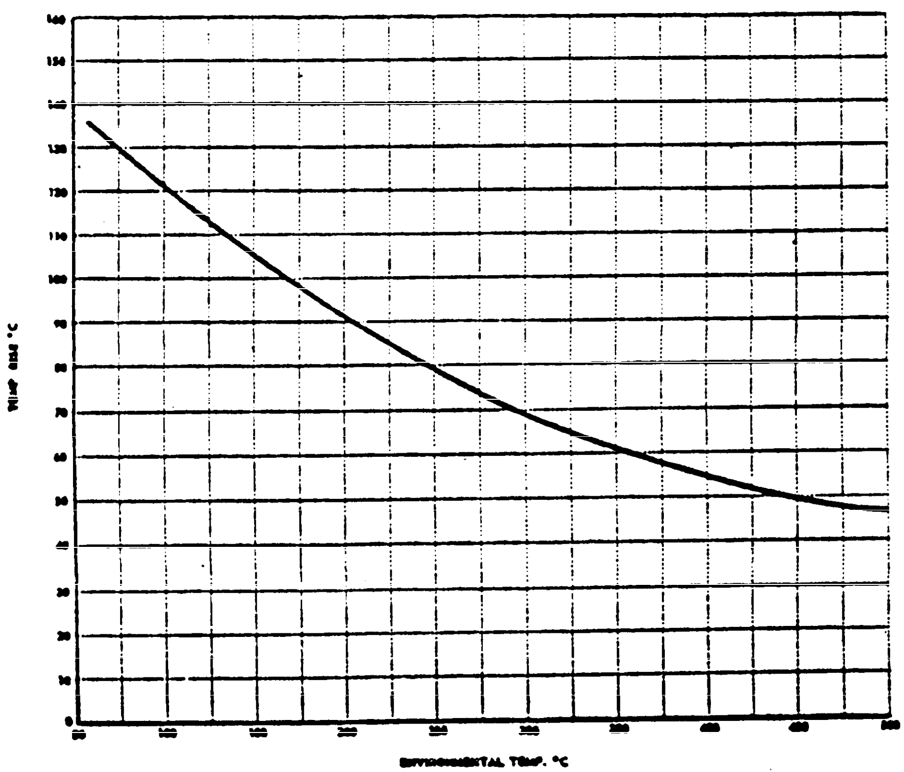


FIGURE 229. Surface Temperature Rise of Miniature 3 Watt Wire Wound Resistor Dissipating 1.1 Watts

The form and arrangement of parts is determined partly by heat transfer considerations:

a. Free convection with gases should not be relied on.

b. Forced convection with gases is effective, but the blower power will increase somewhat.

c. Liquid cooling may be effected with certain silicone, chlorine, and fluorine compounds, which are suitable for immersion and direct spraying since they are insulators. Petroleum oils cannot be used because of sludging, high vapor pressure, and low flash point.

d. Radiation will increase in effectiveness but must be controlled and directed.

e. Conduction will increase in effectiveness because of the larger metal sections required.

Insulation for 200°C is flexible, resilient, and impervious to moisture, so that sealing does not call for any radical differences in construction. In general, the parts arrangements used in low temperature equipments can also be used in the higher temperature range.

18.5.2 350°C and higher ambient. Electronic parts capable of functioning at 350°C and higher will be quite different from conventional parts, in shape, size, and materials. High temperature problems seem to indicate increased application of the concept of packaged "throw-away" modules. Connections must be welded or brazed, or of a mechanical type, making parts replacement very difficult. Oxidation problems suggest hermetic sealing in inert atmosphere. High temperature insulation is brittle, requiring a new treatment of mechanical shock and vibration. Since very few equipments have been built for this temperature range, there is almost no information available on the relative merits of different configurations. Some general ideas on several possible configurations follow.

18.5.2.1 Conventional chassis. Structural parts and hook-up wire sizes are increased. Terminal spacing is increased. Allowance must be made for thermal expansion, creep, corrosion. All materials must resist oxidation. Radiation control surfaces are required. All of these requirements increase size and weight, making this type of construction less attractive with increasing temperature.

18.5.2.2 Printed circuitry. The baseboards must be ceramic. Soft solder cannot be used. Mounting and inter-board connections must be designed to allow for expansion stresses. The ceramic boards will have a relatively high thermal conductivity and if thermally bonded to mounting structure will contribute to the cooling. However, ceramic base plates are brittle and thus, require some resiliency in mounting. Allowance for thermal stresses must be made.

18.5.2.3 Potted modules. As temperature increases, oxidation becomes more troublesome. The idea of potting in refractory material is attractive. Some designers advocate potting in a fine refractory flour or sand, with air replaced by inert gas. The case would be heavy aluminum or stainless steel, with heat flow by conduction through the potting material and case. Radiant heat exchange among parts is prevented. Convection is not used. Such a module is something like a homogeneous block of conductive material containing a distribution of small heat sources. It should be fairly easy to design such a package for hot spot temperature rises (above outside ambient) of quite a small value, say, less than 50°C.

18.5.2.4 Liquid cooling. The only liquid coolants which appear to be usable in this temperature range are metals. Certainly, this is not an attractive prospect for several reasons, i.e., weight, pumping methods, leaks, servicing difficulties. It would seem that gases must be relied on for heat transport if convection is mandated.

18.5.3 Mountings. In addition to furnishing mechanical support, it is expected that the mounting structure will furnish an important conductive heat path. The structure should therefore be made of high conductivity material. All joints should be thermally bonded by welding or clamping under pressure. This bonding is important in such contacts as chassis to racks and subchassis to chassis. The joint surfaces must be protected against corrosion.

Structural elements receive heat by conduction and radiation. The surfaces should be of high absorptivity and should be oriented to present large areas normal to radiation paths from heat sources.

Since ultra-high temperature parts tend to be heavier, the supporting structure must be more substantial to withstand vibration stresses. Reduced strength of materials at high temperature also requires heavier sections.

18.5.4 Methods of electrical connection. Lead-tin solders melt at around 180°C and are useless in high temperature equipment. Brazing is a more complicated technique than soft soldering and would be troublesome for maintenance. Spot welding is easy with stainless steel; more troublesome but practicable with high conductivity metals like silver, and very difficult with aluminum and titanium.

Pressure-type mechanical connections are recommended by some investigators and small hand crimping tools are available. Cold welding and ultrasonic welding techniques would seem to be applicable in some cases.

18.5.5 Potting and encapsulation. Sealing is required in high temperature equipment for the same reasons as at low temperatures. It also can protect against oxidation at high temperature if inert gas filler is used. The heat conductivity of materials suitable for potting and encapsulating high temperature parts is high.

Modules will probably be connected together by special high temperature plugs and jacks. It is important to provide thermal conducting paths from each module to the heat sink. Large contact area and considerable contact pressure are required. Each module would thus have a heat conducting as well as an electrical plug, and the interconnecting structure would contain a "heat bus" leading to a sink.

APPENDIX A

Table of Symbols and Nomenclature

Notes: Some symbols used in isolated places in the text may not appear in this Table. They are defined where used. Many of these symbols are used with subscripts and occasionally superscripts to denote particular quantities. These are defined where used and may not appear in this Table.
In certain paragraphs symbols other than those given in this Table are used. These are defined where used.

Symbol	Definition	Units	Dimensional Form*
A	Area	Sq.in.	L^2
a	Convection Modulus $(= g\rho^2\beta c_p/k\mu)$	1/(cu.in.-°C)	1/LT
B,b	Width	in.	L
Btu.	British Thermal Unit, English System Unit of Heat (1 Btu = 17.568 Watt-Sec)	watt-Sec	ML^2/t^2
B_f	Flow Factor	in.	L
C	A constant	-	-
C, C_{th}	Thermal Capacitance	watt-Sec/°C	ML^2/t^2T
c_p	Specific heat at constant pressure	watt-Sec/(lbm-°C)	L^2/t^2T
c_v	Specific heat at constant volume	watt-Sec/(lbm-°C)	L^2/t^2T
CFM,cfm	Flow Rate, cubic feet/minute	(cu.in/sec.)28.8	L^3/t
D,d	Diameter; characteristic dimension	in.	L
D_e	Equivalent or Hydraulic diameter	in.	L

* In M-L-T-t-θ system, M = mass; L = length; T = temperature; t = time; θ electrical charge

Symbol	Definition	Units	Dimensional Form
D.E.	Denotes a design equation	-	-
d( )	Differential operator	-	-
E	Internally generated heat	watts	ML^2/t^3
e	Efficiency	-	0
E,e,(ϵ)	Emissivity	-	0
ELSD	Equivalent length of straight duct	in	L
e	Partial vapor pressure	lbf/sq in	M/Lt^2
F,f	A factor	-	-
f	Friction factor	-	0
G	Thermal conductance	watts/°C	ML^2/t^3T
G	Electrical conductance	mho	$t\theta^2/ML^2$
G	Mass flow density; mass velocity of flow	lbm/(sec-in^2)	M/L^2t
Gr	Grashof Number $=D^3g\Delta T\beta/\nu^2$	-	0
g	Gravitational acceleration	in/sec^2	L/t^2
go	g at sea level (=386)	in/sec^2	L/t^2
H	Hydraulic head	in	L
HR	Hydraulic radius	in	L
h	Enthalpy	watt-sec/lbm	L^2/t^2
h	Coefficient of heat transfer	watts/(sqin-°C)	M/Tt^3
hfg	Latent heat of vaporization	watts-sec/lbm	L^2/t^2

Symbol	Definition	Units	Dimensional Form
K	A coefficient, especially for head loss	-	0
k	Thermal conductivity	(watts/sq in)/ (^{o}C/in)	ML/Tt^3
kc	Ratio of specific heats, c_p/c_v	-	0
L, l	Length	in	L
lbf	Pound force	lbf	ML/t^2
lbm	Pound mass	lbm	M
LMTD	Logarithmic mean temperature difference	^{o}C	T
M	Mass	lbm	M
M	Velocity of approach factor	-	0
M	Mach No = V/Sonic Velocity	-	0
m	Mass flow rate	lbm/sec	M/t
m	A constant	-	-
N	Rotational speed	rev/sec	1/t
N	Number of items (rows, tubes,etc)	-	0
Nu	Nusselt No = $h_c L/k$	-	0
NTU	Number of heat transfer units (heat exchangers)	-	0
P	Power	watts	ML^2/t^3
Pr	Prandtl No = $c_p \mu /k$	-	0
p	Pressure	lbf/sq in	M/Lt^2

Symbol	Definition	Units	Dimensional Form
Q	Volumetric flow rate	cu in/sec	L^3/t
q	Heat transfer rate	watts	ML^2/t^3
R	Gas constant (=8.317)	watt-sec/(lbm-$^\circ$C)	L^2/Tt^2
R	Thermal resistance	$^\circ$C/watt	t^3T/ML^2
R	Electrical resistance	ohms	$ML^2/t\theta^2$
Re	Reynold's No = $\rho VD/\mu$	-	0
r	Radius	in	L
S	Spacing parameter	-	0
S.G.	Specific gravity	-	0
St	Stanton No = Nu/(Re·Pr) =$h/Vc_p\rho$	-	0
s	Pitch of tube banks; a distance	in	L
s	Entropy	watt-sec/(lbm-$^\circ$C)	L^2T/t^2
s	LaPlace transform of d/dt	-	-
T*	Temperature, $^\circ$K	$^\circ$K	T
T*	Time	sec	t
t*	Temperature, $^\circ$C	$^\circ$C	T
t*	Time	sec	t
U	Overall heat transfer coefficient	watts/(sq in-$^\circ$C)	M/Tt^3

* Normally, t represents temperature in $^\circ$C, and T represents time in seconds or absolute temperature in $^\circ$K. Occasionally it is convenient or customary to use t for time (as in differential expressions). Exceptions to normal usage are defined whenever encountered in the text.

Symbol	Definition	Units	Dimensional Form
V	Velocity	in/sec	L/t
V	Volume	cu.in.	L^3
v	Specific volume	cu in/lbm	L^3/M
W	Weight (=ρg)	lbf.	ML/t^2
W	Mass flow rate	lbm/sec	M/t
W,w	Watts	watts	M
w	width	in	L
X	Geometric factor	-	0
x	Distance, X direction	in	L
Y	Compressibility factor	-	0
Y	Geometric factor	-	0
y	Distance, Y direction	in	L
Z	Geometric factor	-	0
z	Distance, Z direction	in	L
°C	Temperature, degrees Celsius	°C	T
°F	Temperature, degrees Fahrenheit	°F	T
°K	Temperature, degrees Kelvin (=°C + 273)	°K	T
°R	Temperature, degrees Rankine (=°F + 459)	°R	T

Symbol	Definintion	Units	Dimensional Form
α	Thermal diffusivity = $k/c_p\rho$	sq/sec	L^2/t
α	Absorbtivity	-	0
β	Volumetric temperature expansion coefficient	$1/^{o}C$	1/T
Δ	Difference (e.g. ΔT)	-	-
δ	Boundary layer thickness	in	L
e, E, ϵ, ε	Effectiveness (esp. heat exchangers)	-	0
e, E, ϵ, ε	Emissivity	-	0
ϵ	Surface roughness (absolute)	in	L
θ	Electrical charge	columbs	θ
η	Rating factor for coolants	-	0
η	Efficiency, effectiveness	-	0
λ	Latent heat	watt-sec/lbm	L^2/t^2
μ	Dynamic (absolute) viscosity (poise) (= $\rho\nu$)	lbm/(in-sec)	M/Lt
ν	Kinematic viscosity (stokes) (= μ/ρ)	in^2/sec	L^2/t
π	A constant = 3.1415926	-	0
ρ	Reflectivity	-	0
ρ	Mass density	lbm/cu.in.	M/L^3
Σ	Summation operator	-	-

Symbol	Definition	Units	Dimensional Form
σ	Surface tension	lbf/in	M/t^2
σ	Electrical conductivity	ohms-sq in/in	$ML^3/t\theta^2$
σ	Stephan-Boltzman constant (=36.8 x 10^{-12})	watts/(sqin-^{o}C	MT/t^3
τ	Transmissivity	-	0
Φ	Flux, esp radiant energy	watts	ML^2/t^3
ϕ	Angular measure	deg,radians	0
ω	Solid angular measure	steradians	0

APPENDIX B

Table of Conversion Factors

Length

The preferred unit in this Handbook is the inch. Multiply the measures below by the factor indicated to obtain inches:

foot	1.2×10
yard	3.6×10
mile	6.336×10^{4}
centimeter	3.937×10^{-1}
meter	3.937×10
micron	3.937×10^{-5}
Angstrom	3.937×10^{-9}
mils	$1. \times 10^{-3}$

Area

The preferred unit in this Handbook is the square inch. Multiply the measures below by the factor indicated to obtain square inches:

square feet	1.44×10^{2}
square meters	1.55×10^{3}
square mils	$1. \times 10^{6}$
circular mils	7.854×10^{-7}

Volume

The preferred unit in this Handbook is the cubic inch. Multiply the measures below by the factor indicated to obtain cubic inches:

cubic feet	1.728×10^{3}
cubic centimeters	6.1024×10^{-2}
liters	6.10254×10
gallons (U.S. liq)	2.31×10^{2}

Mass

The preferred unit in this Handbook is the pound (lbm). Multiply the measures below by the factor indicated to obtain pounds lbm:

gram	2.205×10^{-3}
kilogram	2.205
ounce	16.
slug	32.174

Force

The preferred unit in this Handbook is the pound (lbf). Multiply the measures below by the factor indicated to obtain pounds lbf:

dynec	2.248×10^{-6}
newtons	2.248×10^{-1}
poundals	3.108×10^{-2}

Temperature

The preferred unit in this Handbook is the degree Celsius (^{o}C). Perform the indicated operations on the measures below to obtain ^{o}C:

	Absolute value	Difference
Degree Fahrenheit (^{o}F)	$(^{o}F - 32)/1.8$	1/1.8
Degree Centigrade (^{o}C)	$= {}^{o}C$	1
Degree Kelvin (^{o}K)	$^{o}K + 273$	1
Degree Rankine (^{o}R)	$(^{o}R + 427)/1.8$	1/1.8

Time

The preferred unit in this Handbook is the second. Multiply the measures below by the factor indicated to obtain seconds:

minute	6.0×10
hour	3.6×10^{3}
day	8.64×10^{4}
millisecond	1×10^{-3}

Velocity

The preferred unit in this Handbook is inches/second. Multiply the measures below by the factor indicated to obtain inches/second

inches/minute	1.667×10^{-2}
feet/second	1.2×10
feet/minute	0.2
miles/hour	17.6
kilometers/hour	10.936

Acceleration

The preferred unit in this Handbook is inches/sec^2. Multiply the measures below by the factor indicated to obtain inches/sec^2.

feet/sec^2	1.2×10
kilometer/(hr-sec)	1.3167×10
meters/sec^2	3.658
miles/(hr-sec)	8.1818

Work, Energy

The preferred unit in this Handbook is the (watt-second). Multiply the measures below by the factor indicated to obtain watt-sec:

Btu	1.05435×10^{3}
foot pound	1.3558
kilowatt-hour	3.6×10^{6}
joule	1.0
Volt-coulomb	1.0
ergs	$1. \times 10^{-7}$
electron volts	1.6021×10^{-19}

Power

The preferred unit in this Handbook is the watt. Multiply the measures below by the factor indicated to obtain watts:

Btu/hr	2.9288×10^{-1}
foot-pound/sec	1.35582
horsepower (elec.)	7.46×10^{2}
cal., gm./hr	1.16222×10^{-3}
ergs/sec	$1. \times 10^{-7}$
joules/sec	1.0

<u>Thermal Conductivity, k</u>

The preferred unit in this Handbook is (watts/sq in)/(°C/in). Multiply the measures below by the factor indicated to obtain (watts/sq in)/(°C/in):

(Btu/hr-sq ft)/(°F/in)	3.663×10^{-3}
(Btu/hr-sq ft)/(°F/ft)	0.0440
(watts/sq cm)/(°C/cm)	2.540
(Cal/sq cm-sec)/(°C/cm)	10.63
(Kg cal./sq.m-hr)/(°C/m)	0.0295

<u>Thermal Resistivity</u>

The preferred units in this Handbook is (°C/in)/(watt/sq in). Multiply the measures below by the factor indicated to obtain (°C/in)/(watt/sq in):

(°F/in)/(Btu/hr-sq ft)	2.73×10^{2}
(°F/ft)/(Btu/hr-sq ft)	2.275×10
(°C/cm)/(watt/sq.cm)	3.937×15^{-1}
(°C/cm)/(Cal/sq cm-sec)	9.405×10^{-2}
(°C/m)/(Kg cal./sq m-hr)	3.386×10

<u>Heat Transfer Coefficient</u>

The preferred unit in this Handbook is watts/(sq in-°C). Multiply the measures below by the factor indicated to obtain watts/(sq in-°C):

Btu/(hr - sq ft - °F)	3.663×10^{-3}
watts/(sq. cm - °C)	6.452
Cal/(sec - sq cm - °C)	2.701×10
Kg cal/(hr - sq m - °C)	7.502×10^{-4}

Specific Heat Thermal Capacity

The preferred unit in this Handbook is watt-seconds/(pound mass - °C). Multiply the measures below by the factor indicated to obtain watt-sec/(lbm - °C):

Btu/(hr - °F)	0.527
Cal/(grm - °C)	9.2048×10^{-3}

Pressure

The preferred units in this Handbook are pounds force/square inch. Multiply the measures below by the factor indicated to obtain lbf/in^2:

Atmospheres	14.696
Bars	14.504
Inches Hg (0 °C)	0.4912
Inches H_2O (4 °C)	3.606×10^{-2}
mm Hg (0 °C)	1.934×10^{-2}
Kg/sq cm	14.223
Feet of air (1 atm, 15°C)	5.208×10^{-4}

Mass Density

The preferred units in this Handbook are pound mass/cubic inch. Multiply the measures below by the factor indicated to obtain lbm/cu.in :

grams/cu. cm.	$3.6.27 \times 10^{-2}$
Kg/cu. meter	3.6127×10^{-5}
lbm/cu.ft.	5.787×10^{-4}

Mass Flow Rate

The preferred units in this Handbook are pound mass/sec. Multiply the measures below by the factor indicated to obtain lbm/sec:

lbm/hr	2.785×10^{-4}
lbm/min	1.667×10^{-2}

Volume Flow Rate

The preferred units in this Handbook are cubic inches/second. Multiply the measures below by the factor indicated to obtain cu. in/sec.

CFM (cu.ft/min)	$\times 10^{-2}$
GPM (gal/min)	3.85

Surface Tension

The preferred units in this Handbook are pound force/inch. Multiply the measures below by the factor indicated to obtain lbf/in:

dynes/cm	1.837×10^{-4}

Viscosity (dynamic)

The preferred units in this Handbook are pound mass/(inch-second). Multiply the measures below by the factor indicated to obtain lbm/(in-sec):

poise	5.6×10^{-3}
centipoise	5.6×10^{-5}
lbm/(ft-sec)	8.333×10^{-1}
lbf sec/sq in	1.389×10^{-12}
lbf sec/sq ft	2.0×10^{-10}

Appendix C

Table of Constants

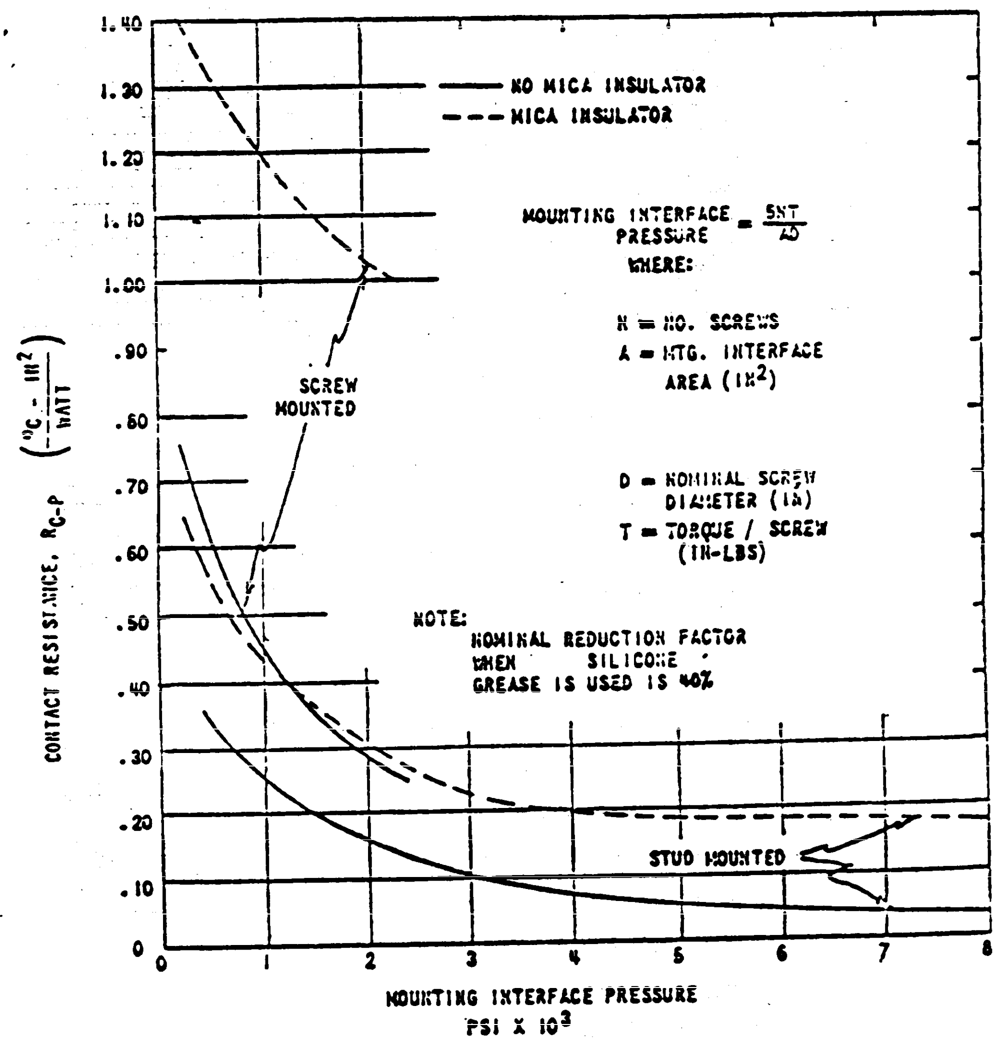
Symbol	Constant	Value	Units
g_o	gravitational acceleration at sea level	3.86×10^2	in/sec^2
σ	Stephan-Boltzmann constant	3.68×10^{-13}	watts/(sq in ^oC)
π	Pi	3.1415926	-
R	Universal Gas Constant	8.317	watt-sec/(lbm^oC)
c	Velocity of light in vacuum	1.18×10^{10}	in/sec
v_s	Velocity of sound in air @ STP	1.305×10^3	in/sec

Appendix D. Contact Resistance Data

Description	Environmental pressure	Approximate interface pressure, psi	R_q (°C)(in.^{2})/watt
Small stud-mounted components (such as stud-mounted transistors)	Sea level	5,000 500	0.05 0.50
	High vacuum	5,000 500	0.08 0.60
Mounting feet of equipment with contact areas of about 1 in.^{2}	Sea level	1,000 100	0.5 1.0
	High vacuum	1,000 100	2.0 5.0
Large-surface contact areas	Sea level	100 10	1.0 3.0
	High vacuum	100 10	7.0 20.0

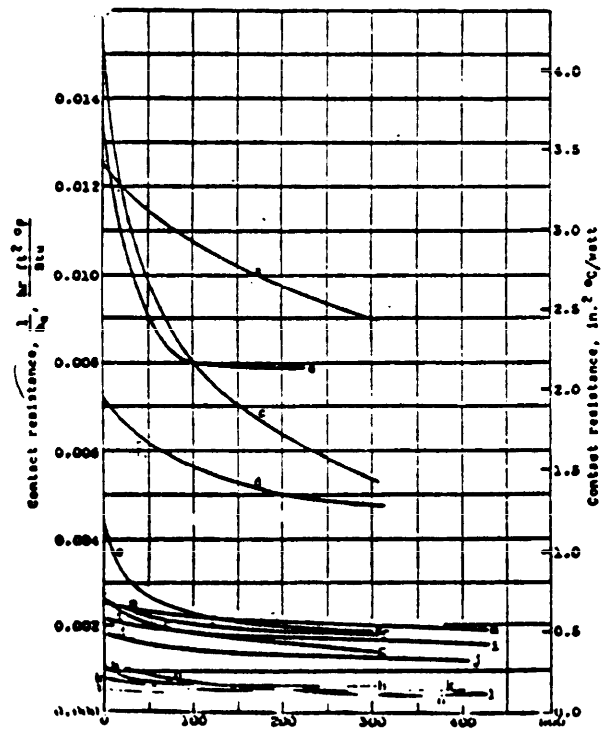
EFFECT OF CONTACT PRESSURE AND ALTITUDE ON INTERFACE THERMAL RESISTANCE

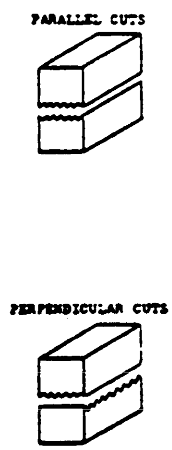
TRANSISTOR TO METAL HEAT DISSIPATOR CONTACT RESISTANCE



STEEL, BARE SURFACE (0 TO 500 PSI) - Solid blocks

Curve	Material	Finish	Roughness Rms (μ in.) block 1 2	Fluid in gap	Temp (°F)	Condition	Scatter of data
a	Cold rolled steel	Shaped	1000-1000	Air	200	Parallel cuts, rusted	
b	Cold rolled steel	Shaped	1000-1000	Air	200	Parallel cuts, clean	
c	Cold rolled steel	Shaped	1000-1000	Air	200	Perpendicular cuts, clean	
d	Cold rolled steel	Milled	125-125	Air	200	Parallel cuts, rusted	
e	Cold rolled steel	Milled	125-125	Air	200	Parallel cuts, clean	
f	Cold rolled steel	Shaped	63-63	Air	200	Perpendicular cuts, clean	
g	Cold rolled steel	Shaped	63-63	Air	200	Parallel cuts, clean	
h	Cold rolled steel	Lapped	4-4	Air	200	Clean	
i	416 Stainless	Ground	100-100	Air	200		± 22%
j	416 Stainless	Ground	100-100	Air	400		± 17%
k	416 Stainless	Ground	30-30	Air	200		± 9%
l	416 Stainless	Ground	30-30	Air	400		± 5%
m	416 Stainless 7075(75S)T6, Al	Ground	30-65	Air	200	Heat flow from stainless to aluminum	± 26%
n	7075(75S)T6 Al to stainless	Ground	30-65	Air	200-400	Heat flow from aluminum to stainless	± 30%

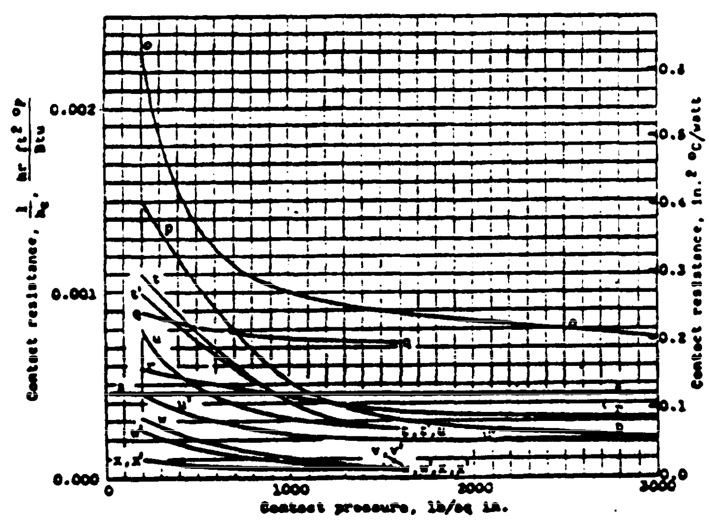




Courtesy - G.E.

STEEL, BARE SURFACES AT HIGH PRESSURES (200 TO 3000 PSI) - Solid Blocks

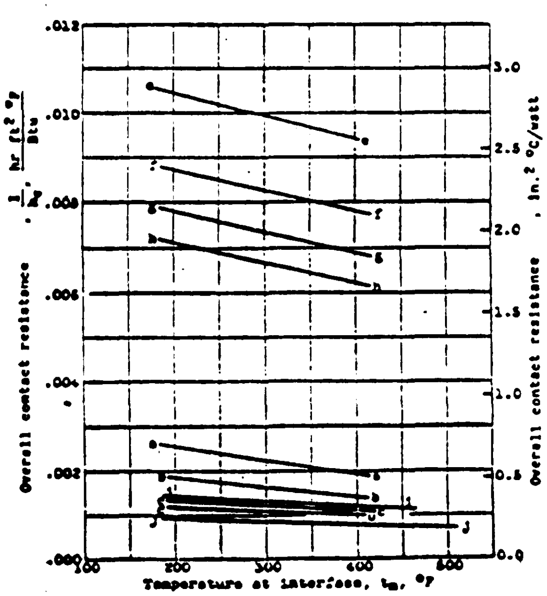
Curve		Material	Finish	Roughness Rms (μ in.) block 1 2	Fluid in gap	Temp (°F)	Condition
o		Carbon steel (.22% C, .48%Mn, .06% Si, .035% S, .006% P) anneal	Ruled	3-3320	Air	180	Clean
p			Ruled	3-1660	Air	180	Clean
q			Ruled	3-85	Air	180	Clean
r			Ruled	3-32	Air	140	Clean
s			Lapped	3-3	Air	180	Clean
t			Ruled	3-3320	Argon	400	Clean
t'			Ruled	3-3320	Argon	600	Clean
u			Ruled	3-55	Argon	400	Clean
u'			Ruled	3-55	Argon	600	Clean
v			Ruled	3-28	Argon	400	Clean
v'			Ruled	3-28	Argon	600	Clean
w			Ruled	3-16	Argon	400	Clean
w'			Ruled	3-16	Argon	600	Clean
x			Lapped	3-3	Argon	400	Clean
x'			Lapped	3-3	Argon	600	Clean



Courtesy GE

STEEL, WITH SANDWICH MATERIAL - Solid blocks

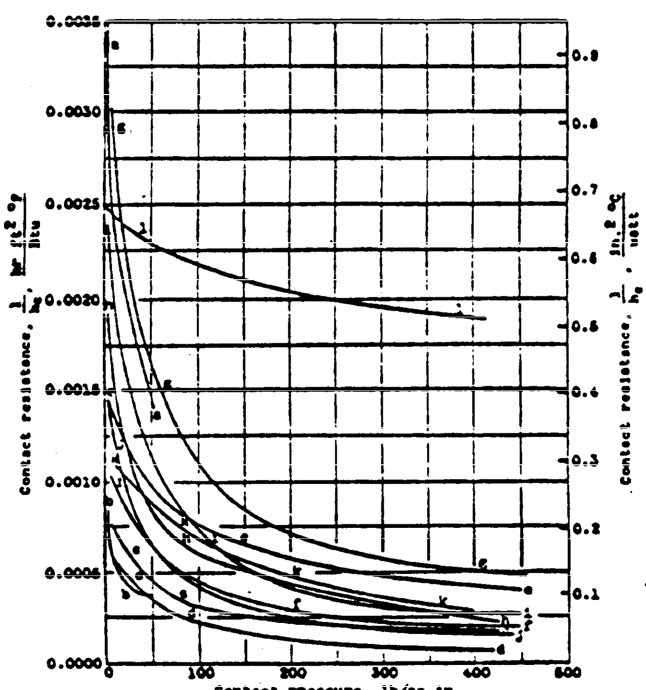
Curve		Material	Finish	Roughness Rms (μ in.) block 1 2	Fluid in gap	Contact pressure (psi)	Sandwich material	Scatter of data
a		416 Stainless	Ground	100-100	Air	5	0.0010 in. brass shim	± 2%
b			Ground	100-100	Air	90	0.0010 in. brass shim	± 3%
c			Ground	100-100	Air	240	0.0010 in. brass shim	±10%
d			Ground	100-100	Air	425	0.0010 in. brass shim	±10%
e			Ground	100-100	Air	5	0.010 in. asbestos sheet	± 6%
f			Ground	100-100	Air	90	0.010 in. asbestos sheet	± 4%
g			Ground	100-100	Air	240	0.010 in. asbestos sheet	± 0%
h			Ground	100-100	Air	425	0.010 in. asbestos sheet	± 0%
i			Ground	60-20	Air	7	0.0010 in. brass shim	±10%
j			Ground	60-20	Air	7	0.0008 in. aluminum foil	±10%



Courtesy G.E.

ALUMINUM, BARE SURFACES - Solid blocks

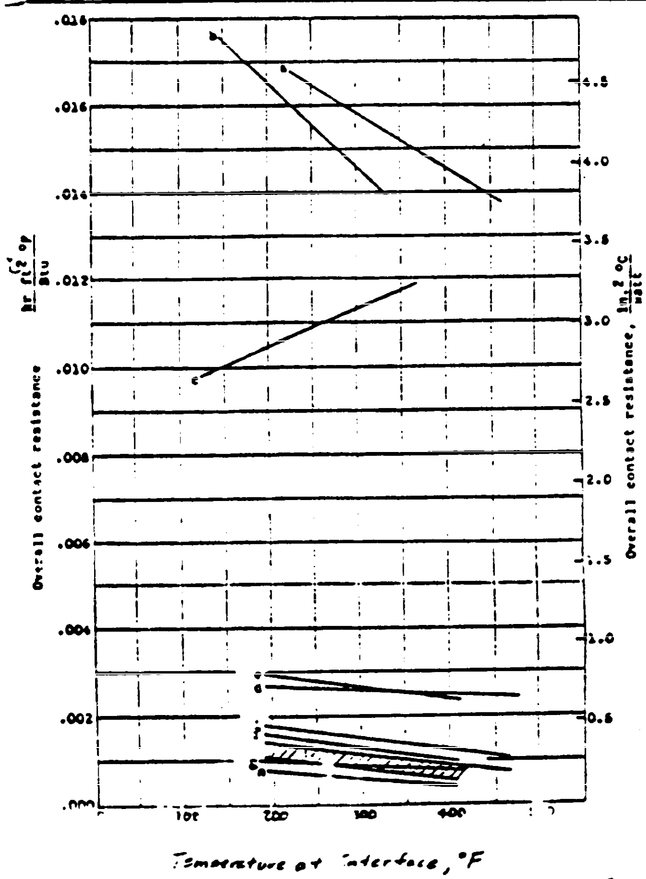
Curve		Material	Finish	Roughness Rms (μ in.) block 1 2	Fluid in gap	Temp (°F)	Condition	Scatter of data
a		6151 (A51S)	Ground	60-60	Air	300	Rough surface	
b		6151 (A51S)	Ground	16-16	Air	400	Smooth surface	
c		7075 (75S) T6	Ground	10-10	Air	200	(These data represent 22 runs on 2 samples)	±100%
d		7075 (75S) T6	Ground	10-10	Air	400	(These data represent 22 runs on 2 samples)	± 65%
e		7075 (75S) T6	Ground	65-65	Air	200		± 18%
f		7075 (75S) T6	Ground	65-65	Air	400		± 13%
g		7075 (75S) T6	Ground	120-120	Air	200		± 21%
h		7075 (75S) T6	Ground	120-120	Air	400		± 25%
i		7075 (75S) T6	Ground	10-120	Air	200		± 17%
j		7075 (75S) T6	Ground	10-120	Air	400		± 17%
k		7075 (75S) T6 416 Stainless	Ground	65-30	Air	200	Heat flow from aluminum to stainless	± 30%
l		416 Stainless 7075 (75S) T6	Ground	65-30	Air	200 to 400	Heat flow from stainless to aluminum	± 28%



Courtesy G. E.

ALUMINUM, WITH SANDWICH MATERIAL - Solid blocks

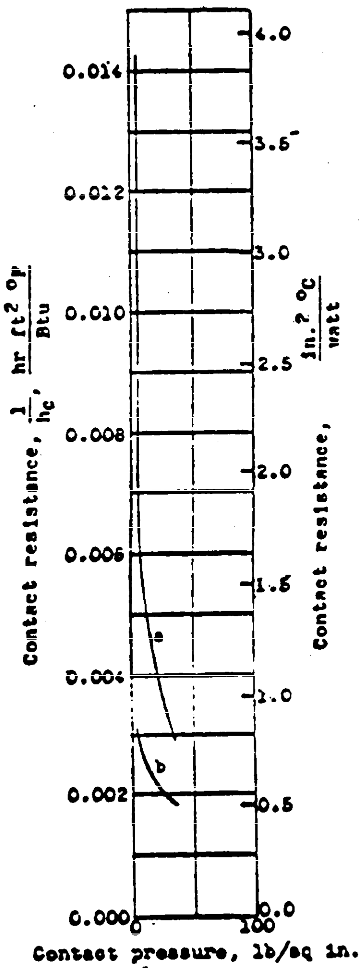
Curve	Material	Finish	Roughness Rms μ in. block 1	Roughness Rms μ in. block 2	Fluid in gap	Press. (psi)	Sandwich material	Scatter of data
a	7075 (75S) T6	Ground	12-	12	Air	7	0.010 in. asbestos sheet	± 2%
b	7075 (75S) T6	Ground	12-	12	Air	7	Redux (8)(10)	±10%
c	7075 (75S) T6	Ground	30-	30	Air	7	Metlbond cement (8)(10)	± 6%
d	7075 (75S) T6	Ground	12-	12	Air	7	Zinc chromate primer(8,10)	±15%
e	7075 (75S) T6	Ground	120-	120	Air	5	0.0010 in. brass shim	± 8%
f	7075 (75S) T6	Ground	120-	120	Air	90	0.0010 in. brass shim	± 6%
g	7075 (75S) T6	Ground	120-	120	Air	240	0.0010 in. brass shim	± 3%
h	7075 (75S) T6	Ground	120-	120	Air	425	0.0010 in. brass shim	± 4%
i	7075 (75S) T6	Ground	12-	12	Air	5	0.0010 in. brass shim	± 3%
j	7075 (75S) T6	Ground	12-	12	Air	5	0.0008 in. aluminum foil	± 3%



Courtesy G.E.

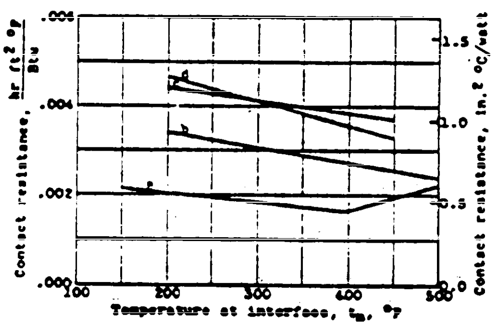
METALS OTHER THAN STEEL OR ALUMINUM - Solid blocks

Curve	Material	Finish	Roughness Rms (μ in.) block 1	2	Fluid in gap	Temp (°F)	Condition	Scatter of data
a	Copper	Ground	-		Vacuum	68	Clean	-
b	Gold, silver	Ground	-		Vacuum	68	Clean	-



RIVETED JOINTS - Solid blocks

Curve		Material	Finish	Roughness Rms(μ in.) block 1 2	Fluid in gap	Number of rivets	Spacing of rivets	Scatter of data
a		7075 (75S) T6 aluminum, sample 1	Ground	12-12	Air	1	-	± 7%
b		7075 (75S) T6 aluminum, sample 1 reheat and sample 2	Ground	12-12	Air	1	-	±13%
c		7075 (75S) T6 aluminum, sample 3	Ground	12-12	Air	3	Equally on 1.5 in. dia circle	± 5%
d		7075 (75S) T6 aluminum	Ground	30-30	Air	1	-	± 5%



APPENDIX E

THERMAL CONDUCTIVITY VALUES (k) of COMMONLY USED MATERIALS

Notes:

1. Values are generally at 25 or 100°C, and may be considered invariant over the temperatures normally encountered by electronic systems.

2. Values for metals can change drastically with small amounts of alloying elements, and in some instances, with heat treatment.

3. Values of plastics and elastomers can change drastically with the use of filler materials. Oriented fibrous fillers may cause non-isotropic thermal conductivity.

Material	k - (watts/sq.in.) /(°C/in.)
Metals	
Ferrous	
Iron, cast, grey/malleable	1.23 - 1.32
Iron, cast, nodular/ductile	0.79 - 0.88
Iron, wrought	1.52
Iron-base superalloys (Cr-Ni), wrought	
Incoloy* 901; w545; AMS 5700	0.46 - 0.48
19 - 9DL, Unitemp 212*, A-286	0.54 - 0.60
16 - 25 - 6	0.66
Steel, alloy/carbon, cast	1.19
Steel, stainless	
Austenitic (200,300 series)	0.41
Ferritic (400 series)	0.53 - 0.69
Martensitic (400 series)	0.63
Martensitic (500 series)	0.93
Cast	0.40 - 0.64

* or equal

Aluminum	
Wrought	
E.C.	5.93
1100	5.63
3003	4.88
5005	5.10
5052	3.51
2017	4.37
2024	4.80
6061	4.40
6063	4.88
6066 T6	3.69
7075 T6	3.08
Cast	
108	3.08
A108	3.60
220	2.24
356	4.04
Die Cast	
218	2.46
A380	2.55
43	3.69
Sand/PM Cast	
319	2.90
A356	4.04
Copper	
Wrought	
OFC	9.93
172 (beryllium copper)	2.73 - 3.30
220 (commercial bronze, 90%)	4.79
230 (red brass, 85%)	4.04
240 (low brass, 80%)	3.56
260 (cartridge brass, 70%)	3.08
314 (leaded commercial bronze)	4.57
330 - 360 (lead brasses)	2.94
505 phosphor bronze E	5.27
710 Cupronickel 10	1.14
Cast	
801, 803	9.93
814	6.59
Beryllium	3.82

Lead	0.86
Magnesium Alloys, Wrought	
AZ31 B-F*	1.93
AZ80A - T5*	1.27
ZK60A - T5*	3.00
HM21A -T8*	3.47
LA141A - T7*	1.11
Nickel Alloys	
200	0.33
210 (nickel)	1.50
610 (inconel*)	0.38
Gold	7.56
Silver	10.64
Platinum	1.85
Palladium	1.80
Rhodium	2.20
Indium	0.61
Tungsten	4.25
Molybenum	3.71
Tin	1.63
Zinc	2.73
Titanium	
Unalloyed	0.40 - 0.50
Alloys, general	0.20 - 0.30
Uranium, Depleted	0.67
Chromel	0.35
Constantin	0.60

*or equal

Plastics	
ABS	0.0044 - 0.0088
ALKYD (encapsulating putty)	0.01 - 0.02
Polycarbonate	0.004
Fluorocarbons	
PTFE	0.0062
FEP	0.0053
Epoxies	
Cast, rigid	0.0044 - 0.0132
Molded	0.0044 - 0.022
High-strength laminate	0.103
Melamine*, electrical grades	
Cellulose filer	0.007 - 0.009
Mineral filler	0.014 - 0.018
Nylons*	0.004 - 0.006
Phenolics	0.004 - 0.0088
Polyimides	0.01 - 0.025
Polyesters	
Cast	0.004 - 0.005
Glass fiber reinforced	0.050 - 0.060
Polyethylene	0.084
Silicone, glass-reinforced	0.079
Rubbers and Elastomers	
Foamed rubber	0.001
Molded, extruded	
Natural	0.080
Nitrile, GR-S	0.140
Neoprene*	0.110
Silicone*	0.130
Ceramics	
Standard electrical	0.038 - 0.069
Steatite*	0.064 - 0.085
Alumina	0.80 - 0.90
Beryllia	6.7 - 7.1
Mica	
Natural (compressed)	0.011 - 0.016
Synthetic	0.013 - 0.018

* or equal

<u>Glass</u>	
Pyrex*	0.03
Schott, Borosilicate	0.03
Silica	0.045
Soda	0.018
Soda Lime	0.026
<u>Quartz</u>	
Glass Cloth Composite Laminates	
G7, G9, G10, G11	0.075
G5	0.127
Graphite (Brush)	0.1 - 0.2
Vulcanized Fibre	0.0074
Thermistor Material	0.03 - 0.05
Concrete	0.017
Ice	0.056

* or equal

APPENDIX F

Table of Emissivity Values,

Notes:

1. Emissivities given are total normal emissivites at normal temperatures (Approx. 25-100°C)
2. Emissivities can change radically at extreme elevated temperatures as with tube filaments and anodes
3. Emissivity is strongly dependent on surface conditon. Surface roughness and oxide presence can cause wide variations in values
4. Emissivity values for coatings may vary with thickness.

Surface	Emissivity, e
Non-metallics	
Glass, smooth	0.94
Rubber	
Hard	0.95
Soft, gray	0.86
Water	0.96
Ice, smooth	0.97
Ice, crystalline	0.99
Paper, white	0.95 - 0.98
Plexiglas*	0.89
Lampblack	0.94
Carbon filament (1100°C)	0.53
Finishes	
Paints	
White enamel	0.91
Black, shiny	0.84 - 0.87
Black, flat	0.91 - 0.96
Oil paints, all colors	0.92 - 0.96
Aluminum 26%	0.30
Aluminum 10%	0.52
Oil Film	
Thicker than 5 mils	0.83
Less than 5 mils (function of thickness)	0.05 - 0.8
Silicone	0.79 - 0.88
Metals	
Aluminum	
Highly polished	0.04
Aged	0.05
Aged (salt atmosphere)	0.10
Sheet, commercial	0.045
Foil	.035 - .06
Rough plate	0.055

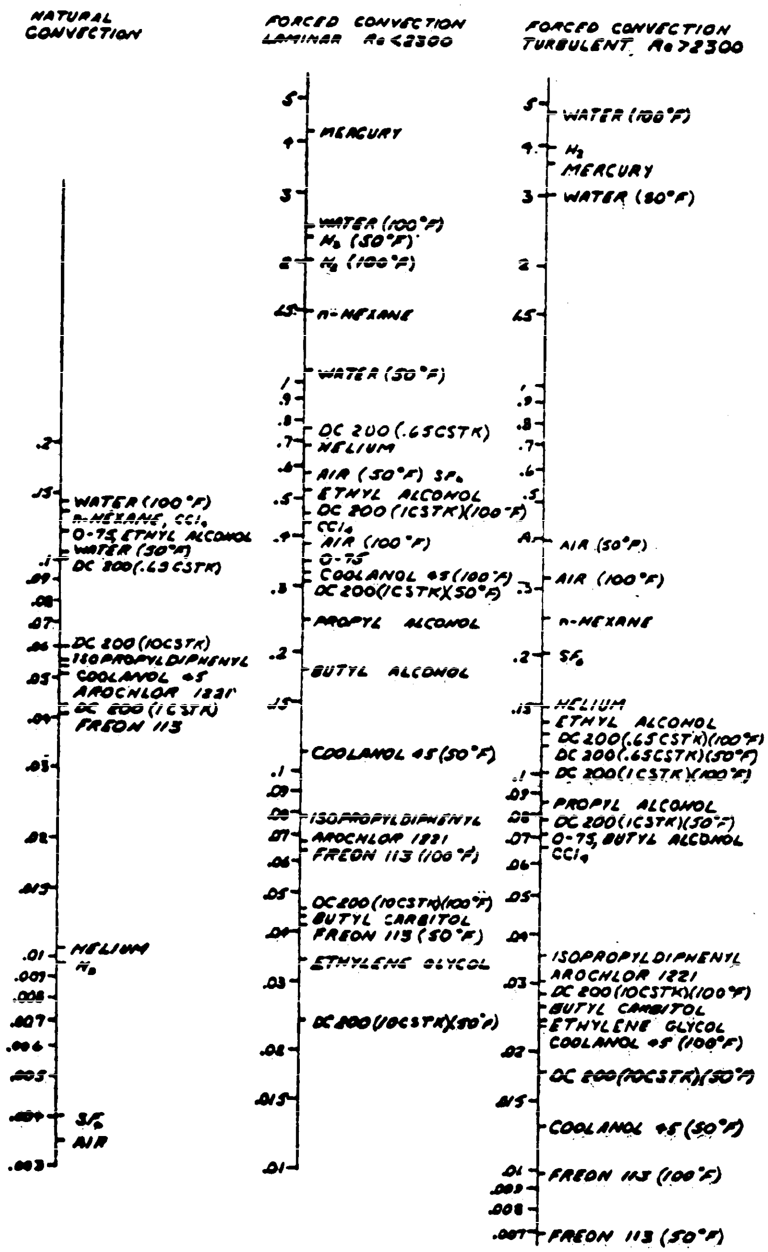
* or equal

Surface	Emissivity, e
Oxidized plate	0.11 - 0.2
Anodized	0.67 - 0.87
Alloys	
Alclad	0.03 - 0.05
3003	0.24
5053 (weathered)	0.74
2024 (weathered)	0.35
7075, anodized	0.61
Chromium, Polished	0.07 - 0.08
Copper	
Highly polished	0.02
Matte	0.22
Oxidized	0.26
Commercial	0.03
Heavily oxidized (black)	0.78
Brass	
Polished	0.03 - 0.04
Rolled	0.06
Dull	0.22
Heavily oxidized	0.61
Gold	0.02
Iron	
Cast, turned	0.44
Rolled steel sheet	0.66
Cast plate, smooth	0.80
Cast plate, rough	0.82
Cast, rough, heavy oxide	0.95
Steel	
Aluminized	0.20
Rolled, oxidized	0.66
Rough oxide	0.80 - 0.94
Polished	0.07 - 0.09
Stainless, commercial	0.31
Acid treated	0.57 - 0.66
Sand blasted	0.50
Lead, gray oxidized	0.28
Mercury	0.09
Magnesium alloys	0.59 - 0.70
Nickel plate, polished	0.045
Inconel*	0.25 - 0.35
Nichrome, oxidized	0.95
Rhodium, clean, polished	0.01 - 0.05
Silver, deposit	0.01
Tin, bright	0.43

* or equal

Surface	Emissivity, e
Tantalum filament, 1330°C	0.19
Tantalum filament, 2530°C	0.31
Wolfram (tungsten) filament, 3320°C	0.39

Appendix G. Coolant Effectiveness



COMPARATIVE VALUES OF COOLANT EFFECTIVENESS

APPENDIX H

Fluid Coolant Properties

F1 Air
F2 Water
F3 Silicone Fluids (Dow Corning)*
F4 Fluorochemicals (3M Co.)*
F5 Perfluorocarbon Liquids (DuPont)*
F6 "Freons" (Dupont)*
"TF"
"C-51-12"
116
"E" Series
F7 Typical Light Mineral Oil

* or equal

PROPERTIES OF AIR

Temp. °F.	Temp. °C.	** c_p Specific Heat $\frac{Btu}{lb.-°F}$	*** ρ Density $\frac{lb.}{cu. ft.}$	μ Viscosity $\frac{lb.}{ft.-hr.}$	k Thermal Conduct. $\frac{Btu}{hr.-ft.-°F}$	$\frac{c_p \mu}{k}$ Prandtl No.	β Coeff. of Vol. Expan. $\frac{1}{°R}$	*** $(a)10^{-6}$ Free Conv. Modulus $\frac{1}{cu. ft.-°F}$
-50	-46.0	0.239	0.0968	0.036	0.0116	0.74	0.00244	5.46
0	-17.8	0.239	0.0863	0.040	0.0132	0.72	0.00217	3.00
50	10.0	0.240	0.0779	0.043	0.0145	0.71	0.00196	1.81
100	37.8	0.240	0.0708	0.046	0.0158	0.70	0.00179	1.20
150	65.6	0.241	0.0651	0.049	0.0170	0.70	0.00164	0.82
200	93.3	0.241	0.0601	0.052	0.0182	0.69	0.00152	0.58
250	121.1	0.242	0.0559	0.055	0.0192	0.68	0.00141	0.41
300	148.9	0.242	0.0522	0.058	0.0204	0.68	0.00132	0.31
350	176.7	0.243	0.0490	0.060	0.0216	0.68	0.00123	0.23
400	204.4	0.245	0.0461	0.062	0.0227	0.67	0.00116	0.18
450	232.2	0.246	0.0436	0.065	0.0239	0.67	0.00110	0.14
500	260.0	0.247	0.0413	0.067	0.0250	0.66	0.00104	0.11
550	288.0	0.249	0.0393	0.070	0.0264	0.66	0.00099	0.086
600	315.6	0.250	0.0374	0.072	0.0271	0.66	0.00094	0.069
650	343.3	0.252	0.0358	0.074	0.0282	0.66	0.00090	0.055
700	371.1	0.253	0.0342	0.076	0.0291	0.66	0.00086	0.044

** Specific heat at constant pressure
*** Density and convection modulus for atmospheric pressure (29.92 in. Hg)

Properties of Water

Temp. °F	Temp. °C	c_p Btu. / (lb.)(°F)	μ* lb. / (ft.)(hr.)	k Btu. ft / (hr.)(ft.²)(°F)	Pr	$a \times 10^{-8}$ 1 / (ft.³)(°F)
32	0.0	1.009	4.33	0.327	13.4	- -
40	4.4	1.005	3.75	0.332	11.3	0.3
50	10.0	1.002	3.17	0.338	9.4	1.0
60	15.6	1.000	2.71	0.344	7.9	1.7
70	21.1	0.998	2.37	0.349	6.8	2.3
80	26.7	0.998	2.08	0.355	5.8	3.0
90	32.2	0.997	1.85	0.360	5.1	3.9
100	37.8	0.997	1.65	0.364	4.5	5.2
110	43.3	0.997	1.49	0.368	4.0	6.6
120	48.9	0.997	1.36	0.372	3.6	7.7
130	54.4	0.998	1.24	0.375	3.3	8.9
140	60.0	0.998	1.14	0.378	3.0	10.2
150	65.6	0.999	1.04	0.381	2.7	12.0
160	71.1	1.000	0.97	0.384	2.5	13.9
170	76.7	1.001	0.90	0.386	2.3	15.5
180	82.2	1.002	0.84	0.389	2.2	17.1
190	87.8	1.003	0.79	0.390	2.1	- -
200	93.3	1.004	0.74	0.392	1.9	- -

*lb. of mass.

Properties of Silicone Fluid - (DC550-112 Centistoke Grade)*

Temp. °F	Temp. °C	c_p Btu. / (lb.)(°F)	μ lb. / (ft.)(hr.)	k Btu. ft. / (hr.)(ft.²)(°F)	ρ lb. / ft.³	a x 10^{-8} 1 / (ft.³)(°F)
60	15.6	0.378		0.0783	67.0	- -
80	26.7	0.386		0.0778	66.4	- -
100	32.2	0.395		0.0773	65.9	- -
120	37.8	0.405	132	0.0767	64.3	0.32
140	60.0	0.414	98.0	0.0761	64.8	0.43
160	71.1	0.425	74.0	0.0755	64.2	0.59
180	82.2	0.437	57.0	0.0749	63.7	0.78
200	93.3	0.452	44.0	0.0742	63.2	0.99
220	104.4	0.470	36.0	0.0736	62.7	1.22
240	115.6	0.487	30.0	0.0729	62.2	1.45
260	126.7	0.501	25.5	0.0721	61.7	1.69
280	137.8	0.514	22.1	0.0714	61.2	1.94
300	148.9	0.523	19.6	0.0707	60.8	2.19
320	160.0	0.531	17.3	0.0700	60.3	2.45
340	171.1	0.538	15.9	0.0692	59.8	2.72
360	182.2	0.544	14.3	0.0685	59.4	2.99

Flash point min.	300°C
Freezing point	-50°C
Coefficient of expansion, k x 1000/°C (25 to 100°C)	.75
Dielectric strength	800 volts/mil
Power factor at 25°C	
10^2 cps	.00158
10^6 cps	.00003
Dielectric constant at 25°C at 10^2 and 10^6 cps	2.9
Color	slightly yellow

Courtesy, Dow Corning Corp.

*or equal

Properties of DC-701 Silicone Fluid *

Color	Clear water white
Viscosity at 25°C	10 centistokes
Freezing point	-60°C
Boiling point at atmospheric pressure	340° C
Flash point, min.	143°C
Specific gravity at 25°C	1.04
Coefficient of expansion per 1000/°C, 0°-100°C	.93
Dielectric strength, volts per mil	500
Dielectric constant at 25°C and 10^2 and 10^5 cps	2.63
Power factor at 25°C	
at 10^2 cps	0002
at 10^5 cps	.0001

Courtesy - Dow Corning Corp.

* or equal

PHYSICAL PROPERTIES OF DOW CORNING 200 FLUIDS *

Viscosity in Cstks. at 25°C	Viscosity in SSU at 100°F	Viscosity Temperature Coefficient	Dielectric Constant	Freezing Point °C	Freezing Point °F	Boiling Point Temperature °C	Boiling Point Temperature °F	Boiling Point Press mm Hg	Flash Point Minimum °F	Thermal Conductivity	Specific Gravity 25°C/25°C
0.65	28	0.31	2.18	-68	-90	99.5	211	760	30	.00024	0.761
1.0	28	0.37	2.32	-86	-123	152	305	760	110	.00024	0.818
1.5	30	0.46	2.40	-76	-105	192	377	760	160	.00025	0.853
2.0	31	0.48	2.45	-84	-119	230	446	760	175	.00026	0.873
				Pour Point							
3.0	33	0.51	2.52	-65	-85	70-100	158-212	0.5	215	.00027	0.900
5.0	38	0.55	2.58	-65	-85	120-160	248-320	0.5	275	.00028	0.920
10	52	0.57	2.65	-65	-85	> 200	> 392	0.5	325	.00032	0.940
20	80	0.59	2.68	-60	-76	> 200	> 392	0.5	520	.00034	0.955
50	185	0.59	2.72	-55	-67	> 250	> 482	0.5	535	.00036	0.960
						Volatility after 48 hrs. at °C	at °F	%			
100	350	0.60	2.74	-55	-67	200	392	< 2	575	.00037	0.970
200	720	0.62	2.74	-53	-63	200	392	< 2	600	.00037	0.971
350	1,250	0.62	2.75	-50	-58	200	392	< 2	600	.00038	0.972
500	1,750	0.62	2.75	-50	-58	200	392	< 2	600	.00038	0.972
1,000	3,500	0.62	2.76	-50	-58	200	392	< 2	600	.00038	0.973
				Solidification Temperature							
12,500	45,000	0.58	2.82	-46	-51	200	392	< 2	600	.00038	0.974
30,000	115,000	0.61	2.77	-44	-47	200	392	< 2	600		0.975

* or equal

Properties of Fluorochemical 0-75 *

Formula	$C_8F_{16}O$
Physical state at room temperature	Colorless liquid
Odor	Odorless
Formula weight	416
Boiling point	101°C (214°F)
Freezing point (Glass point)	-113°C (-171°F)
Pour point	-100°C (-148°F)
Density (g/cc at 25°C, 77°F)	1.760
Viscosity (centistokes at 25°C, 77°F)	0.82
Refractive index (25°C, 77°F)	1.276
Surface tension (dynes/cm at 25°C, 77°F)	15.2
Coefficient of volume expansion (per °C) (25-40°C, 77-104°F)	1.6×10^{-3}
(40-80°C, 104-176°F)	2.0×10^{-3}
Specific heat (cal/g/°C at 25-40°C, 77-104°F)	0.26
Heat of vaporization (cal/mole at b.p.)	8,700
(cal/g)	20.9
Thermal conductivity. Liquid (Btu/hr./sq.ft./°F.ft.)(25°C, 77°F)	0.071
(60°C, 140°F)	0.054
Dielectric strength (ASTM-D877)	37 KV
Dielectric constant (100 cps at 25°C, 77°F)	1.85
Power factor (100 cps at 25°C, 77°F)	<0.0005
Volume resistivity (ohm-cm at 25°C, 77°F)	10^{14}-10^{16}

Courtesy, Minnesota Mining & Mfg. Co.

* or equal

Properties of Fluorochemical N-43*

Formula	$(C_4F_9)_3N$
Physical state at room temperature	Colorless liquid
Odor	Odorless
Formula weight	671
Boiling point	177°C (351°F)
Freezing point (Glass Point)	-66°C (-87°F)
Pour point	-50°C (-58°F)
Density (g/cc at 25°C, 77°F)	1.872
Viscosity (Centistokes at 25°C, 77°F)	2.74
Refractive index (25°C, 77°F)	1.2910
Surface tension (dynes/cm at 25°C, 77°F)	16.1
Coefficient of volume expansion (per °C) (25-40°C, 77-104°F)	1.2×10^{-3}
(140-160°C, 284-320°F)	2.1×10^{-3}
Specific heat (cal/g/°C at 25-40°C, 77°- 104°F)	0.27
Heat of vaporization (cal/mole at b.p.)	11,100
(cal/g, Btu/lb.)	16.5
Vapor pressure (mm Hg at 25°C, 77°F)	0.3
Trouton ratio	24.6
Dielectric strength (ASTM-D 877)	40 KV
Dielectric constant (100 cps at 25°C, 77°F)	1.86
Power factor (100 cps at 25°C, 77°F)	<0.0005
Volume-resistivity (ohm-cm at 25°C, 77°F)	10^{14} - 10^{16}

Courtesy, Minnesota Mining & Mfg. Co.

* or equal

TYPICAL PROPERTIES

The following data indicate typical characteristics of these fluids but are not to be used as purchase specifications.

	FC-78 **	FC-77 **	FC-75 **	FC-43 **
Nominal Boiling Point, °F	122	207	216	345
Pour Point, °F	-100	-150	-135	-58
Density at 77°F, lbs/ft^3	106	111	110	117
Density at -65°F, lbs/ft^3	119	123	122	---
Kinematic Viscosity at 77°F, cs.	0.44	0.80	0.82	2.6
Kinematic Viscosity at -65°F, cs.	1.98	6.90	7.40	---
Vapor Pressure at 77°F, mm Hg.	260	42	30	0.3
Specific Heat at 77°F, Btu/lb-°F	0.24	0.25	0.25	0.27
Heat of Vaporization at the Boiling Point, Btu/lb	41	36	38	30
Thermal Conductivity at 77°F, $Btu/hr\text{-}ft^2\text{-}°F/ft.$	0.036	0.037	0.037	0.039
Coefficient of Expansion, $ft^3/ft^3\text{-}°F$	0.0009	0.0009	0.0009	0.0008
Surface Tension at 77°F, dynes/cm	13	15	15	16
Critical Temperature, °F	383*	484*	441	578*
Critical Pressure, psia	297*	218*	232	207*
Refractive Index at 77°F	1.267	1.280	1.276	1.291
Dielectric Strength at 77°F, volts/mil	430	450	550	560
Dielectric Constant at 77°F and 1 KC	1.81	1.86	1.86	1.90
Dissipation Factor at 77°F and 1KC	<.0005	<.0005	<.0005	<.0005
Dissipation Factor at 77°F and 3KMC	0.0023	0.0074	0.0065	0.0062

*Estimated values ** or equal

Courtesy 3M Co.

TYPICAL PROPERTIES OF FC-88 **

Nominal Boiling Point	88°F. (31°C.)
Average Molecular Weight	~300
Vapor Pressure at 77°F	610 mm Hg. (11.8 psia)
Density at 77°F	102 lbs./ft.^{3} (1.64 gms/cc.)
Viscosity at 77°F	0.4 cs.
Pour Point	-150°F (-101°C)
Heat of Vaporization at the Boiling Point	37.8 Btu/lb. (21 cal./gm.)
Thermal Conductivity at 77°F	0.032 Btu/hr./ft.^{2}/°F/ft. (0.00056 watts/cm.^{2}/°C/cm.)
Heat Capacity at 77°F	0.25 Btu/lb.-°F. (0.25 cal./gm/-°C.)
Critical Temperature	293°F. (145°C.)
Critical Pressure	294* psia (20* atmos. absolute)
Coefficient of Expansion	0.0009 ft.^{3}/ft.^{3}-°F (0.0016 ft.^{3}/ft.^{3}-°C)
Refractive Index at 77°F (Sodium D line)	1.238
Surface Tension at 77°F	13* dynes/cm.
Vapor Pressure	$\text{Log P mm Hg} = 7.9768 - \frac{1548}{T(^{\circ}K)}$

*Estimated values

Courtesy 3M Co.

* or equal

TYPICAL ELECTRICAL PROPERTIES OF FC-88*

Dielectric Strength at 77°F	42 kv/0.1 in (420 v/mil)
Dielectric Constant at 77°F	
@ 1 KC	1.72
@ 1 KMC	1.77
@ 3 KMC	1.73
@ 8.5 KMC	1.75
Dissipation Factor at 77°F	
@ 1 KC	< 0.0003
@ 1 KMC	0.0007
@ 3 KMC	0.0013
@ 8.5 KMC	0.0035

Courtesy 3M Co. * or equal

Typical Solubility Properties of Perfluorocarbon Liquids*
(Wt. % at 27°C)[1]

SOLVENT	FCX-326 (C_7F_{14})	FCX-327 (C_8F_{16})	FCX-328 ($C_8F_{15}Cl$)
$CHCl_3$	7	10	Miscible
CCl_4	Miscible	43[2]	Miscible
CH_3OH	Insoluble	2.5	3.8
Ethyl acetate	15	13	Miscible
Acetone	10	9	42
Petroleum ether	Miscible	Miscible	Miscible
Ethyl ether	Miscible	Miscible	Miscible
Benzene	3	4.6	21
O-dichloro-benzene	Insoluble	Insoluble	Insoluble

(1) Solubilities of less than 1.0% are reported as "insoluble".

(2) (At which point solution separated into two substantially equal phases.)

Courtesy, duPont

*or equal

Typical Properties of Perfluorocarbon Liquids

Identification Number	FCX-326	FCX-327	FCX-328
Chemical Formula	C_7F_{14}	C_8F_{16}	$C_8F_{15}Cl$
Molecular Weight	350	400	416.5
Boiling Point (°C)	76	102	129
Index of Refraction	1.2762 (30°C)	1.2858 (30°C)	11.3170 (25°C)
Specific Gravity	1.7999 (20°C)	1.853 (20°C)	1.869 (20°C)
Dielectric Constant (1)	1.69-1.70	1.75	2.03
Power Factor (1)	0.0045-0.0005	0.0098-0.0010	0.0110-0.0008
Dielectric Strength (Volts) (2)	16,600	15,000	12,900
Direct Current Resistance (ohms/cm.^{3})	5.2×10^{12}	1.2×10^{12}	1.6×10^{12}

The following are the characteristics of Standard Transformer Oil:

Dielectric constant = 2.00 Power factor = 0.0053 - 0.0004
Resistivity = 1.1×10^{12} Dielectric strength = 14,400

(1) From 100 cycles to 100 kc.
(2) Method ASTM D 117-43, modified using 0.054-in. gap.

Courtesy, duPont

Freon TF Dielectric Liquid*

PHYSICAL PROPERTIES

Chemical Formula		CCl_2FCClF_2
Molecular Weight		187.4
Boiling Point at One Atmosphere,	°F	117.6
	°C	47.6
Freezing Point,	°F	−31
	°C	−35
Critical Temperature,	°F	417.4
	°C	214.1
Critical Pressure,	psia	495
	atm	33.7
Density at 77°F (25°C)		
Liquid,	lbs/gal.	13.06
	lbs/ft³	97.69
	grams/cm³	1.565
Sat'd Vapor at boiling point,	lbs/ft³	0.4619
	grams/liter	7.399
Latent Heat of Vaporization at b.p.,	Btu/lb	63.12
	cal/gram	35.07
Specific Heat at 70°F (21.1°C), Btu/(lb)(°F) or cal/(gram) (°C)		
Liquid		0.213
Sat'd Vapor (C_p)		0.152
Thermal Conductivity at 70°F (21.1°C), Btu/(hr) (ft²) (°F/ft)		
Liquid		0.043
Sat'd Vapor		0.00430
Viscosity at 70°F (21.1°C), Centipoises		
Liquid		0.694
Sat'd Vapor		0.0102
Refractive Index of Liquid at 79.7°F (26.5°C)		1.355
Surface Tension at 77°F (25°C), dynes/cm		17.3

Courtesy Du Pont.

* or equal

HEAT-TRANSFER PROPERTIES OF LIQUID "FREON" C-51-12 *

Density, g/cc, 77°F lb/gal, 77°F	1.6718 13.9512
Heat of Vaporization at B.P., cal/g Btu/lb	21.4* 38.5*
Specific Heat, C_p, Btu/(lb)(°F), 77°F	0.27
Thermal Conductivity, Btu/(hr)(ft)(°F), 77°F	0.0343
Coefficient of Cubical Expansion, -20 to 80°C, 1/°C - 5 to 180°F, 1/°F	 2.116×10^{-3} 1.176×10^{-3}
Thermal Expansion, ft^3/(lb)(°F), 77°F	0.000007
Vapor Pressure, 77°F, psia 130°F, psia	7.3 20.5
Viscosity, 77°F, Centipoise Centistoke	0.98 0.59

* Estimated values

DIELECTRIC PROPERTIES OF "FREON" C-51-12 *

Breakdown Voltage, liquid, KV (rms)[1]	42
Breakdown Voltage, vapor, KV (rms)[2]	32 (1 atm, 44°C) 55 (2 atm, 66°C)
Resistivity[3], ohm/cm	$>4 \times 10^{14}$
Dielectric Constant, κ, @ 10^2-10^5Hz @ 3 GHz	1.85 1.85
Dissipation Factor, tan δ @ 10^2-10^5Hz @ 3 GHz	<0.00006 0.004

1 ASTM D 877, 0.1" gap between planes, one atm, 25°C
2 Modification of ASTM D 2477, 0.1" gap, sphere to plane
3 ASTM D 257

Courtesy DuPont. * or equal

PROPERTIES OF DIELECTRIC GASES

	"Freon" * 116	Sulfur Hexafluoride
Chemical Formula	C_2F_6	SF_6
Molecular Weight	138.0	146.1
Boiling Point, °C	−78.2	−63.7*
Freezing Point, °C	−101	—
Critical Temperature, °C	19.7	45.6
Critical Pressure, psia	432	546
Liquid Density, lb/cu ft		
−40°C	87.63	111.6
15°C	56.72	88.65
Vapor Density at 100 psia and 26.7°C, lb/cu ft	2.50	3.2
Heat Capacity of the Vapor at Constant Volume, cal/(gram) (°C)		
10°C	0.16	0.14
65.6°C	0.18	0.16
Latent Heat, cal/gram		
Fusion	4.65	8.22
Vaporization @ B.Pt.	27.97	30.80
Thermal Conductivity at 1 Atm., PCU/(hr) (ft) (°C)		
−40°C	0.0058	0.0063
25°C	0.0087	0.0080
Relative Dielectric Strength (N_2 = 1) at one atmosphere and 25°C with a 1" gap (ASTM D-2477)	2.0	2.4

*Sublimation point.

Courtesy DuPont. *or equal

PROPERTIES OF "FREON" E SERIES FLUOROCARBONS

Property	"Freon" E 1	"Freon" E 2	"Freon" E 3	"Freon" E 4	"Freon" E 5
Formula: $F(CFCF_2O)_nCHFCF_3$ (with CF_3 branch)	n = 1	n = 2	n = 3	n = 4	n = 5
Molecular Weight	286.03	452.08	618.12	784.15	950.18
Boiling Point, °C	40.8	104.4	152.3	193.8	224.2
Boiling Point, °F	105.4	220.0	306.1	380.8	435.6
Approx. Pour Point (200,000 cs) °C	-154.4	-123.5	-106.7	- 94.4	- 83.9
Approx. Pour Point (200,000 cs) °F	-246.0	-190.0	-160.0	-138.0	-119.0
Compressibility, % @ 77°F, 500 atm.	8.20	6.48	5.64	5.18	4.85
Crit. Temp., °C	157.55	218.28	263.17	295.19	322*
Crit. Temp., °F	315.59	424.90	505.71	563.34	611.6*
Crit. Pres., Psia	283.6	197.9	157.7	122.1	112*
Crit. Pres., Atm.	19.29	13.46	10.73	8.31	7.6*
Heat of Vaporization @ B.P., cal/g	23.0*	17.4*	14.5*	12.5*	11.0*
Heat of Vaporization @ B.P., Btu/lb	41.4*	31.3*	26.1*	22.5*	19.9*
Refractive Index Liquid, 80°F	1.2434	1.2570	1.2654	1.2704	1.2724
Solubility in Water, 77°F, ppm	<25	<25	<25	<25	<25
Solubility of Water, 77°F, ppm	149	103	83	68	
Solubility of Air in Liquid, 77°F at 1 Atm, Total Pressure, cc air/100 cc liq.	28.0	39.8	36.4	26.6	22.5
Surface Tension, 77°F, dynes/cm.	10.4	12.9	14.2	15.2	15.9

* Estimated Values ** by equ.

Courtesy DuPont

(Continued on next page)

HEAT TRANSFER PROPERTIES OF LIQUID "FREON" E FLUOROCARBONS ***
(@ 77°F)

Density, g/cc lb/gal	1.538 12.835	1.653 13.836	1.723 14.378	1.763 14.712	1.792 14.954
Specific Heat, Cp., Btu/(lb)(°F)	0.247*	0.242*	0.240	0.239*	0.238*
Thermal Cond. Btu/(hr)(ft)(°F)	0.0364	0.0372*	0.0373	0.0333*	0.0337
Thermal Expansion cc/(gram)(°C) cuft/(lb)(°C) cuft/(lb)(°F)	 1.3×10^{-3} 2.0×10^{-5} 1.2×10^{-5}	 8.9×10^{-4} 1.4×10^{-5} 7.9×10^{-6}	 7.7×10^{-4} 1.2×10^{-5} 6.8×10^{-6}	 7.0×10^{-4} 1.1×10^{-5} 6.2×10^{-6}	 6.5×10^{-4}* 1.0×10^{-5}* 5.8×10^{-6}*
Vapor Pres., psia	7.8	0.6**	0.11**	0.016**	0.003**
Viscosity, cp. cs.	0.5 0.3	1.1 0.6	2.2 1.3	4.1 2.3	7.0 3.9

* Estimated   ** Extrapolated

DIELECTRIC PROPERTIES OF "FREON" E SERIES LIQUIDS ✗

Breakdown Voltage Liquid, KV (rms)[1]	29	34	41	45	49
Breakdown Voltage Vapor, KV(rms)[2] @ 1 Atm @ 2 Atm	 22 38	 27 43	 35 52	 - -	 - -
Resistivity[3], ohm-cm	-	$>4\times10^{14}$	$>4\times10^{14}$	$>4\times10^{14}$	$>4\times10^{14}$
Dielec. Constant ε	3.02	2.75	2.58	2.50	2.45
Dissipation Factor, Tan δ, 100 Hz-100kHz	<0.00006	<0.00006	<0.00006	<0.00006	<0.00006

1 ASTM D 877, 0.1" gap between planes, 1 atm., 25°C
2 Modification of ASTM E 2477, 0.1" gap, sphere to plane
3 ASTM D 257

Courtesy D. Pont   *** or equal

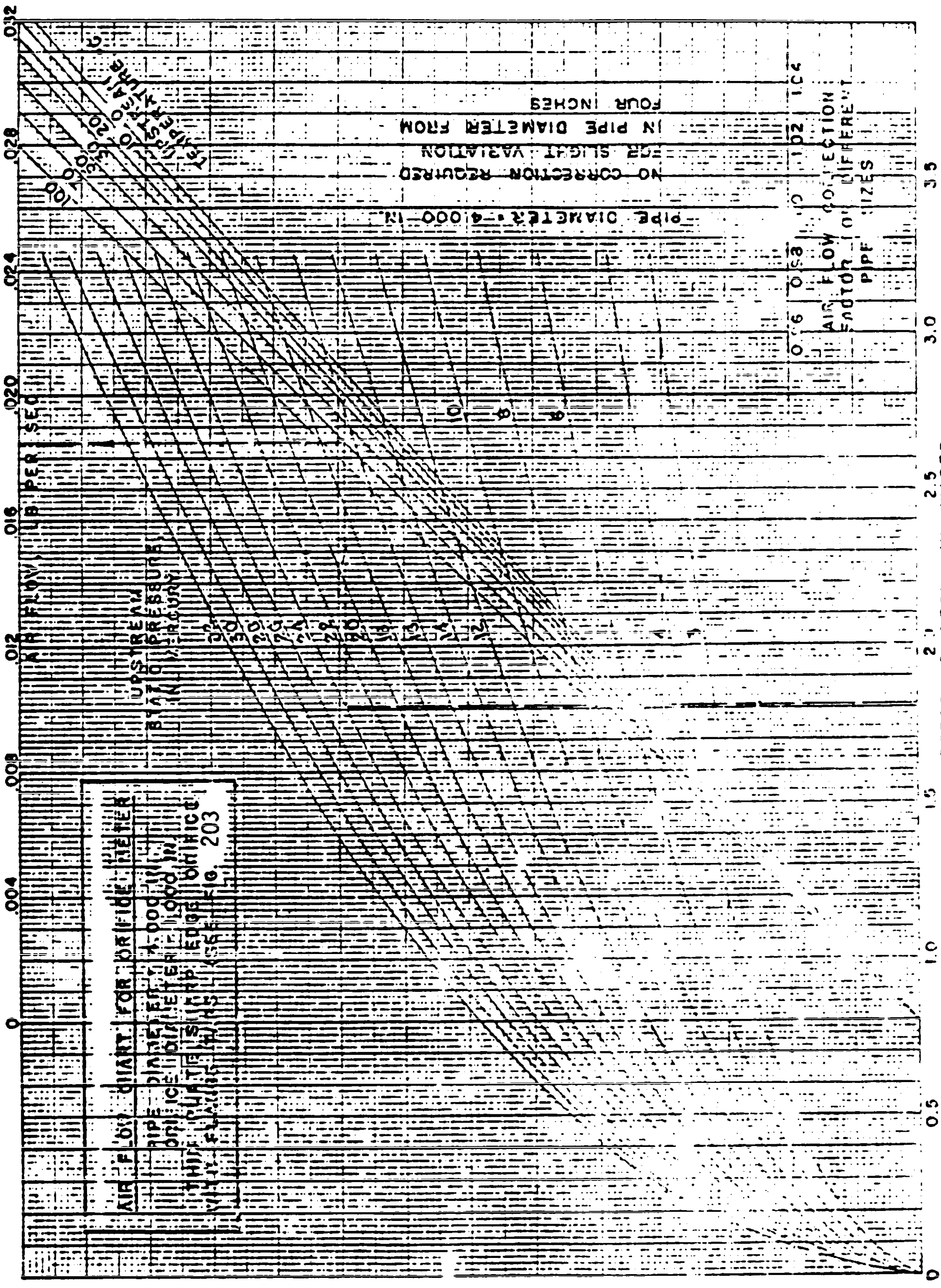
Properties of A Typical Light Mineral Oil

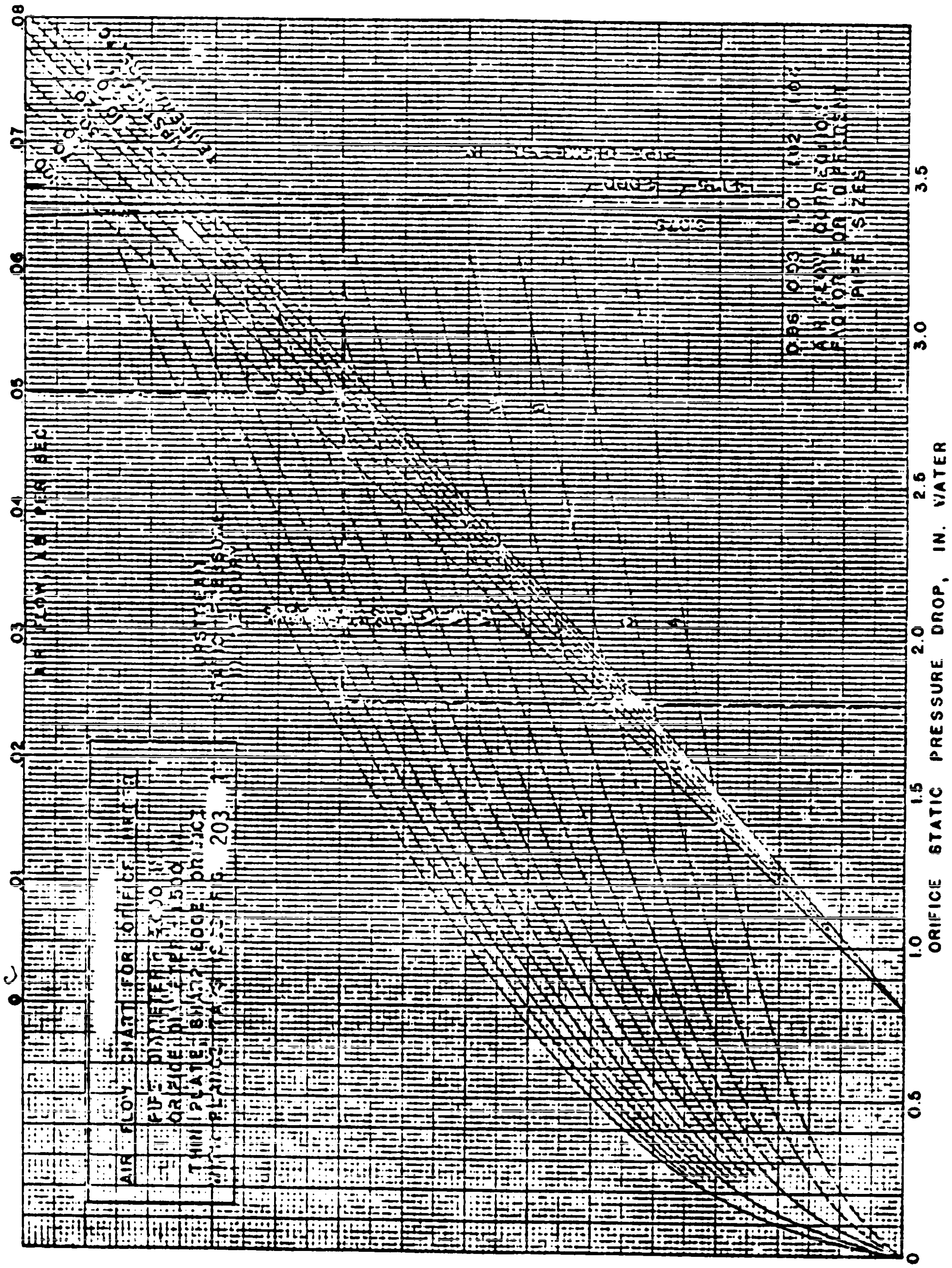
Temp. °F	Temp. °C	c_p Btu./(lb.)(°F)	μ lb./(ft.)(hr.)	k Btu. ft./(hr.)(ft^2)(°F)	a x 10^{-8} 1/(ft.^{3})(°F)
50	10	0.43	315.	0.0770	0.10
60	16	0.43	210.	0.0768	0.14
70	21	0.44	140.	0.0766	0.22
100	38	0.45	55.	0.0763	0.58
150	65	0.48	19.	0.0756	1.76
200	93	0.50	9.	0.0749	3.92
250	121	0.52	5.	0.0743	7.22
300	149	0.54	3.	0.0736	12.42

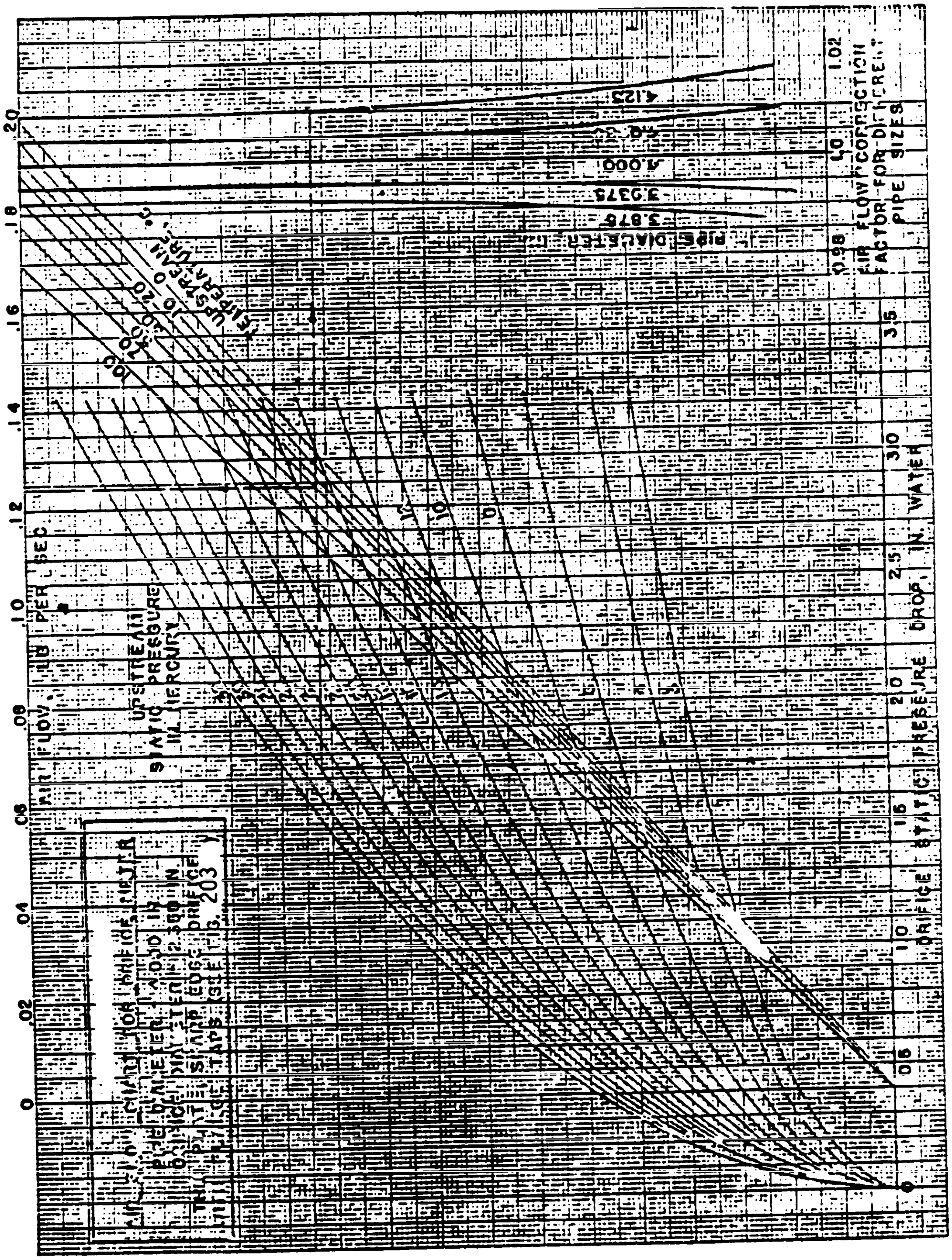
Appendix I

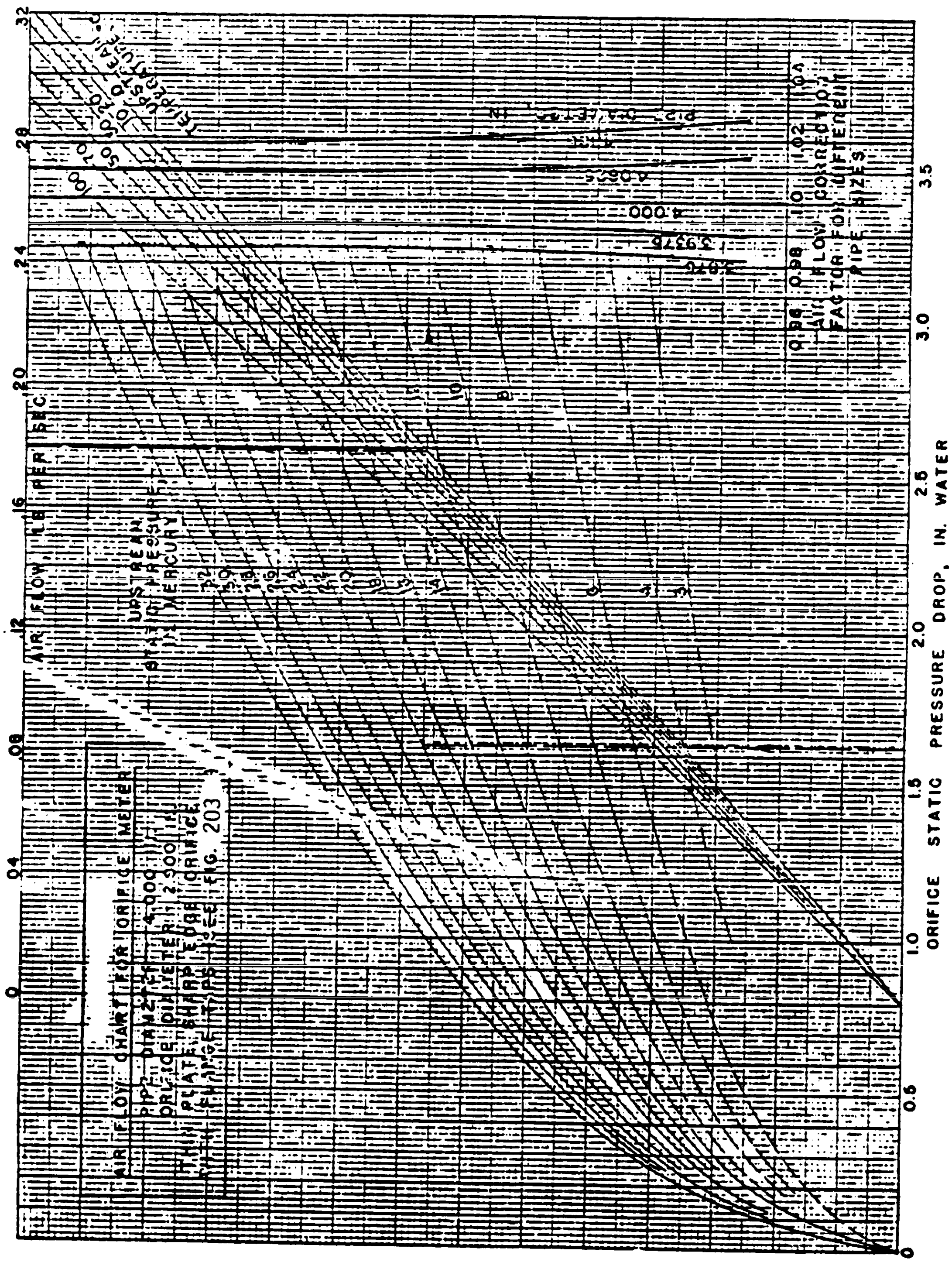
Orifice Calibration Curves

(See Figure 13-20)


AIR FLOW CHART FOR ORIFICE METER
PIPE DIAMETER 4.000 IN.
ORIFICE DIAMETER 1.000 IN.
THIN SQUARE EDGE ORIFICE
WITH FLANGE TAPS (SEE FIG. 203)
AIR FLOW, LB PER SEC
ORIFICE STATIC PRESS DROP, IN. WATER
UPSTREAM STATIC PRESSURE, IN. MERCURY
UPSTREAM AIR TEMP DEG F
PIPE DIAMETER = 4.000 IN.
NO CORRECTION REQUIRED
FOR SLIGHT VARIATION
IN PIPE DIAMETER FROM
FOUR INCHES
AIR FLOW CORRECTION FACTOR ON DIFFERENT PIPE SIZES


AIR FLOW, LB PER SEC
0
.01
.02
.03
.04
.05
.06
.07
.08
ORIFICE STATIC PRESSURE DROP, IN. WATER
0
0.5
1.0
1.5
2.0
2.5
3.0
3.5
AIR FLOW CHART FOR ORIFICE
203


AIR FLOW, LB PER SEC
0
.02
.04
.06
.08
.10
.12
.14
.16
.18
.20
ORIFICE STATIC PRESSURE DROP, IN. WATER
0
0.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
UPSTREAM STATIC PRESSURE IN. MERCURY
UPSTREAM TEMPERATURE
100
40
0
-20
PIPE DIAMETER
4.125
4.000
3.9375
3.875
AIR FLOW CORRECTION FACTOR FOR DIFFERENT PIPE SIZES
0.98
1.0
1.02
203


AIR FLOW CHART FOR ORIFICE METER
PIPE DIAMETER 4.000 IN.
ORIFICE DIAMETER 2.900 IN.
THIN PLATE SHARP EDGE ORIFICE
WITH FLANGE TAPS (SEE FIG. 203)
AIR FLOW, LB PER SEC
UPSTREAM STATIC PRESSURE IN. MERCURY
ORIFICE STATIC PRESSURE DROP, IN. WATER
AIR FLOW CORRECTION FACTOR FOR DIFFERENT PIPE SIZES

Appendix J

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