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DESIGN MANUAL
OF
NATURAL METHODS OF COOLING ELECTRONIC EQUIPMENT

DEPARTMENT OF THE NAVY
BUREAU OF SHIPS

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DESIGN MANUAL OF NATURAL METHODS
OF COOLING ELECTRONIC EQUIPMENT

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This publication, "Design Manual of Natural Methods of Cooling Electronic Equipments", NAVSHIPS 900,192 (C.A.L. Inc. report HF-845-D-8) is the fourth in a series of publications prepared for the Bureau of Ships by Cornell Aeronautical Laboratory Incorporated. This series of publications is intended to guide designers of electronic equipment in the use of convection, conduction and radiation heat transfer methods.

The other publications in the series are: "Survey Report of the State of the Art of Heat Transfer in Miniaturized Electronic Equipment", NAVSHIPS 900,189 (C.A.L. Inc. report HF-710-D-10); "Manual of Standard Temperature Measuring Techniques, Units and Terminology for Miniaturized Electronic Equipment," NAVSHIPS 900,187 (C.A.L. Inc. report HF-845-D-2); and "Guide Manual of Cooling Methods for Electronic Equipment", NAVSHIPS 900,190 (C.A.L. Inc. report HF-710-D-16).

These publications are written for electronics personnel who are not well versed in thermodynamics. Information contained therein may be used in part or entirety in the preparation of other government publications.

Errors found in this publication (other than obvious typographical errors) should be reported to the Electronic Publications Section of the Bureau of Ships.

All Navy requests for copies of these publications should be directed to the nearest Bureau of Supplies and Accounts Forms and Publications Supply Point. Other requests should be directed to the Government Printing Office.

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II INTRODUCTION

The demand for military electronic equipment having improved performance, decreased size, greater reliability and complexity of function will not wane in the foreseeable future. Consequently, electronic design problems are increasing and the desired reliability can only be achieved if the equipment is designed with reliability as a primary objective. Much has been written and said regarding the need for reliable electronic equipment, but, it has only recently become apparent that reliability is more closely related to the adequacy of cooling than to any other single factor. Unless adequate heat removal is provided in future military electronic equipment, the resultant reliability will be unsatisfactory.

Reliability can only be achieved if the electronic, thermal, and mechanical designs are all well executed. This result can best be accomplished by the equipment designers, who must control all pertinent factors. The thermal design is fully as important as the circuit design. Unless effective heat removal is provided, electronic parts will become too hot and malfunctioning and failures will follow. Thus, the science of heat transfer must be employed in electronic design.

This Manual is the fourth report of a series related to the cooling of electronic equipment. It has been prepared to assist electronic engineers in the thermal design of equipment cooled by natural means. It does not supercede NAVSHIPS 900,190 in whole or in part. Only material necessary for continuity has been duplicated between the two Manuals. In most cases methods of computation have been simplified by the use of nomographic charts. It has been deliberately written at a technical level such that engineers without heat transfer background can design acceptable equipment. Therefore, this Manual outlines procedures to temper the designer's judgment and lead to the best practical solution. Design formulae contained herein are usually simplified approximations. Frequently, qualitative, rather than quantitative, information is presented. When specific recommendations are given, care must be taken in adopting such recommendations so that they do not conflict with the requirements of the contracting activity.

This Manual emphasizes ground-based and shipboard electronic equipment. No consideration has been given to non-steady-state heat transfer or operation at environmental air pressures significantly less than one atmosphere. Further, for the purposes of the Manual, it has been assumed that unitized construction will be generally used, the equipment being composed of assemblies.

The thermal design of electronic equipment is a relatively new science. Obsolete, incomplete, scarce, and unreliable data hamper the development of heat transfer as an exact science and technology. Methods of accurately predicting the thermal performance of electronic equipment are not commonly known, and most organizations that have produced successful designs have achieved their goals by techniques peculiar to a specific equipment design. Not enough engineering groups have been assigned the time or money to ferret out the underlying principles governing details of heat flow in electronic parts and assemblies. Accordingly, the U. S. Navy, Bureau of Ships, has sponsored this work to aid in the improvement of electronic reliability. Much remains to be accomplished in this relatively new field. There is a need for the standardization of cooling means and systems. If the operating temperatures of electronic parts and materials can be increased, some of the current thermal problems will be alleviated. In addition, techniques should be developed to obtain an analytical method of determining the rating and limits of each equipment type under given environmental conditions.

We wish to mention particularly the productive cooperation we have received during this program from government agencies, universities and industries. References of source materials are listed in the bibliography, together with reference numbers at pertinent locations in the discussion. The terminology used herein is presented in Cornell Aeronautical Laboratory Report No. HF-845-D-2. Symbols are listed in Appendix A. Design equations are identified by (D.E.).

Attention is called to NAVSHIPS 900, 189 of 3 March 1952, an earlier report in this series. Section III B of this report contained information regarding the upper temperature limits of electronic parts. Progress in the state of the art since March 1952, coupled with part reratings has made some of this information obsolete. It is therefore requested that section III B of NAVSHIPS 900, 189 be disregarded.

Work on heat transfer in electronic equipment is continuing at this Laboratory. Future publications will supplement this Manual. Appendix C consists of a summary of reports already written and also planned under this program. Detailed information in these matters can be obtained from Mr. Rodney Hall, Code 818H, Bureau of Ships, Washington 25, D.C. Comments are solicited.

This Manual is not to be construed as an endorsement of any commercial products mentioned.

III THE FUNDAMENTALS OF NATURAL METHODS OF HEAT TRANSFER

A. GENERAL

Free convection, conduction, and radiation are the most common means of heat rejection within and from electronic equipment. Natural methods are those wherein heat transfer occurs without additional energy being supplied to accelerate the process. The majority of electronic equipment and parts have been designed for natural cooling in a free air environment at atmospheric pressure. 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 forced air and circulating liquids for internal cooling.

Heat or thermal energy is transferred from one region to another by virtue of temperature difference. The two fundamental axioms are that heat flows only 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.

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 unsteady state. An example of the latter is the warm-up period of an electronic assembly. This Manual does not consider such transient conditions, but only those occurring after thermal equilibrium has been reached.

In general, there are three modes or methods of heat transfer: conduction, convection, and radiation. They may occur singly or simultaneously. While evaporation and condensation may be classified under convection, they are usually considered separately, since change of state occurs as well as heat transfer.

B. 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 the steady state and in its most simple form is:

$$q = \frac{k A \Delta t}{L} \qquad (1)$$

where:

- q is the rate of heat transfer
- k is the thermal conductivity of the material
- A is the cross-sectional area perpendicular to the direction of heat flow
- L is the length of heat flow path
- Δt is the temperature difference causing the heat flow

Heat flow by conduction is analogous to current flow and bears a relation like Ohm's Law. Rewriting equation (1) as:

$$q = \frac{\Delta t}{\frac{L}{kA}} \qquad (2)$$

it can be seen that q is analogous to I, Δt to E, and L/kA to R.

Despite the similarity to Ohm's Law, there are two significant practical differences:

Practical absolute insulators and conductors of heat are non-existent compared to the many electrical conductors and insulators.

The motion of molecules is limited and greater energies are involved than for movement of electrons.

The thermal conductivity k is a constant, representing the quantity of heat which flows across unit area in unit time when the length of heat path is unity and the temperature gradient across this path is unity. The numerical value depends on the material, being high for metals and low for insulators. For example, the thermal conductivity of copper is over 300 times that of glass.

If q (the rate of heat transfer) is measured in watts, area in square inches, length in inches, and temperature in degrees C, k has the units of watts/(sq.in.)(^{o}C)/in. While k varies with temperature, the variation for metals over the range of temperature of interest in electronic cooling design is not great. Consequently, for the purpose of this Manual, the variation of thermal conductivity with temperature is disregarded. Table XXVI in Appendix F presents the thermal conductivity values for various materials.

1. An electrical analogy which may be readily applied by electronic engineers is presented by Fig. 3-1.

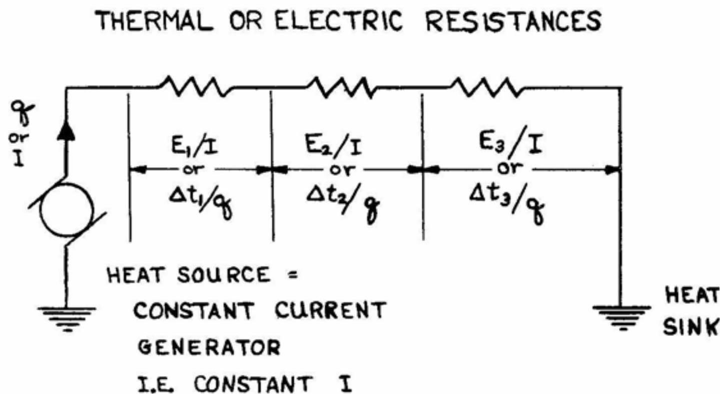


FIGURE 3-1
ELECTRICAL ANALOGY OF HEAT FLOW
FROM ELECTRONIC EQUIPMENT

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:

$$R = \frac{\Delta t}{q} \qquad (3) \qquad (D.E.)$$

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

This analogy will be utilized wherever possible in this Manual when conduction predominates. It is believed that the use of thermal resistance in ^{o}C/watt will aid in simplifying design calculations, since the usage of thermal conductivity with its complex units of BTU-in./hr.-ft.^{2o}F or watts/sq.in.-^{o}C/in. can be minimized. Further, resistance is more commonly used in electronics than conductance.

2. Example 1. - Calculation of the Thermal Resistance of a Bar Insulated on the Sides:

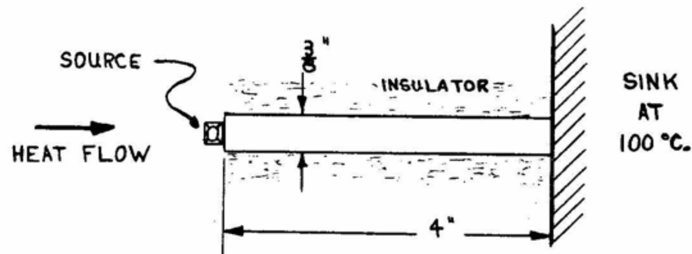


FIGURE 3-2
CONDUCTION THROUGH A BAR

Fig. 3-2 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.

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

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

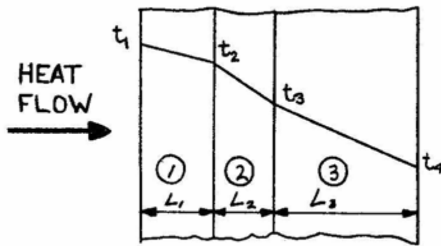
$$A = \frac{1}{4}\pi (3/8)^2 = .110 \text{ sq. in.}$$

Thus, the conductance of the bar is:

$$\frac{1.18 \times .110}{4} = .0324 \text{ watts}/^{\circ}\text{C}$$

and the thermal resistance is $\frac{1}{.0324}$ or 31°C/watt. The temperature rise for 2 watts is 2x31 or 62°C. Consequently, the temperature of the end of the bar and the surface of the resistor is 162°C.

3. Conduction through Composite Walls



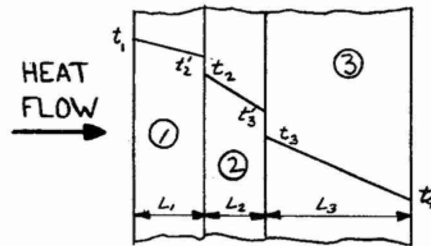


FIGURE 3-3a. FIGURE 3-3b.
CONDUCTION THROUGH COMPOSITE WALLS

Fig. 3-3a. 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:

$$q = \frac{(t_1 - t_4)}{\frac{L_1}{Ak_1} + \frac{L_2}{Ak_2} + \frac{L_3}{Ak_3}} \qquad (4).$$

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:

$$t_2 - t_3 = \frac{\frac{L_2}{Ak_2}}{\frac{L_1}{Ak_1} + \frac{L_2}{Ak_2} + \frac{L_3}{Ak_3}} (t_1 - t_4) \qquad (5)$$

In cases where the materials are not actually bonded together, the thermal resistance at each joint should be considered. Fig. 3-3b. 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:

$$q = \frac{(t_1 - t_4)}{\frac{L_1}{Ak_1} + R_{1-2} + \frac{L_2}{Ak_2} + R_{2-3} + \frac{L_3}{Ak_3}} \quad (6)$$

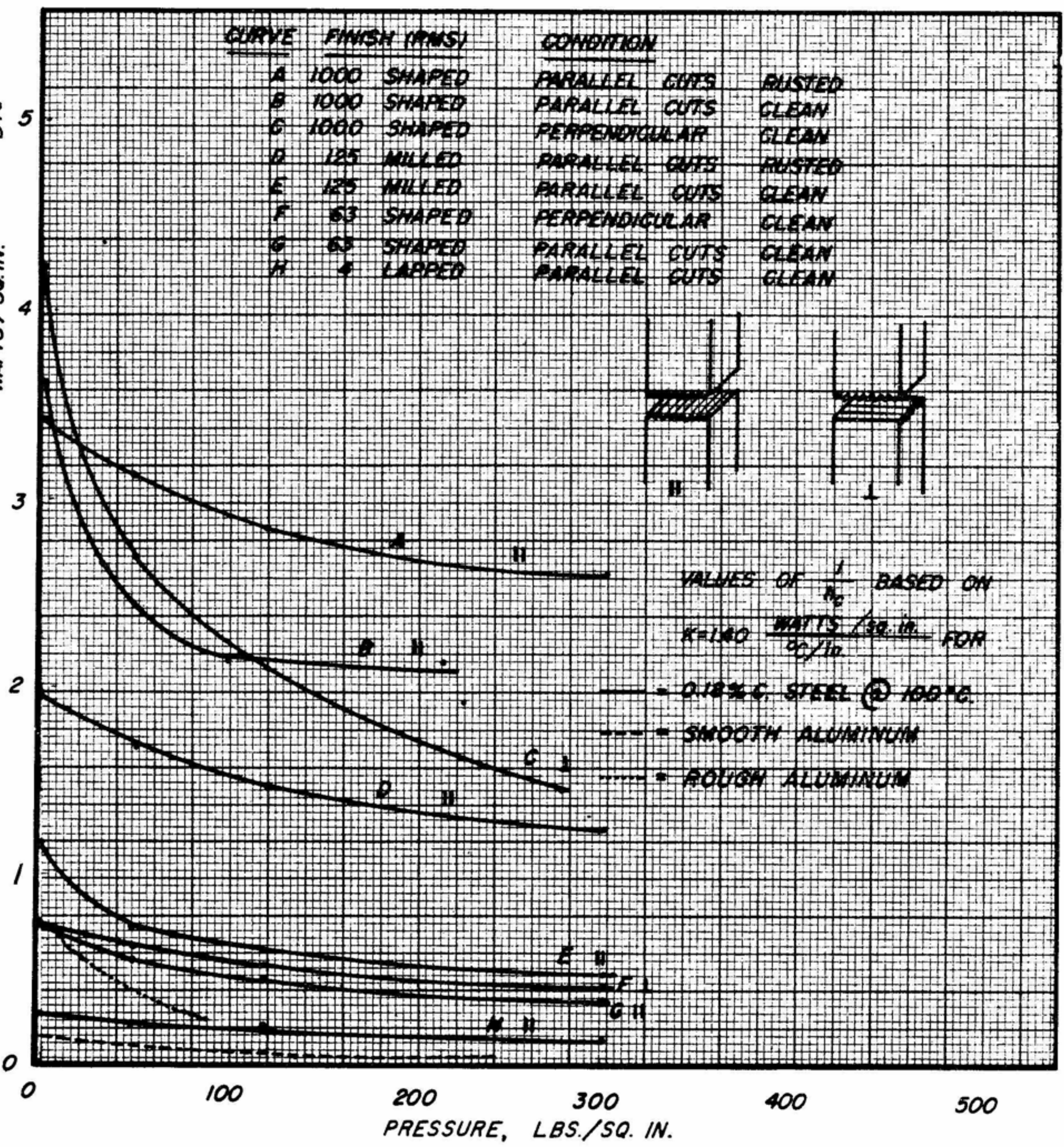
where:

R_{1-2} and R_{2-3} are the thermal contact resistances 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. (See References 21 and 22 in Bibliography Appendix D.)

4. Contact Resistance

The contact resistance investigation results of Reference 21 are reproduced in Fig. 3-4, in which contact resistance is correlated as a function of contact pressure using various steel surfaces. The surface 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-in. (4 millionths of an inch) would constitute an extremely smooth surface.

Fig. 3-4 shows a wide variation in contact resistance, especially at the lower contact pressures. It is of interest to convert the contact resistance into an equivalent length of material whose resistance due to



CONTACT RESISTANCE AS A FUNCTION OF CONTACT PRESSURE

Fig. 3-4

pure conduction would be the same. For example, a contact resistance for a cross-section of unit area, (i.e. actually the contact resistivity) of $1.0^{o}C\text{-in.}^2/\text{watt}$ is equivalent to 1.18 inches of steel of unit cross-sectional area

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

in pure conduction. (The thermal conductivity of mild steel is 1.18 watts/(sq.in.)(^{o}C)/in.) With aluminum the thermal resistance of a good joint will be equivalent to that of a length of from $\frac{1}{2}$ in. to 1 in. of aluminum of the same area. Hence, a relatively rough surface contact may easily result in a higher thermal resistance than the metal itself. Metal foil placed in a joint usually decreases the thermal resistance.

Table (I) summarizes the contact resistance of various joints, all tested at 10 psi. One column presents the resistance of the joints when filled with oil. These general conclusions follow: (Ref. 22)

> The thermal resistance of dry joints decreases linearly with pressure for steel. The thermal resistance of dry joints decreases exponentially with pressure for bronze and aluminum. It is believed that this is due to the difference in hardness between these materials.
>
> The thermal resistance of both dry and oil-filled joints decreases with a decrease in roughness.
>
> At 10 psi the thermal resistance of an oil-filled joint is about one-half that of a dry joint. The effect of the oil decreases at higher pressures.
>
> The thermal resistance decreases if one surface of a steel joint is copper plated.

<u>Example 2</u>: Conduction with and without contact resistance.

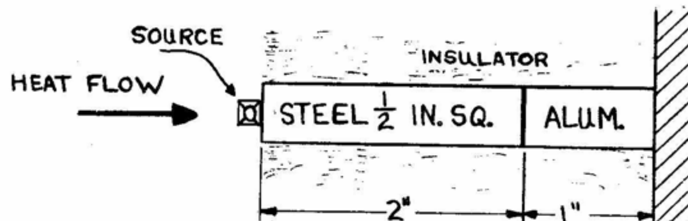


FIGURE 3-5
COMPOSITE BAR WITH CONTACT RESISTANCE

TABLE I.

COMPARISON OF THERMAL RESISTANCE MEASUREMENTS OF JOINTS AT 10 PSI

Joints		Surface Roughness RMS, Microinches			Thermal Conductance per Unit Area BTU/(hr)(sq.ft.)(°F)			Thermal Contact Resistivity or Thermal Resistance per Unit Area, °C/watt/in.^{2}
		Surface #1	Surface #2	Mean	300°F. Dry	500°F. Dry	300°F. Oil	300°F. Dry
Steel	Smooth vs. Smooth Rough vs. Rough	3 70	3 85	3 78	2200 400	3600 800	- 1350	.12 .68
No. 1 Alum.*	Smooth vs. Smooth Rough vs. Rough	16 60	17 60	16 60	1800 1300	3500 1500	- 2000	.15 .21
No. 2 Alum.**	Smooth vs. Smooth Rough vs. Rough	15 20	10 50	13 50	1900 500	2500 650	- 1600	.14 .55
Bronze	Rough vs. Rough	70	80	75	800	1200	1200	.34
No. 1 Alum.*	Smooth vs. Smooth Steel Surface	15	90	66	800	-	1600	.34

Thermal conductivity of oil, $k_x = 271$ to 398×10^{-6} cal./(cm)^2(sec.)(°C)/cm

* Alcoa No. A-51-S
** Alcoa No. 18-3

Also:

Steel is SAE 4140
Bronze is AMS 4846

Fig. 3-5 presents a composite bar $\frac{1}{2}$ in. sq., cross-sectional area .25 sq. in., 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 1.36 x 10 = 13.6°C.
The thermal resistance of the steel R_s

$$r_s = \frac{1}{k} = \frac{1}{1.18} = .85^{\circ}\text{C-in.}^2/\text{watt-in.}$$

$$R_s = \frac{r_s L}{A}$$

$$R_s = \frac{.85 \times 2}{.25} = 6.8^{\circ}\text{C/watt}$$

The temperature rise in the steel is 6.8x10 = 68°C.
The thermal resistance of the aluminum, R_A

$$r_A = \frac{1}{k} = \frac{1}{5.1} = .196^{\circ}\text{C-in.}^2/\text{watt-in.}$$

$$R_A = \frac{.196 \times 1}{.25} = .78^{\circ}\text{C/watt}$$

The temperature rise in the aluminum is .78x10 = 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 $\frac{1}{2}$ in. sq. aluminum bar 1 in. long is .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 .41°C/watt.

a. Theoretical Considerations

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 for 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. Such a configuration is capable of changes in an infinite number of ways; 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 (Ref. 45).

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 at pressures less than 1000 psi, not because of the difference in over-all deformation of the specimens but because of the local deformation of the so-called "peaks". Similarly, it is noted that the effect of pressure is more pronounced in softer materials than in harder materials (Ref. 45).

5. Conduction Across Thin Air Gaps

It has been found that some heat transfer, other than by radiation and convection, occurs between close parallel surfaces by gaseous conduction. This mode is effective for distances up to about $\frac{1}{4}$" in. Beyond this distance a convective effect becomes apparent, especially for high vertical surfaces. At distances greater than $\frac{1}{2}$ in. the convective heat transfer increases and approaches, for distances greater than 1 in., a value equivalent to that for a heated surface in free air.

In dealing with thin air gaps, say $\frac{1}{4}$ in. or less it can be assumed that convection is negligible, but not radiation, and that conduction offers a resistance of:

$$R = \frac{L}{Ak} \qquad (7) \quad (D.E.)$$

where:

L is the thickness of the air gap

A is the cross-sectional area, and

k is the thermal conductivity of air (see Table XXIV) at the average air temperature.

6. Conduction through Other Shapes

The foregoing indicates that pure conduction through a wall or bar, cylinder and sphere is relatively simple to evaluate. Complex shapes can be predicted by graphical or numerical means. Where conduction through complicated shapes is accompanied by convection or radiation or both, it is sometimes difficult to define the problem mathematically. This results from the complexity of the thermal paths associated with densely packed electronic equipment. These complexities should not give the electronic designer the impression that thermal design is impossible or even very difficult. The heat transfer characteristics of typical assemblies or modules can be determined, if necessary, empirically, and thermal "bench marks" can be established. Further, an electrical method of determining the thermal characteristics of conduction cooled devices may be utilized in certain instances because of the relationship of thermal and electrical conductance and resistance.

7. The Relation of Thermal and Electrical Resistance

a. General

The possibility of simplifying determinations of thermal resistance in electronic equipment by measuring the electrical resistance of and between connected metal parts and converting these values to thermal resistances mathematically has been investigated. It appears that this technique will aid in evaluating the cooling characteristics of electronic equipment and will apply with acceptable accuracy to assemblies wherein heat transfer by metallic conduction predominates and both convection and radiation are negligible.

b. Good Conductors

In 1853 Wiedemann and Franz discovered that the electrical and thermal conductivities (σ and k) of various pure metals at any given temperature (T) have a practically constant ratio. In 1940 Seitz (Ref. 12) showed that the free electron theory of metals predicts that the so-called Wiedemann-Franz ratio $k/T\sigma$ should be a universal constant having a value of 2.45×10^{-8} watt-ohm/$^{\circ}C^2$. This theoretical value is approached by good conductors at temperatures of 0 to 200°C. For example, the value for copper at room temperature is 2.28×10^{-8} watt-ohm/$^{\circ}C^2$, or 93% of the theoretical value.

CALCULATION OF $k/T\sigma$ FOR COPPER

Seitz (Ref. 12) gives the relation

$$k/T\sigma = (\pi^2/3)(B/e)^2 \text{ watt-ohm}/^{\circ}C^2 \qquad (8)$$

where:

k = thermal conductivity, watts/°C-cm.

T = absolute temperature, deg. Kelvin (273 + °C.)

σ = electrical conductivity, $ohm^{-1}\ cm^{-1}$

B = Boltzmann's constant, 1.380×10^{-16} erg/°C-cm.

e = electron charge, 1.602×10^{-20} emu (abcoulombs)

Commonly available tables give data for copper, thus

k = 0.918 cal.-cm/sec.-cm^2-°C.(or cal./sec.-cm-°C.), at 18°C.

$\rho_e = 1/\sigma = 1.7241 \times 10^{-6}$ ohm-cm *(I.A.C.S.), at 20°C.

Converting units of k to watts/°C-cm., and σ to $ohm^{-1}\ cm.^{-1}$,

it follows that:

$$k/T\sigma = \frac{4.186 \times 0.918}{(273+18)\times 0.58\times 10^6}$$

$$= 2.28 \times 10^{-8} \text{ watt-ohm}/^{\circ}C^2 \qquad (9)$$

The theoretical value of $k/T\sigma$ is 2.45×10^{-8} watt-ohm/°C^2

Thus, the ratio of actual to theoretical values is 2.28/2.45 = 0.93

Table II shows that the ratios of k/σ at room temperature and hence the Wiedemann-Franz ratios ($k/T\sigma$) of common pure metals and certain alloys, except constantan and manganin, are roughly constant. Arbitrary units appear in the table; k is in cgs units and σ is a dimensionless ratio based on copper. Values of k/σ are close to unity, but this fact is not significant. However, it happens that the value of k/σ for copper (0.92) almost equals the ratio of the actual value to the theoretical value of $k/T\sigma$ (0.93).

*International Annealed Copper Standard

TABLE II.

ELECTRICAL VERSUS THERMAL CONDUCTIVITY
OF COMMON METALS AND ALLOYS AT 20°C.

(From Ref. 13 and 14)

Metal	σ * El.Conductivity Metal/Copper	k Th.Conductivity cgs Units	k/σ Th.Conductivity / El.Conductivity
Alum., Commercially Pure	0.61	0.50	0.83
Alum., 24S-0	0.50	0.45	0.90
Alum., 24S-T	0.30	0.29	0.97
Alum., 52S-0, H	0.35	0.33	0.94
"Berylco" #10 Alloy	0.51	0.586	1.15
Brass (Cu-Zn)	0.26 - 0.43	0.29 - 0.44	1.11 - 1.02
Chrome-Copper	0.829	0.773	0.93
Constantan (Cu-Ni)	0.035	0.054	1.54
Copper	1.00	0.92	0.92
Iron	0.17	0.16	0.94
Magnesium	0.38	0.37	0.97
Manganin-(Cu-Ni-Mn)	0.039	0.052	1.33
Mercury	0.018	0.019	1.05
Nickel	0.16	0.14	0.88
Silver	1.05	1.00	0.95
Tin	0.15	0.15	1.00
Wood's Alloy	0.033	0.032	0.97
Zinc	0.30	0.27	0.90
Non-Metal (For Comparison)			
Gas-Carbon	0.0005	0.01	20.

* Relative electrical conductivity obtained by computing copper/metal resistivity ratio.

Seitz derived the relation for $k/T\sigma$ by using a relation for k in which T is a factor and a relation for σ in which T is absent. However, k is not proportional to T and σ is not independent of T. Both k and σ vary with T rather differently; it can only be said that k increases and σ decreases as T increases. The value of $k/T\sigma$ for any metal is only approximately constant; in general, it increases gradually with T and approaches the theoretical value at temperatures above $0^{o}C$.

The following data for aluminum indicates that k is almost directly proportional to T (because k/T is fairly constant): (Ref. 13, p. 2086)

TABLE IIIA.

TEMPERATURE VARIATION OF THE WEIDEMANN-FRANZ RELATION FOR ALUMINUM

T, °C.	T, °K.	k (cgs)	k/T
100	373	0.49	1.31×10^{-3}
200	473	0.55	1.16×10^{-3}
300	573	0.64	1.12×10^{-3}
400	673	0.76	1.13×10^{-3}
600	873	1.01	1.16×10^{-3}

The following data for copper indicates that σ is roughly inversely proportional to T (because σT is roughly constant): (Ref. 13, p. 2187)

TABLE IIIB.

TEMPERATURE VARIATION OF THE WEIDEMANN-FRANZ RELATION FOR COPPER

T, °C.	T, °K.	σ (onm-cm)$^{-1}$	σ T
20	293	$10^{6}/1.72$	170×10^{6}
100	373	$10^{6}/2.28$	164×10^{6}
200	473	$10^{6}/2.96$	160×10^{6}
500	773	$10^{6}/5.08$	152×10^{6}
1000	1273	$10^{6}/9.42$	135×10^{6}

If k/T and σT were strictly constant for any metal, $k/T\sigma$ would not be constant but proportional to T. (Actually $k/T\sigma$ increases gradually with T, as noted before). Unfortunately, little information on values of k and σ of various metals at various temperatures is available, but the above data will serve to indicate a general trend.

c. Semi-Conductors

As the name semi-conductor implies, semi-conductors exhibit poor conductivity by comparison with metals. For example, carbon in the form of gas-carbon blocks has about one two-thousandth of the electrical conductivity of copper, although it is one of the best non-metallic electrical conductors. However, the thermal conductivity is only 1/24 that of copper. Another outstanding difference between good conductors and semi-conductors is that the former have higher electrical resistivity and the latter have lower resistivity at elevated temperature than at room temperature. Pure metals have positive and fairly constant temperature coefficients of electrical resistivity (.003 to .005/°C. at 20°C. for pure metals) whereas semi-conductors have negative and quite variable temperature coefficients of resistivity. Temperature coefficients of conductivity are not given in the literature, but they would be equal numerically to the resistivity coefficients and would have opposite algebraic signs.

Semi-conductors, like metals, usually show measurable Hall effect. Some hybrid cases occur; some semi-conductors are linear; they obey Ohm's law well at constant temperature. However, linear semi-conductors other than graphite (which is semi-metallic) are seldom used in electric circuits. Little information on the electrical conductivities of semi-conductors at various temperatures is available, and data obtained from different sources often disagree. Disagreements are usually attributable to differences in test methods or the "purity" of samples. Obviously, semi-conductors may exhibit variable ratios of thermal to electrical conductivity compared to metals.

d. Insulators

Unfortunately, it is practically impossible to determine k/σ for insulators, since their electrical conductivities are extremely low and almost immeasurable.

e. Joints

Both electric circuits and heat transfer systems usually contain a series of conductive parts and joints between them. The electrical and thermal conductances of parts having uniform cross-section are expressed by similar relations:

$$G = \sigma A/L \text{ and } C = kA/L \quad \text{(c.g.s. units)} \quad (10)$$

where:

σ , k = electrical, thermal conductivity of material (c.g.s. units)

A = cross-section through which current or heat flows ($cm.^2$)

L = path length of current or heat (cm.)

G = electrical conductance (mhos)

C = thermal conductance (cal./sec.-^{o}C)

Both electrical and thermal conductances of parts are usually measurable and also predictable from known values of conductivity and dimensions. The conductances of joints are usually measurable but not entirely predictable because of non-uniform or indeterminate values of conductivity and dimensions, which are usually unknown functions of the joining force, in the case of common pressure joints or of the quality of workmanship, in the case of brazed, or welded joints.

Electric circuit joints are short and adequate in cross-section as compared with wire lengths, and normally offer little electrical resistance as compared with the total circuit resistance. However, despite careful design, installation and maintenance, the resistance offered to the flow of current across the interface separating two conductors is not always negligible. Contact resistance often produces significant local heating and voltage drops, therefore it is normally avoided.

Electrical contact resistance is the sum of two parts: (1) spreading resistance, due to new boundary conditions for current flow between contacts, and (2) interfacial resistance due to surface potential barriers and material between contacts. The spreading resistance is the sum of two such resistances, one on each side of the contact, and is attributable to the constriction of current flow lines or variable current density in the interface region. The interfacial resistance is a complex function of film thickness and work function for electron emission from the metals or films at the interfaces.

The thermal and electrical resistances of contacts appear to obey similar laws, and the following experimentation indicates that these two contact resistances seem to bear the same ratio as the thermal and electrical resistances of a single metallic conductor. As one might expect, the law of Wiedemann and Franz applies to clean and tight metal-to-metal joints, but not to internally corroded and loose joints.

f. Theory of Method

While the law of Wiedemann and Franz applied only to good conductors, known values of $k/\sigma T$ for all conductive materials should prove helpful in many investigations where only one kind of conductivity is readily measurable but the other kind of conductivity is particularly interesting. For example, if it is difficult but desirable to determine k under certain test conditions, and σ is easily determined, then σ can be used as a measure of k. If σ and T are observed, k is obtained by multiplying $k/\sigma T$ by σT. This presumes that the value of $k/\sigma T$, as determined in preceding tests, is reasonably constant. If $k/\sigma T$ varies with T according to a known law, a different but straightforward procedure must be applied.

If similar current and heat flow patterns exist between two points in a test sample during electrical and thermal tests, and if the same dimensions, material, and average temperature conditions occur, the ratio of thermal conductance to electrical conductance and the corresponding conductivity ratio should obey the relation:

$$C/G = k/\sigma \quad \frac{\text{cal.}}{\text{sec.-}^{\circ}\text{C-mho}} \qquad (11) \qquad (\text{D.E.})$$

This relation applies to all electrically and thermally conductive and semi-conductive parts where the actual or theoretical spacing of the potential probes equals that of the thermocouples so that corresponding electrical and thermal gradients are detectable. The actual and relative values of these gradients is unimportant, as the following analysis will indicate.

Electrical conductance is normally obtained by taking the reciprocal of a measured resistance. This involves the relation

$$G = I/E = 1/R' \text{ ohms}^{-1} \text{ (mhos)} \qquad (12)$$

where I, E, and R' are in amperes, volts, and ohms, respectively.

The resistance depends upon the resistivity (ρ_e) and dimensions (L and A):

$$R' = \rho_e \, L/A \text{ ohms} \qquad (13)$$

Equations (11), (12) and (13) may be combined to yield

$$CR' = k\,\rho_e, \text{ in c.g.s. units} \qquad (14)$$

The thermal conductance between two points may be defined as

$$C = \left(\frac{P}{T_1 - T_2}\right) \text{ watts/}^{\circ}\text{C} \qquad (15)$$

where

P = heat flow, watts (joules/sec.), and

T_1 and T_2 = temperatures in $^{\circ}$C. at the points 1 and 2.

The thermal conductivity is,

$$k = PL/A(T_1 - T_2) \text{watts/cm.-}^{\circ}\text{C}. \qquad (16)$$

Values of k in common c.g.s. units of cal.-cm./sec.-cm.^{2}-$^{\circ}$C. (often shortened to cal./sec.-cm.-$^{\circ}$C.) must be multiplied by 4.186 watt-sec./cal. in order to agree with the units of equation (16). The result of combining equations (14) and (15) is

$$PR'/(T_1 - T_2) = 4.186k\,\rho_e,$$

whence

$$P = 4.186\,k\,\rho_e\,(T_1 - T_2)/R' \text{ watts,} \qquad (17)$$

$$\text{and } (T_1 - T_2) = 0.239\,PR'/k\,\rho_e\ ^{\circ}\text{C}, \qquad (18)$$

where

k is in cal.-cm./sec.-cm.^{2}-$^{\circ}$C and ρ_e is in ohm-cm.^{2}/cm. (or ohm-cm.)

The value of $k\,\rho_e$ is substantially constant for many pure metals and alloys at room temperature. In such cases (where $k/T\sigma$ is constant) equation (17) may be simplified by replacing $4.186k\,\rho_e$ by a constant. Using values of k and ρ_e which correspond to copper, one obtains a constant of $4.186 \times 0.918 \times 1.724 \times 10^{-6}$ or 6.64×10^{-6} watt-ohm/$^{\circ}$C, hence:

$$P = 6.64 \times 10^{-6}\,(T_1 - T_2)/R' \text{ watts, approximately} \qquad (19) \quad \text{(D.E.)}$$

Similarly, equation (18) reduces to the form

$$(T_1 - T_2) = 150{,}000\ PR'\ {}^{\circ}C, \text{ approximately} \quad (20)$$

where R' is in ohms, as usual. If R' is in microhms, the factor 10^{-6} in (19) is omitted and the constant in (20) reduces to 0.15, or

$$R = \frac{(T_1 - T_2)}{P} = .15\ R'\ {}^{\circ}C/\text{watt} \quad (21) \quad (D.E.)$$

Example 3

Sample Calculations - Using Equation (19)

(1.) Let us assume that a temperature difference of 50°C happens to occur between two points 6 inches apart on a length of #10 copper wire (diameter = 0.102 inch). From wire tables, $R' = 0.0005$ ohm, and, from (19), the rate of heat transfer by conduction is

$$P = 6.64 \times 10^{-6}\ (50)/500 \times 10^{-6} \text{ or } 0.664 \text{ watt}$$

This result can be checked by using the relation

$$P = kA\ (T_2 - T_1)/L \text{ watts} \quad (22)$$

where

$k = 9.7$ watt-in./in.^{2}°C (thermal conductivity of copper)

$A = (\pi/4)(0.102)^2$ in.^{2} (cross-section of #10 AWG)

$(T_2 - T_1) = 50$°C (temperature difference)

$L = 6$ in. (path length)

$P = 0.65$ watt (checks within 2.1%)

(2.) Let us assume that the same temperature difference (50°C.) occurs across a metal-to-metal joint having a contact resistance of only 0.0005 ohm (as above). In this case, equation (19) is also applicable, with the same result (0.66). Incidentally, Specification AN-S-27, for radio-shielded ignition harness assemblies, permits a maximum resistance of 0.0005 ohm at any joint.

(3.) Let us assume that the same temperature difference may occur in the wire or joint of case (1) or (2), but the maximum rate of heat transfer must be ten times as great. This condition can be satisfied

in case (1) by shortening the #10 wire to one tenth of its original length, or to 0.6 inch, which reduces R' to 0.00005 (50 microhms). If the wire length is unchangeable (6 inches) the diameter may be increased to $\sqrt{10} \times 0.102 = 0.322$ inch. In case (2) the contact resistance may be reduced ten-to-one in some assemblies by cleaning or tightening the mating surfaces.

(4.) Let us assume, in view of the above, that an excellent conductor or joint has a resistance not exceeding 50 microhms. Equation (19) then yields the corresponding thermal conductance,

$$P/(T_1 - T_2) = 6.64 \times 10^{-6}/50 \times 10^{-6} \text{ or}$$

$$0.133 \text{ watt}/^{\circ}\text{C},$$

which is about as low as can be tolerated in most well-designed equipment. The "50-microhm thermal resistance" (reciprocal of above) is equivalent to 7.5°C /watt (or 13.5°F /watt).

h. Advantages and Disadvantages of the Method

The above indicates the advantage of using electrical measurements to determine thermal properties, where possible. The validity of the method, as applied to good conductors, has been established. Experimentation will show the extent to which the method is applicable to semi-conductors and joints. In the case of the latter, some difficulty may arise when dissimilar metals cause thermo-electric effects which could lead to erroneous measurements of contact resistance. However, it is easier, in general, to measure electrical conductivity than thermal conductivity.

8. Electrical Measurement of Thermal Resistance

a. General

Measurement of the electrical resistance of metallic conductors requires some care and instrumentation capable of measuring extremely low electrical resistances. For the purposes of this Manual, a good heat conductor has been defined as a conductor having an electrical resistance below 100 microhms and a corresponding thermal resistance below 15°C./watt. This arbitrary definition indicates the need for instruments capable of measuring roughly 1 to 100 microhms (0.15 to 15°C /watt), a range of resistance much lower than normal circuit resistances.

b. Instruments

An electrical resistance of 100 microhms is below the range of ohmmeters and Wheatstone bridges; it is just readable on some low-resistance test sets (e.g., G.E. Portable Double Bridge); but it is quite readable on certain special instruments (e.g., Shallcross' Bond Testers and Evershed's Ductor).

Low resistances are also measurable by the voltmeter-ammeter or fall-of-potential method, which requires calculations based on Ohm's law. This method requires relatively large current in order to obtain a readable voltage drop, and care must be used to avoid overheating. A moderate current of 10 amperes produces a drop of only 1 millivolt across a 100-microhm resistance.

If a-c is used, the voltage may be measured with a sensitive VTVM such as a Ballantine a-c Voltmeter with a Decade Amplifier (x10, x100). The a-c voltmeter-ammeter method requires only the voltmeter-amplifier, an a-c ammeter (0 to 5 or 10A.), a step-down transformer (115 to 2.5v, 60 cps), and a "VARIAC" or a rheostat. This method avoids thermo-electric effects and permits simultaneous resistance and temperature measurements with thermocouples, if desired.

Example 4

Typical Applications of Low Resistance Measurements

(1.) Calculation

Aluminum tube, 11 in., $\frac{1}{2}$ in., 52 S-0 Alloy (35% I.A.C.S. conductivity). O.D. of 0.502 in., I.D. of 0.432 in.

$$A = 0.7854 (D_2^2 - D_1^2) = 0.0511 \text{ in.}^2$$

$$R' = \rho_e L/A = (0.67 \times 10^{-6}/0.35)(1/0.0511) =$$

37.5 microhms/in.

(2.) G.E. Portable Double Bridge

Length, in.	Resistance, Microhms	Microhms/in.
5	160	32.0
10	320	32.0
4	125	31.3
8	250	31.3
	Av.	31.7

(3.) Ballantine Electronic Voltmeter, Decade Amplifier, 5-Amp. A-C Ammeter, etc.

Length, in.	Resistance, Microhms	Microhms/in.
8	980 μv/4.1A = 239	29.9
8	530 μv/2.0A = 265	33.1
8	750 μv/3.0A = 250	31.3
4	400 μv/3.0A = 133	33.2
		Av. 31.9

Note: The result (1) exceeds (2) and (3) by 18%. This may be due to a low conductivity value, taken from an ALCOA table. Most aluminum alloys have 40 to 61% conductivity. The average of (2) and (3), 31.8 microhms/in. is equivalent to a thermal resistance of 0.15 x 31.8 or 4.77 $^{\circ}$C/watt-in. from Eq..(21).

d. Precautions

Three precautions must be observed in order to insure fairly accurate electrical measurements of thermal resistance.

First, the actual spacing of "potential" probes must equal the theoretical spacing of thermocouples. This equal-spacing concept is important as the electrical and thermal resistances must relate to the same dimensions and materials in order to maintain the validity of Eq. (19).

Second, the electrical resistance between probes must be measured at a temperature which is nearly an average of the two thermocouple temperatures during actual heat transfer. This requirement is necessary because Eq. (21) is based on the assumption that the electrical and thermal constants of the metal correspond to the same temperature. Fortunately, the constants are not very temperature-sensitive, thus the actual and relative values of electrical and thermal gradients during real and imaginary tests are unimportant. However, it is well to limit these gradients to moderate values, as the temperature coefficients of electrical and thermal resistance normally have unlike signs.

Third, the potential probe locations must lie between the current inlet and outlet locations, likewise the imaginary thermocouples must lie between the heat source and sink. This requirement tends to avoid contact resistance errors and dissimilarity of current and heat flow patterns.

The third precaution deserves further discussion, since it is difficult to establish any good rule for spacing of the measuring points with respect to the points at which current and heat enter and leave the body. Manufacturers of low-resistance test sets usually provide adequate spacing between current and potential leads in probes, but some instruments are found or used with only one lead per probe, in which case the measured resistance includes two electrical contact resistances in series. The effective contact resistance may or may not be a significant fraction of the total resistance, depending upon the contact surface conditions and the clamping force as well as the body resistance. Electrical contact resistance usually follows a relation given by Little and Kouvenhoven (Ref. 20).

$$R'_c = R'_s + R'_i = C_1/(Fn)^{\frac{1}{2}} + C_2 \qquad (23)$$

where C_1 and C_2 are constants,

F is the load (clamping force),

n is the number of contact subdivisions.

The following gives the theory and method whereby two successive measurements of electrical resistance along a uniform conductor suffice to evaluate both contact and body resistance.

e. Determination of Electrical Contact Resistance

Let x and y be two successive distances between a pair of contact clips located along a conductor of uniform cross section, and let ρ'_e be the resistance per unit length of conductor. The measured resistances R'_x and R'_y must be the sum of the two constant but perhaps unequal terminal contact resistances and the variable body resistance:

$$R'_x = R'_c + x\rho'_e \qquad (24)$$

$$R'_y = R'_c + y\rho'_e \qquad (25)$$

If x is the greater distance, then R'_x exceeds R'_y. Subtracting (25) from (24),

$$R'_x - R'_y = (x-y)\rho'_e \qquad (26)$$

If $x-y = a$, $R'_x - R'_y = a\rho'_e$ and, from (24) and (26)

$$R'_c = R'_x - x\,(R'_x - R'_y)/a$$
$$= (1-x/a)\,R'_x + (x/a)R'_y$$
$$= (x/a)R'_y - (x/a-1)R'_x \qquad (27)$$

If the difference between the two distances (a) equals the smaller distance (y), then

$$a = y,\ x = 2y,\ R'_x - R'_y = y\rho'_e \text{ or } x\rho'_e/2, \text{ and}$$
$$\rho'_e = (R'_x - R'_y)/y \text{ or } 2(R'_x - R'_y)/x, \text{ whence}$$
$$R'_c = R'_x - 2(R'_x - R'_y)$$
$$= 2\,R'_y - R'_x \qquad (28)$$

Equation (28) is easier to calculate than (27) as it contains no distance factors such as x/a and (x/a-1). It is only necessary to subtract the larger reading R'_x from the doubled smaller reading R'_y, a process which usually requires only mental arithmetic. For example, let R'_y = 180 microhms (for 2 inch length) and let R'_x = 305 microhms (for 4 inch length), then R'_c = 2 x 180 - 305 = 55 microhms. The true body resistances $x\rho'_e$ and $y\rho'_e$ are then obtainable from (24) and (25) thus:

$$x\rho'_e = R'_x - R'_c = 305\text{-}55 \text{ or } 250 \text{ microhms, (for 4 in.)}$$
$$y\rho'_e = R'_y - R'_c = 180\text{-}55 \text{ or } 125 \text{ microhms, (for 2 in.)}$$

C. FREE CONVECTION

1. General

For the purpose of this Manual 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. For example, the surface of a warm object situated in still air at a lower temperature heats the air adjacent to the surface. The heated air becomes less dense as its temperature increases and induces convection currents. When the circulation is caused only by differences in

density, the process is called natural or free convection. Circulation may be forced mechanically by blowers or pumps, in which case the heat transfer is by forced convection.

The mechanism of convection may be explained by considering a cool stream of air flowing past a heated surface. Immediately adjacent to the surface there exists a film of air varying in velocity from zero at the surface to the velocity of the main stream at its outer side. This film offers a resistance to heat flow and is influenced by the nature of the flow. In free convection the film is usually in laminar or streamline motion and relatively thick, causing high resistance to heat flow.

The basic equation for convection is:

$$q = h_c A \, \Delta T, \qquad (29)$$

where

q is the heat transfer rate,

h_c is the coefficient of free convection,

A is the surface area, and

ΔT is the temperature difference between the surface and the main fluid stream.

The value of h_c (the film or surface conductance per unit area) is influenced by many factors, including not only the properties of the fluid, such as viscosity, density, etc., but the flow conditions and surface characteristics as well. Moreover, h_c is a function of temperature, which results in a nonlinear relation between q and ΔT. Under these circumstances, the solution of convection problems is not as simple a matter as the basic equation suggests. Unlike the similar equation for conduction wherein k, the thermal conductivity, is relatively constant for a given material, the above equation, which contains the elusive variable h_c, must be replaced by another relation. While the thermal resistance concept is as applicable to convection as to conduction, the variable nature of h_c prevents the use of the simple relation $R = \Delta T/q = 1/Ah_c$. The difficulties just noted will be considered later.

Electronic devices situated in gases often lose appreciable percentages of their heat by free convection. Usually the gas is air, but other gases, such as helium,* which exhibit higher coefficients of free convection, may sometimes be used. The coefficients of free convection are

*Care must be exercized in the use of certain gases. Helium, for example, penetrates glass and should not be used with vacuum tubes.

usually relatively low and the corresponding thermal resistances are usually relatively high as compared with other factors in thermal circuits. Thus, if conduction and convection resistances are virtually in series, the convection is usually the dominant limit to heat transfer.

2. Theoretical Considerations

The basic relation between the factors in free convection for any fluid (gas or liquid) is expressed by the equation

$$h_c L/k = C(g\beta\Delta T L^3 \rho^2/\mu^2)^m (c_p\mu/k)^n, \quad (30)$$

which can be shortened to a relation of three dimensionless numbers: the Nusselt, Grashoff, and Prandtl numbers, as follows:

$$(Nu) = C(Gr)^m (Pr)^n, \quad (31)$$

where

$$(Nu) = h_c L/k, \quad (32)$$

$$(Gr) = g\beta\Delta T L^3\rho^2/\mu^2, \quad (33)$$

and

$$(Pr) = c_p\mu/k. \quad (34)$$

C is a factor related largely to the size, shape, and position of surfaces, and m and n depend upon the range of values of the named groups of variables.

The symbols, nomenclature, and units commonly used by non-electrical engineers in solving convection problems based upon the above equations are tabulated in Table IV. While any consistent set of units might be used, these conventional units will be retained, with few exceptions, in what follows, because such units generally appear in engineering tables of physical properties of fluids.

The three named groups, (Nu), (Gr), and (Pr), are important in free convective heat transfer, and their relation has been used to correlate experimental data. It has been found experimentally that the exponents m and n of Eq. (31) are very nearly equal. This fact justifies the use of the equation

$$h_c = C(k/L)(aL^3 \Delta T)^m, \quad (35)$$

where

$$a = g\beta\rho^2 c_p/\mu k$$

and

$$(aL^3 \Delta T) = (Gr)(Pr)$$

TABLE IV.

CONVENTIONAL SYSTEM OF UNITS
FOR FREE CONVECTION EQUATIONS

Symbols	Quantity	Units
h_c	Coefficient of convection	Btu/(hr.)(sq.ft.)(°F)
L	Significant dimension (See Table V)	ft.
k	Thermal conductivity	Btu/(hr.)(sq.ft.)(°F)/ft.
g	Acceleration due to gravity	4.17×10^8 ft./hr.^{2}
β *	Coefficient of thermal expansion	cu.ft./(cu.ft.)(°F)
ρ	Density	lbs./cu.ft.
μ	Viscosity	lbs./(ft.)(hr.)
c_p	Specific heat at constant pressure	Btu/(lb.)(°F)
V	Velocity	ft./(hr.)
ΔT	Difference in temperature of surfaces and outer body of fluid.	°F

* For a gas, the coefficient of thermal expansion is numerically equal to the reciprocal of the absolute temperature, 1/°R., where °R. = °F. + 460.

L is a significant dimension which varies with the size, shape, and position of the convective surface. Table V lists L-values for common conditions and may be used for irregularly-shaped electronic parts if the most similar shape of surface is selected. For values of L less than 2 feet in height in a gaseous medium and for $aL^3 \Delta T$ values between 10^3 and 10^9 (in Table IV units) the exponent m is about 0.25. When $aL^3 \Delta T$ exceeds 10^9, m approaches 0.33, however, almost all free convection calculations for electronic equipment can be made on the former basis, replacing m by 0.25 in Eq. (35).

C, like L, is a function of shape and position, and is almost independent of dimensions, however, Table VI shows that C is much larger for small parts such as tubes, resistors, relays and transformers in confined spaces. Reference (24) shows that free convection test data for such small, enclosed, irregular-shaped parts are best fitted by the equation

$$(Nu) = 1.45 \ (Gr \times Pr)^{0.23} \qquad (36)$$

where the constant 1.45 replaces the C of previous equations, and the exponent 0.23 can be changed to 0.25 for convenience without causing serious error.

The unit heat dissipation q/A is obtained by combining Equations (29) and (35), where m is 0.25:

$$q/A = h_c \Delta T = C(k/L)(aL^3 \Delta T)^{0.25} \Delta T$$

$$= Ck(a/L)^{0.25} \Delta T^{1.25} \text{ Btu/hr.ft.}^2 \qquad (37)$$

(See Table VI for C of large parts where 1.45 is not applicable.)

In order to apply the above heat transfer theory effectively to the design of electronic equipment, it is desirable to express the unit heat dissipation in watts/in.^{2} and ΔT in $^{\circ}$C.

Then:

$$P/A = 0.00425 \ Ck(a/L)^{0.25} \Delta T^{1.25}, \text{ w./in.}^2, \qquad (38) \quad (D.E.)$$

where k, a, and L are in conventional units of Table IV ΔT in $^{\circ}$C.

The significant dimension L should be taken as the height (in ft.) of miniature and subminiature tubes, the height of relays and transformers, and the product over the sum of

TABLE V.

SIGNIFICANT DIMENSION "L"

Surface	Position	Characteristic Length, L, ft.
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	any	radius, (diameter/2)

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TABLE VI.

VALUES OF C TO BE USED IN EQUATION (35)

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

the diameter and length of horizontal resistors. Most free convection problems related to electronic equipment in air involve significant dimensions or heights of surfaces not over 2 ft. In such cases, by taking C as 1.45, Eq. (38) can be rewritten with a single constant as

$$P/A = 0.00616k\ (a/L)^{0.25} \Delta T^{1.25},\ \text{w./in.}^2 \qquad (39)\quad (\text{D.E.})$$

Both k and a for air vary with temperature, but $ka^{0.25}$ is fairly constant over a large practical range of temperature, as shown by Table VII. In the range of 0 to 100°C., for example, $ka^{0.25}$ has a value of 0.52 ± 0.02 ($\pm$ 4%), and can be considered a constant. On this basis, Eq. (38) can be simplified to yield

$$P/A = 0.0022C\ (\Delta T)^{1.25}/L^{0.25},\ \text{w./in.}^2, \qquad (40)$$

or

$$P = 0.0022C\ (\Delta T)^{1.25} A\ /L^{0.25},\ \text{w.} \qquad (41)\quad (\text{D.E.})$$

Fig. 3-6 is a nomographic chart of Eq. (41). It was prepared to solve common, free-air convection problems simply and speedily. Because there are four independent variables (instead of the usual two), it is necessary to apply a straight edge three times (instead of once) in order to solve for the dependent variable. First, if P is the unknown, the straight edge is applied to given values of C and L to get a crossing point on the uncalibrated X scale, which represents a partial product (or quotient). Second, the X-point and a given ΔT value determine a line which crosses a P/A scale and gives the unit heat dissipation, which is a further partial product. Third, and last, the P/A value is transferred to another scale (the C-scale) and a given area value is found on the A-scale (the L- ΔT scale), thus giving two points which determine a third line and a crossing (final product) on the P scale (the first-used P/A scale). If necessary, correction for altitude can be made by multiplying P (or P/A) by the square root of the pressure expressed in atmospheres (a decimal).

The chart is not limited to the type of problem just described for purpose of illustration. It may be used for the solution of unknowns other than P, but the sequence of operations must be varied correspondingly, according to rules given on the chart. The required steps are much simpler to take than the alternative mathematical procedure. Thus, for example, one would find difficulty in solving Eq. (39) for ΔT, which comes out in the form

$$\Delta T = 58.8\ L^{0.2}(P/A)^{0.8}/a^{0.2}k^{0.8}\quad {}^{\circ}\text{C.} \qquad (42)$$

(The constant is 0.75 and P is q if only conventional units are used.)

TABLE VII.

CONVECTIVE PROPERTIES OF AIR

I Temperature °F.	II °C.	III Thermal Conductivity, $k \times 10^3$ Btu/hr.ft.°F	IV $a \times 10^{-6}$ Free Convection Modulus, $(ft.^3 °F)^{-1}$	V $a^{0.25}$ From Col. IV	VI $ka^{0.25}$ From Col's. III & V	VII $2ka^{0.25}$ From Col. VI
-50	-45.6	11.6	5.46	48.2	0.559	1.118
0	-17.8	13.2	3.00	41.7	0.550	1.110
50	10.0	14.5	1.81	36.7	0.531	1.062
100	37.8	15.8	1.20	32.9	0.518	1.036
150	65.6	17.0	0.82	30.0	0.510	1.020
200	93.3	18.2	0.58	27.5	0.501	1.002
250	121.1	19.2	0.42	25.3	0.486	0.972
300	148.9	20.4	0.31	23.4	0.477	0.954
350	176.7	21.6	0.23	21.8	0.471	0.942
400	204.4	22.7	0.18	20.6	0.468	0.936
450	232.2	23.9	0.14	19.3	0.461	0.922
500	260.0	25.0	0.11	18.0	0.450	0.900
550	288.0	26.4	0.086	17.1	0.452	0.904
600	315.6	27.1	0.069	16.1	0.436	0.872
650	343.3	28.2	0.055	15.2	0.429	0.858
700	371.1	29.1	0.044	14.6	0.425	0.850

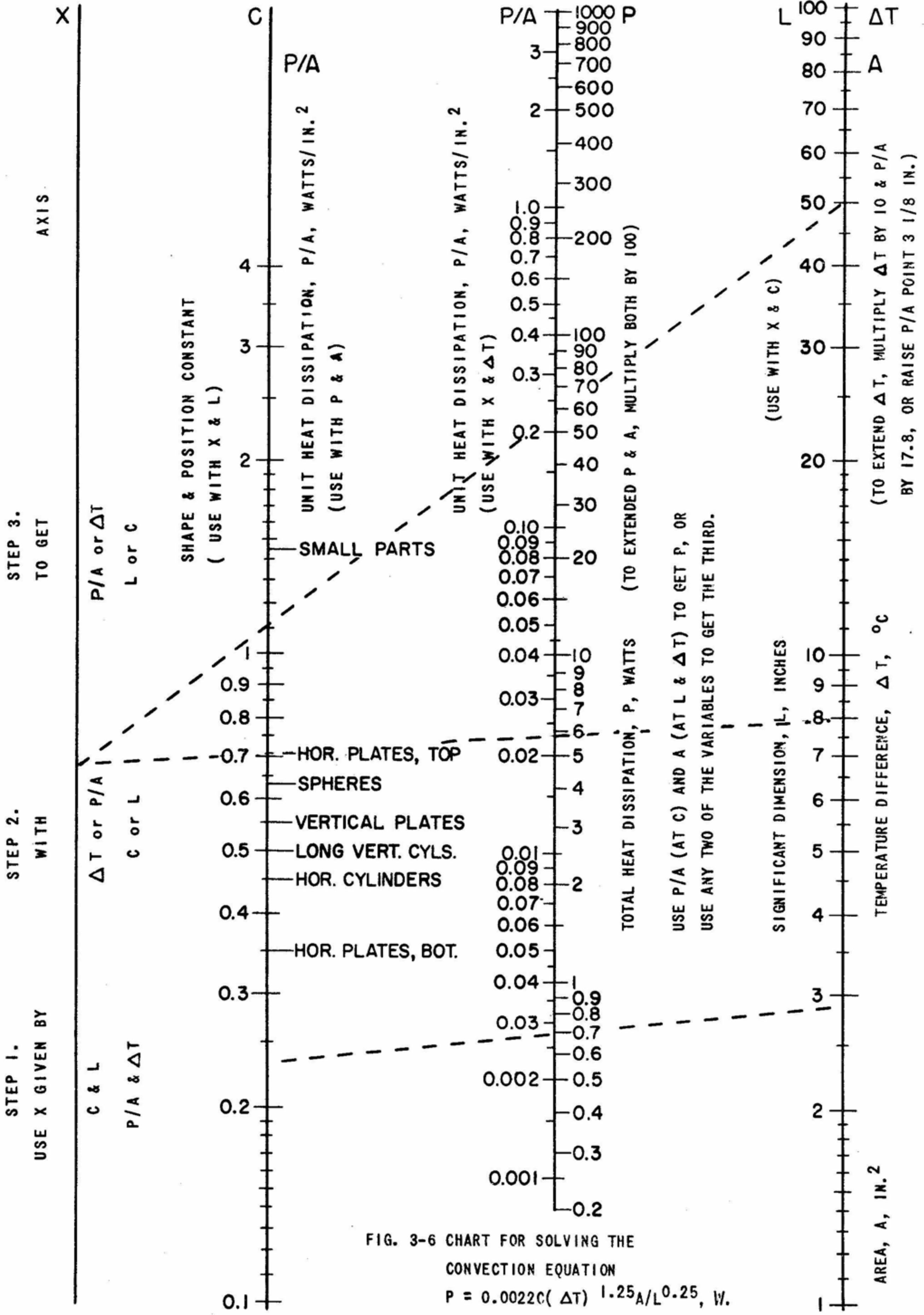


FIG. 3-6 CHART FOR SOLVING THE CONVECTION EQUATION

$P = 0.0022C(\Delta T)^{1.25}A/L^{0.25}$, W.

The same five variables occur in Eq. (42) as in Eq. (39), namely: ΔT, L, P, A, and a "hidden" C (1.45); however, a and k are now raised to different powers and the denominator is not a previously-determined constant. The advantage of the chart is obvious, and its accuracy is adequate for most engineering purposes.

It is evident that, in both Eq. (44) and Eq. (42), the dependent variable (P or ΔT) varies slowly with variations of L. This means that it is unnecessary to determine L with much accuracy in order to obtain reasonably accurate solutions. The L scale on the chart has been calibrated in inches instead of the conventional feet, for easier use by electronic equipment designers. The calibrated C scale contains typical labelled constants, which apply to air or nitrogen.

In the previous Manual, convection problems were solved with the aid of families of curves. These curves presented unit heat dissipation vs. temperature rise for various surfaces (sizes, shapes, and orientations), and implied certain working ranges of significant dimensions. Curves of total heat dissipation vs. area for various temperature rises (and surface conditions) may also give equivalent information. In either case, it is evident that the curves do not always give accurate solutions because they largely disregard variations in C and L, which are variables in Eq. (41). Fortunately, the ranges of C and $L^{-0.25}$ are usually rather small. The solutions also ignore average air temperature (between the hot surface and the outer body of air) which affects values of a and k. Fortunately, as noted before, the value of $ka^{0.25}$ for air is relatively constant, so this disregard of theory causes little error.

3. Sample Calculations

<u>Example 5</u> Use of the Free Convection Chart and Eq. (41).

The use of the free convection chart and Eq. (41) is illustrated by the following problem:

a. Problem

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.)

b. Solution by Chart

Step (1) Determine significant dimension L by reference to the first entry of Table 2.

L = (length x width)/(length + width), for a horizontal plane.

= (24 x 12)/(24 + 12) or 8 in.

Step (2) Determine constant C by reference to the fourth entry of Table VI. (C is 0.71 for a horizontal plate facing upward.)

Step (3) Determine ΔT by subtraction of air temperature from surface temperature.

$\Delta T = (85-35)$ or $50^{\circ}C$

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 $50^{\circ}C$.-point on the ΔT-scale to determine a crossing point on the right-hand P/A-scale. Read 0.23 $w./in.^2$.

Step (7) Transfer this value (0.23) to the left-hand P/A-scale, and connect with 2.88 $in.^2$ on the A-scale or one hundredth of the actual area (because 288 $in.^2$ exceeds the scale limit). As the A-scale is virtually multiplied by 100, the P-scale crossing point must be multiplied by 100, to read 68 watts.

c. Solution by Calculation

Step (1), (2), (3), and (4) are the same as in b.

Step (5) Substitute the values obtained from Steps (1) - (4) in Eq. (12):

$$P = 0.0022C\ (\Delta T)^{1.25}\ A/L^{0.25},\ w.$$

$$= 0.0022 \times 0.71 \times 50^{1.25} \times 288/0.667^{0.25},\ w.$$

$$= 66\ \text{watts}$$

This is in fair agreement with the nomographic solution. The difference is 2/66 or 3%.

D. RADIATION

1. General 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 or 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.

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 "blackbody". There are no perfect absorbers in nature although some surfaces approach blackbody characteristics. The ratio of the energy absorbed by body to that of a perfect "blackbody" is the "absorptivity". In the absence of conduction and convection, a body at thermal equilbrium 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 blackbody is the "emissivity" and is numerically equal to the absorptivity. 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.

Table XXIX lists 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" at the infrared 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 blackbody is:

$$q_b = \sigma_s A T^4 \qquad (43)$$

where:

q is the rate of energy emitted by a blackbody

σ_s is the Stefan-Boltzmann constant

A is the surface area

T is the absolute Temperature

For actual bodies, Eq. (43) 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-blackbodies is:

$$q_r = F_e F_a A \sigma_s (T_1^4 - T_2^4) \qquad (44)$$

where:

F_e is an emissivity factor to allow for departure from blackbody 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 the temperatures of the hot and cold bodies respectively.

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:

$$F_e = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \qquad (45) \qquad (D.E.)$$

where:

ϵ_1 and ϵ_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

$$F_e = \epsilon_1$$

where:

ϵ_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 (Ref. 30).

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 most cases, such as the large parallel planes and enclosed bodies, F_a is unity. For certain other configurations F_a may vary widely and much information is given in Reference (30).

Equation (44) may be written in a slightly different form for ease of solution. Using the Btu-hr.-sq.ft.-°R system, it is:

$$q_r = 0.173\ A\ F_e F_a \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \text{ Btu./hr.} \qquad (46)\ (D.E.)$$

and in the watt-sec.-sq.in.-°K system:

$$q_r = 0.0037\ A\ F_e F_a \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \text{ watts} \qquad (47)\ (D.E.)$$

A radiation chart, Fig. 3-7, has been prepared to aid in the calculation of heat exchanged by radiation. The chart is based upon the equation:

$$P/A = 0.00368e(T/100)^4, \text{ w./in.}^2 \qquad (48) \qquad (D.E.)$$

where:

e = emissivity of the radiating surface, and

T = absolute temperature, °K.

The chart contains three scales and relates the three variables, P/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. 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.

Equation (48) 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. For example, the sky on a clear night radiates about 0.1 w./in.^{2}, which indicates a blackbody (e=1) temperature near -45°C. (-49°F.). The sun on a summer noon radiates about 0.7 w./in.^{2}, which indicates a blackbody temperature near 100°C.

The chart is to be used twice: first, with the emissivity and temperature of the enclosed surface to obtain the total radiation from the source and then 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 twice - backwards.

<u>Example 6</u> 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 150°C. and the objects surrounding the box are at 80°C. Also, the surfaces surrounding the box are in relatively close proximity. Estimate the radiant energy dissipated from the box.

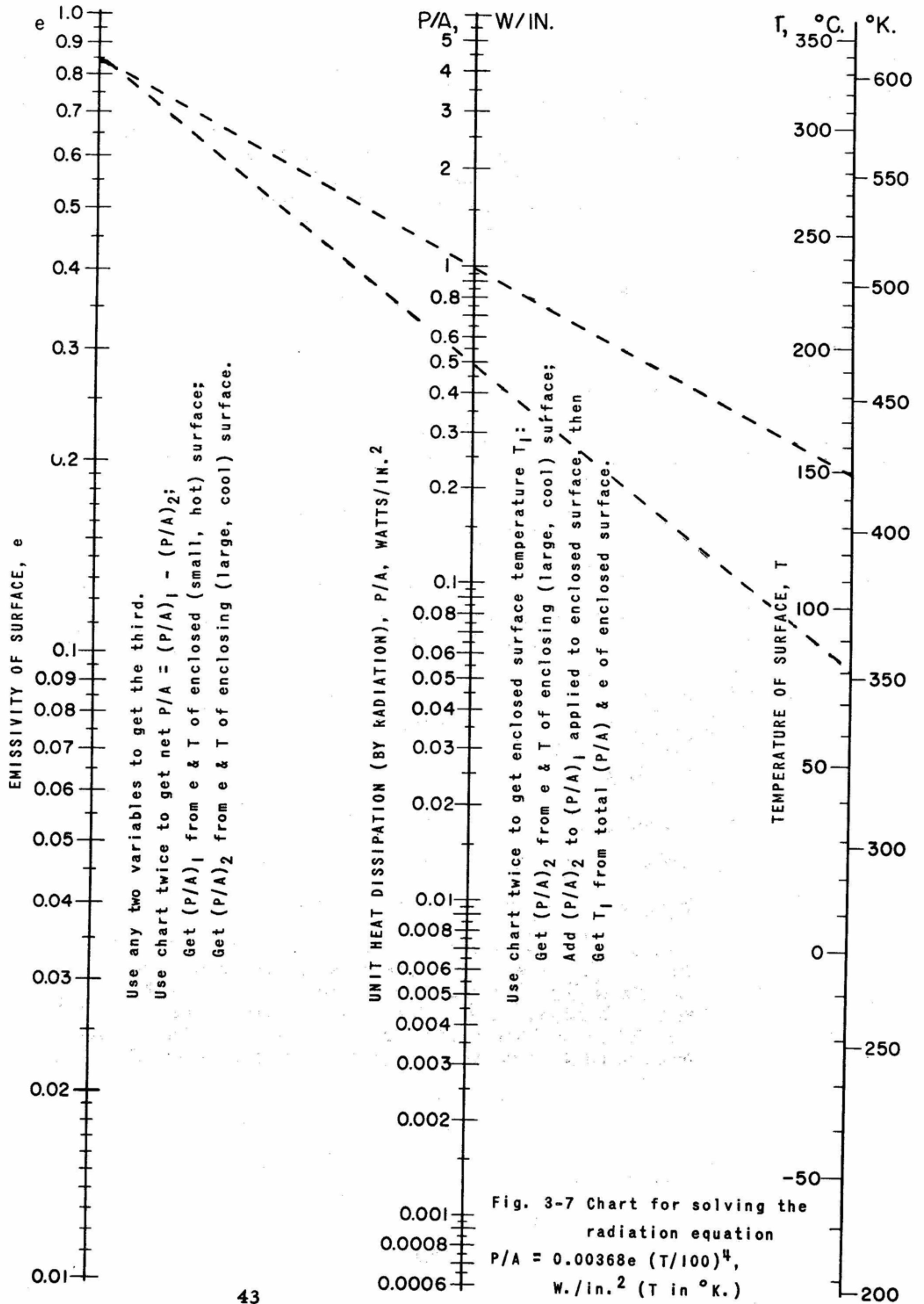


Fig. 3-7 Chart for solving the radiation equation $P/A = 0.00368e\,(T/100)^4$, W./in.^{2} (T in °K.)

Solution:

A_1 = 360 sq. in.

F_e = Assume that the emissivity of the surrounding objects is 0.90. From Table XXIX 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 = 150 + 273 = 423°K.

T_2 = 80 + 273 = 353°K.

$$q_r = 0.0037(360)(0.852)(1.0)\left[\left(\frac{423}{100}\right)^4 - \left(\frac{353}{100}\right)^4\right]$$

q_r = 186.8 watts total radiant heat dissipation.

An alternative solution is by use of the radiation chart, Fig. 3-7. Compute F_e as above

F_e = .852

Draw a line from .85 to 150°C. = 1.0 watts/sq. in.
Draw a line from .85 to 80°C. = .48 watts/sq. in.
The difference is 1.00 - .48 = .52 watts/sq. in.

.52 x 360 = 187 watts

The error between the two methods is small.

<u>Example 7</u> 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 80°C. The following tabulated calculations for the free convective heat transfer are from the free convection chart, Fig. 3-6.

FREE CONVECTION FROM BOX OF EXAMPLE 6

Surface	Significant Dimensions, in.	Area sq. in.		Watts/ sq. in.		Total Watts
Top	$\frac{12 \times 6}{12 + 6} = 4$	72	x	0.393	=	28.3
Bottom	$\frac{12 \times 6}{12 + 6} = 4$	72	x	0.193	=	13.9
Sides	height = 6	216	x	0.276	=	59.7
				Total	=	101.9

Hence, in this case, the total dissipation from the box is 186.8 (radiation) plus 101.9 (convection), or 288.7 watts, of which 65 percent is by radiation.

Radiation and convection usually occur simultaneously. Since free or forced convection 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 at 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 radiant receiver surface at temperature t_r, the total heat transfer rate from the surface by these two modes is:

$$q_t = q_c + q_r = (h_c + h_r) A (t_s - t_a) \qquad (48a)$$

The radiation coefficient is:

$$h_r = \frac{\sigma F_e F_a (T_s^4 - T_r^4)}{t_s - t_a} \qquad (48b)$$

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

2. 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. Some judgment must be exercised. For example, vacuum tubes should not 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. Transparent coatings such as clear lacquer are not recommended for use on reflecting 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.

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 the following properties at a temperature of 25°C. (77°F.): electrical resistance per unit length, 0.67 microhm/in. or 8.0 microhms/ft.; thermal resistance per unit length 0.10°C/w.-in. or 1.2°C/w.-ft. Also, an electrical resistance of 100 microhms represents a length of 150 in. or 12.5 ft. of copper bar one sq. in. in cross section, and a thermal resistance of 15°C/w., while a thermal resistance of 1°C/w. represents a length of 10 in. or 0.833 ft. of copper bar one sq. in. in cross section and an electrical resistance of 6.7 microhms.

Since the thermal resistance to conduction between faces of a 1-inch copper cube is 0.1°C/w., a unit heat dissipation of 1 w./in.^{2} at one face 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 blackbody 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.

Table VIII presents the relative magnitudes of various heat transfer processes.

TABLE VIII.

REPRESENTATIVE MAGNITUDE OF HEAT TRANSFER PROCESSES

	$\frac{\text{Btu}}{(\text{hr.})(\text{sq.ft.})(^{\circ}\text{F})}$	$\frac{\text{Watts}}{(\text{sq.in.})(^{\circ}\text{C})}$	Unit Thermal Resistance $\frac{(^{\circ}\text{C})(\text{sq.in.})}{\text{Watt}}$
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°C., air at 80°C.	0.96	0.00348	287.
Forced convection air over 6 in. plate at 8 ft./sec., mean temp. air & plate of 100°C.	2.84	0.0104	96.
Forced convection 40°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°C. and 50°C.	1.72	0.0063	158.
Radiation between two blackbodies at 500°C. and 50°C.	7.81	0.0287	34.8

E. THE ULTIMATE SINK

Unlike the closed electrical circuit, the thermal circuit is an open one. 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 low resistance and a low temperature sink. In the final analysis the ultimate sink is the earth's atmosphere, large bodies of water, or the earth itself. However, from the practical view, the electronic designer will have available intermediate sinks. In steady-state heat transfer, it is erroneous to consider the chassis of a conventional electronic assembly as a heat sink because the chassis has finite heat capacity and heat must be removed from it at the same rate as that entering.

F. UNITS AND TERMINOLOGY

The Units and Terminology peculiar to electronic cooling should be carefully analyzed dimensionally because of their hybrid nature. The watt, the degree Centigrade, the pound mass, and the inch represent a practical system of simple units even though they are not in strict accordance with standard heat transfer practice. This situation is discussed at length in a companion Manual, C.A.L. Report No. HF-845-D-2 (NAVSHIPS 900,187).

The following units are recommended:

TABLE IX.

Physical Dimension	Unit	Conversion
Length	Inch	1 ft. = 12 in.
Force	Pound Force	
Mass	Slug	Mass in slugs = $\frac{\text{mass in pounds}}{32.2}$
Time	Hour	1 hr. = 3600 secs. = 60 mins.
Temperature	^{o}C.	^{o}C. = 5/9 (^{o}F. - 32)
Absolute temperature	^{o}K. (Kelvin)	^{o}K. = 273.2 + ^{o}C.
Absolute temperature	^{o}R. (Rankine)	^{o}R. = 459.7 + ^{o}F.
Temperature difference	^{o}C	1°C equivalent to 1.8^{o}F temperature difference.
Heat	Watt-hour	One watt-hr. = 3.413 Btu
Rate of heat flow	Watt	One watt = 3.413 Btu/hr.
Unit heat dissipation (rate of heat flow per unit area)	Watts/sq.in.	One watt/sq.in. = 491.4 Btu/hr.-sq.ft.
Thermal resistance	^{o}C/watt	

The following terminology, which is utilized throughout this Manual, 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, a vacuum tube.

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 commonly used term to identify a region 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.

Ambient Temperature - The average temperature of the medium surrounding an electronic device. (One of the factors contributing to the thermal environment.)

NOTE: (1) It is recommended that thermal environment be used in lieu of ambient temperature in specifications and other technical definitions of the thermal conditions surrounding electronic devices. Ambient temperature alone cannot describe the entire thermal configuration because the temperature of the medium surrounding an object is not necessarily related to the heat radiation and conduction effects from nearby heat sources. Ambient temperature ratings have sufficed in the past since the radiation and conduction effects were of small order. However, in modern electronic equipment with its high power densities these effects can be very significant.

The determination of the thermal environment can sometimes be difficult. It is not always possible to measure all of the factors involved. However, for design purposes a comprehensive determination of the important thermal conditions must be made. Included therein are all of the terms mentioned under the above definition of thermal environment before and, if possible, after the installation of the electronic device or devices.

NOTE: (2) The above units and terminology were originally selected as standards for use on a companion contract (NObsr-49228). These standards have since been coordinated with Air Force personnel at W.A.D.C. and their contract personnel at The Ohio State Research Foundation.

IV APPROACHES TO THE THERMAL DESIGN OF EQUIPMENT FOR NATURAL COOLING

A. THE COOLING SYSTEM CONCEPT

1. Fundamental Principles and their Application to Electronic Heat Removal

Electronic equipment must incorporate means for adequate heat rejection in order to provide reliable performance at thermal equilibrium. As long as power is dissipated it will be rejected in the form of heat. The purpose of any electronic cooling system is to provide a low resistance thermal path to a heat sink to absorb this waste heat. When used in conjunction with a heat sink at a reasonably low temperature, such a system will reduce the temperature rise of electronic parts and equipment.

There are several important factors which must be thoroughly understood prior to the initiation of the design of any electronic equipment.

a. The quantity of heat dissipated controls the temperature rise in any given configuration. The division of the total heat generated into the three modes, conduction, convection, and radiation, will be in inverse proportion to the three thermal resistances. Review of Chapter III may lead to the selection by the equipment designer of one or two modes to be emphasized in the thermal design of his particular equipment.

b. Under steady state conditions, thermal equilibrium or a heat balance is always maintained. The natural law of the conservation of energy applies and all of the heat generated will be rejected, if necessary, by means of high temperature gradients.

c. In order to define the thermal parameters, the electronic engineer must first determine temperature and rate of heat production. The dissipated power can usually be measured. Even though temperature measurement is not as easy, the determination of temperature is necessary, since it is the only other measurable quantity in the thermal circuit. Further, temperature is a measure of the quality of the thermal design.

d. The electronic designer, in determining his heat transfer system, must first select the most simple and economical cooling system applicable to the proposed design, environment and specifications. Many factors must be considered: space, economy, the temperature limitations of the electronic parts, the circuit configuration, the thermal environment, the heat concentration and the ultimate sink. A satisfactory thermal design should start on the drafting board simultaneously with the electrical and mechanical design activities.

e. The thermal design must be such that the electronic performance is not significantly affected. In certain instances the optimum cooling technique may not be consistent with electronic performance. When this situation arises, it is recommended that alternate cooling means or circuit designs be utilized. Compromise designs are more often the rule than the exception.

f. Heat transfer design inherently includes reasonably wide tolerances. Due to the nature of heat transfer, a high degree of design accuracy can seldom be achieved. However, this does not impede the design of a practical cooling system. Once the type of cooling system has been tentatively selected, the thermal analysis must be made. The thermal circuit should be approximated as closely as possible to permit mathematical analysis.

2. Approaches to the Thermal Design of Electronic Equipment

There are two basic approaches to the design of electronic equipment with satisfactory thermal performance:

a. The "brute force" method, wherein high temperature electronic parts are used to permit operation without special cooling means, can be used to provide heat rejection through the operation of parts at high temperatures. When used at low ambient temperatures to aleviate the effects of excessive hot spots, this approach is inherently inefficient and expensive. High temperature electronic parts should only be used for operation in environment with high ambient temperatures and in conjunction with an adequate heat removal means. The utilization of high temperature parts is not recommended as a remedy for the deficiencies of an inferior heat removal system.

b. The controlled heat removal method should be used in all Military electronic equipment. This approach embodies the most effective methods of heat transfer and includes special techniques. It requires the application of careful design of the entire thermal system and the establishment of low temperature gradients to protect temperature sensitive parts and circuits. Of prime importance is the necessity of directing the heat from the sources along specified paths to a low temperature sink so that the heat is not indiscriminately scattered and transferred into adjacent electronic parts. The magnitude and location of heat flow must be controlled. For these reasons, certain heat removal methods which will cool heat sources and transfer their heat into other parts are considered undesirable.

3. Methods of Thermally Rating Electronic Equipment

In general, electronic equipments are rated for certain performance at specified ambient temperatures. As stated in Chapter III, ambient temperature rating alone is indeterminate because ambient temperature is only the temperature of the medium surrounding an object and it does not define the true thermal situation as it may exist around an electronic device. With densely packaged equipments, the local air temperature is not directly related to the heat radiation or conduction effects from nearby heat sources.

It is believed that military specifications and equipment ratings should be modified to incorporate thermal environment ratings which can provide the equipment designer and user with definite thermal parameters.

Military specifications for electronic equipment should provide and define facilities for the removal of the dissipated heat at the installation location. The thermal situation should be described both before and after the installation of the equipment.

The electronic equipment manufacturers should furnish the user with the heat dissipation rate of each unit of equipment, the cooling requirements and the maximum temperature of the equipment at specified locations which are indicative of the thermal situation.

B. DESIGN CONCEPTS

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 tubes and 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.

The temperature rise of electronic devices is not normally linear over a significant temperature range. Consequently, temperatures should not be predicted for another ambient condition by extrapolation even though the thermal performance for a given situation is known. Only in those instances, wherein heat transfer by conduction through metals is the predominant mode, will the temperature rise be relatively constant. In general, when conduction does not predominate, the temperature rise at higher temperatures will be smaller than that at lower temperatures due primarily to the increased heat transfer by radiation.

Electronic equipments having a low thermal resistance to a sink will reach thermal equilibrium in a relatively short time. Conversely, equipments which are not well cooled and have a higher thermal resistance to a sink will require a long time to achieve thermal equilibrium. This is analogous electrically to the time constant of an RC network; i.e. if the R is small, the time constant is small. Consequently, a very rough approximation of the degree of cooling of an electronic equipment can be made by noting the warm-up time. In general, well cooled equipments will stabilize in from five to ten minutes.

V METHODS OF COOLING ELECTRONIC PARTS

A. GENERAL

The majority of electronic parts are designed for cooling in sea level air by natural means. Radiation and free convection are normally the primary cooling modes, and conduction is usually a secondary mode. Most heat conduction occurs via the leads and/or wiring to the parts and occasionally through the structure as in the case of a transformer. In general, the surfaces of the parts are such that their emissivity is relatively high and their free convective characteristics are favorable.

Unfortunately, the ideal free ambient air environment for electronic parts is seldom achieved, especially in miniaturized equipment. Thus, the heat producing parts must be derated or cooled by supplemental means. Cooling of parts by natural means is discussed in detail in this section. The recommended method may not always be compatible with a particular design and where possible alternate means are presented.

In order to improve the cooling of parts by natural methods, it is necessary to increase the heat rejection by radiation, convection or conduction. Increasing any of these modes of cooling will reduce correspondingly the temperature of any given part. Simple techniques such as improving the emissivity of a part or its surroundings, providing adequate convection, or thermally bonding a part to a chassis can aid in frequently obtaining adequate cooling. Because of the size of certain heat producing parts, such as small resistors with high heat concentration, simple cooling techniques may not suffice. In such instances, more complex cooling methods utilizing a combination of several heat transfer modes may be required. Parts should only be operated as hot as is reliably tolerable - no hotter!

B. VACUUM TUBES

1. Modes of Heat Transfer in Vacuum Tubes

The transfer of heat within a vacuum tube is a complicated process. A high temperature emitting surface is necessary to maintain electronic emission. Heater temperatures range from 1000°C. to 1300°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 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 a vacuum 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 (see C.A.L. Report#HF-1053-D-3).

Plate temperatures in vacuum 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, (see Table X) vacuum 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 vacuum 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 Fig. 5-1). 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 per cent) 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.

TABLE X.

COEFFICIENTS OF TRANSPARENCY IN THE INFRARED REGION FOR VARIOUS GLASSES*

Wave length (μ)	.7	1.1	1.7	2.3	2.7	3.1	3.7
Crown, borate	1.	.55	.21	.025	.04	--	--
Boro-silicate							
heat resisting**	--	.85	.67	.50	.38	.31	.19
ordinary	--	.74	.61	.33	.034	.021	--
Flint, light	1.	.91	.82	.45	.083	.019	--
Flint, heavy	1.	1.	1.	1.	.45	.019	--
Silica (1 mm. thick)	--	.90	.70	.55	.45	.38	.22

*from Ref. 2.
**from Corning Glass Works

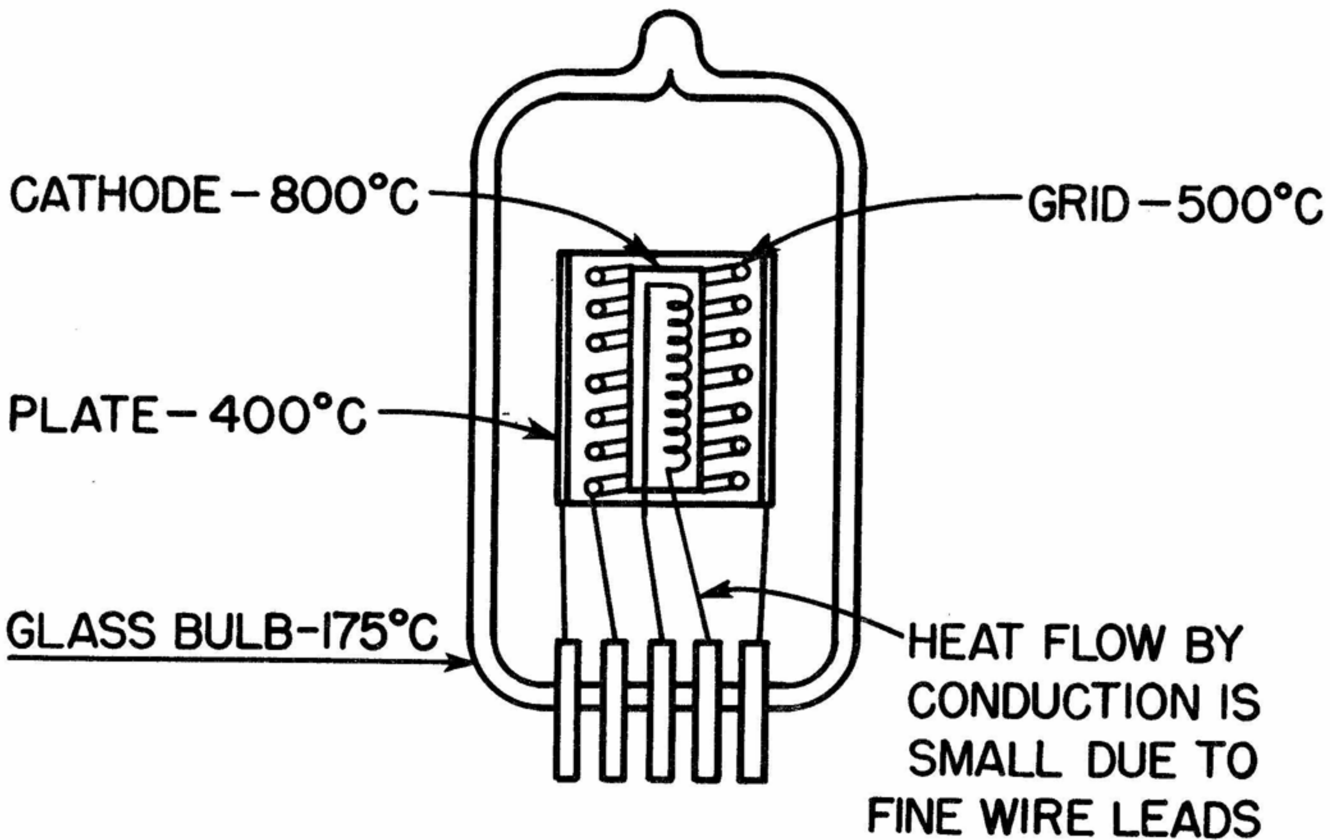


FIG. 5-1 HEAT TRANSFER IN A VACUUM TUBE

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 a vacuum 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 (see Fig. 5-2). Glass is a relatively poor heat conductor and temperature gradients will appear in the envelope adjacent to the upper and lower edges of the plate structure. It is desirable to cool vacuum 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 (see C.A.L. Report No. HF-1053-D-3). In general, only less than 10% of the heat can be removed through the leads of miniature tubes such as the 6AQ5, 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.

2. The Effects of Overheating and Temperature Limiting Factors

a. Electron Tube Glass

(1.) General

The upper temperatures of vacuum tubes are sometimes limited by the physical characteristics of the glass bulbs. Overheating of the glass can ultimately lead to serious malfunctioning. Receiving type tube bulbs are usually made of soda-lime-silicate glasses,

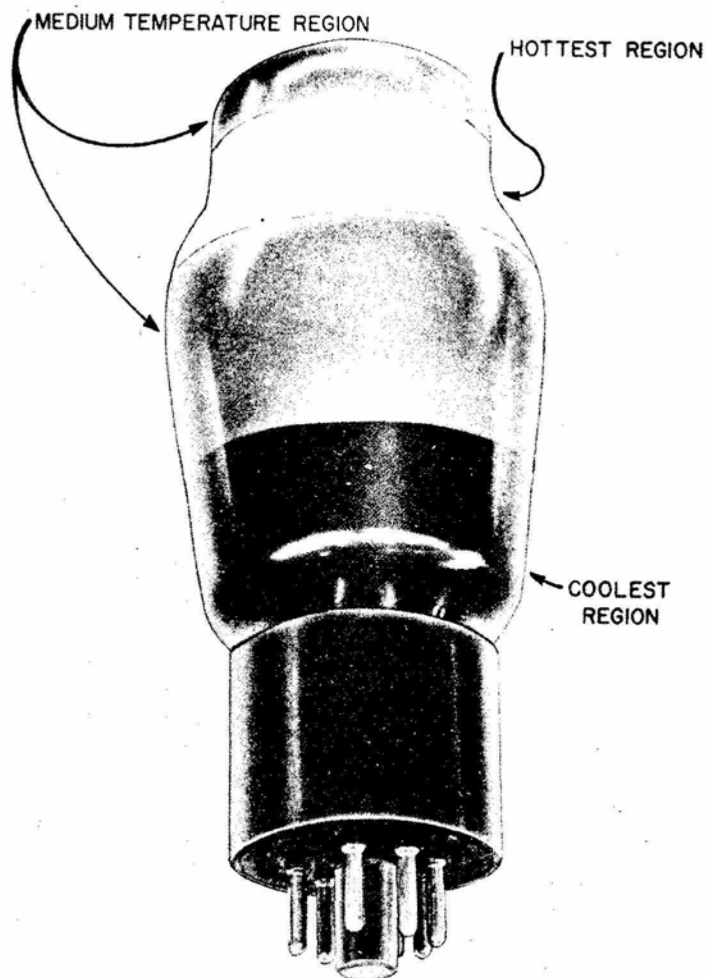


FIG. 5-2 TEMPERATURE DISTRIBUTION ON GLASS ENVELOPE OF A TYPICAL VACUUM TUBE

similar to Corning numbers 0080 and 0081 (see Table XI). These glasses have an annealing point in the neighborhood of 500°C., a strain point of 480°C. and a dielectric constant of 7.2. 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.

(2.) Electrical Properties

"The dielectric properties of glasses may be correlated not only with chemical composition, but also with the amorphous nature of glass itself. The structural arrangement of glasses is a three dimensional network, the skeleton of which is, for silicate glasses, made up largely of silica tetrahedrons, but which lacks the periodic order of the crystal lattice. This type of structure results in interstices or holes in which the modifying ions may be accommodated. These latter ions, and in particular the sodium ions, have some degree of mobility even at room temperature, so that they migrate throughout the structure from one interstice to another. With higher temperatures, this mobility increases greatly.

"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. It is believed, however, that while some of the sodium ions are relatively free to migrate, others are bound into the structure sufficiently strong to contribute little to the conductivity of the glass. It has also been found that the mobility of the sodium ion can be decreased by introducing other ions into the composition which tend to block migration, and thus decreases the conductivity.

"It may be mentioned that resistivity and other dielectric properties are subject to considerable variation with the thermal history of the glass, that is, with the degree of stabilization of the structure, and in addition resistivity measurements are also subject to polarization effects. The currents used are sufficiently low, a few micro-amperes per square cm., so that polarization does not occur to any appreciable degree.

"The dielectric losses of glasses as well as their conductivities correlate with the mobility of the sodium ions. Glasses with high conductivities are found generally with high losses as well and losses increase rapidly with temperature. For the best electrical glasses such as 96% silica, and silica glass (fused silica), the power-factor may drop to as low as 0.0002 to 0.0003 at room temperature.

TABLE XI.

VACUUM TUBE GLASS PROPERTIES

per Corning Glass Works

Glass Type	Corning Glass No.	Aux. Glass Part of Seal	Expansion	Metals Sealed to Glass for Lead
Lead-Alkali Silicate	0010 0120 *	same as bulb	89 ± 2	Dumet
Soda-Lime-Silicate	0080 0081 **	0010 0120		"
Barium-Alkali Silicate	9010	same as bulb		17% chrome iron alloy
Lead Boro-silicate (Nonex)	7720 ***	" " or tungsten or moly sealing glass		tungsten or moly
Kovar Sealing Glasses	7052 ***	same as bulb		Kovar or Fernico
UV Transmitting Boro-silicate	9730 9741	lead boro-silicate Kovar sealing glass		tungsten Kovar or Fernico
Heat Resisting Boro-silicate (hard to seal)	7740 (not used for tubes)	lead boro-silicate		tungsten
Electrical Boro-silicate	7760	" "		chromium iron alloy (Housekeeper seal)
96% Silica Glass and Silica Glass	7900-7911 7910-7912	graded seal 1)to lead boro-silicate 2)to Kovar		tungsten or Kovar or Fernico
Lead Glass High Expansion	8160	subminiature	91×10^{-7}	Dumet sealing

* occasionally used for octal tube
** commonly used for miniature and most octal tubes
*** hard glass miniature and high temp. subminiature

The emissivity of glass varies between .94 and .96 up to 250°C. Greater transmission occurs at higher temperatures. The absorption coefficient is the same at 250°C. as at room temperature.

"Soda-lime glass, with the highest dielectric loss of those listed in Table XI, has a loss nearly fifty times greater than that of the best glass known: a specially processed 96% silica glass. It may be pointed out that certain glasses which may have only moderately low losses at room temperature, may show superior qualities at high temperatures of 350 to 400°C. or higher.

"Although glass has always been recognized as possessing high dielectric strength, our understanding of the breakdown mechanism has developed only within the past decade or so. While the conductance in glass is an ionic phenomenon, true dielectric breakdown is apparently electronic. It is now believed that each solid insulating material has an intrinsic breakdown gradient and that this gradient is somewhat dependent upon temperature. This is particularly true above a critical temperature, which is characteristic of the material involved. When the intrinsic gradient is reached, electrons within the structure are accelerated by the field to a velocity which permits them to liberate additional electrons by collision, and thus to start an avalanche of electrons which constitutes breakdown.

"Another glass type failure is associated with 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 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 of the insulating material". (Ref. 1.)

The glasses with the best electrical characteristics are seldom used in vacuum tubes, due to undesirable chemical or mechanical properties. Vacuum 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. Table XI displays the pertinent glasses and the metals which may be sealed thereto. Note that 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.

(3) Mechanical Properties

The mechanical properties of vacuum 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 vacuum 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 vacuum tubes. Thus, any vacuum tube cooling device must not impose thermal or mechanical stresses on the fragile glass bulbs.

(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.

Table X presents the coefficients of transparency in the infrared region for several glass types at normal incidence for 1 mm. thickness. Note that glasses in general do not transmit radiation to any degree beyond 2.5 μ. With special glass, transmission can be extended to 5.0 μ. Table XII presents the calculated increments of plate dissipation for several glasses and plate temperatures. These data are only approximate and are presented to indicate the relative values as a function of plate temperature.

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 (see C.A.L. Report #HF-1053-D-3). 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.

TABLE XII.

CALCULATED INCREMENTS OF PLATE DISSIPATION FOR SEVERAL GLASSES AND PLATE TEMPERATURES

Plate Temp.	Max. Energy Distribution in microns (μ)	Energy Increment below 2.5μ	Energy Increment below 3.1μ	Energy Increment below 3.7μ	Energy Increment below 3.25μ	Energy Increment between 2.5&3.85μ	Calc. Plate Diss. Increment thru Bulb	
							Hard Glass*	Soda-lime Glass **
300°C.	5.1	.9%	3.6%	9.5%	4.6%	3.7%	2.5%	1.0%
350°C.	4.65	1.7%	5.7%	12.0%	7.0%	5.3%	3.9%	2.0%
400°C.	4.3	2.7%	8.0%	16.0%	9.5%	6.8%	6.2%	3.5%
450°C.	4.0	4.0%	11.0%	20.0%	13.0%	9.0%	8.7%	5.1%
500°C.	3.75	6.0%	14.0%	24.0%	16.0%	10.0%	11.0%	7.2%
550°C.	3.5	8.0%	17.5%	28.5%	20.0%	12.0%	15.0%	9.7%
600°C.	3.3	10.0%	20.5%	34.0%	24.0%	14.0%	18.0%	11.2%

* mostly subminiature types.

** most miniature and octal types.

In general, plate and bulb temperature of a glass vacuum 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 III 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 (see Fig. 5-3).

In general the thermal conductivity of glass is relatively low, being in the neighborhood of .03 watts/(in.^{2})($^\circ$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. Calculations have indicated that the 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 (see C.A.L. Report #HF-1053-D-3).

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 vacuum 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.

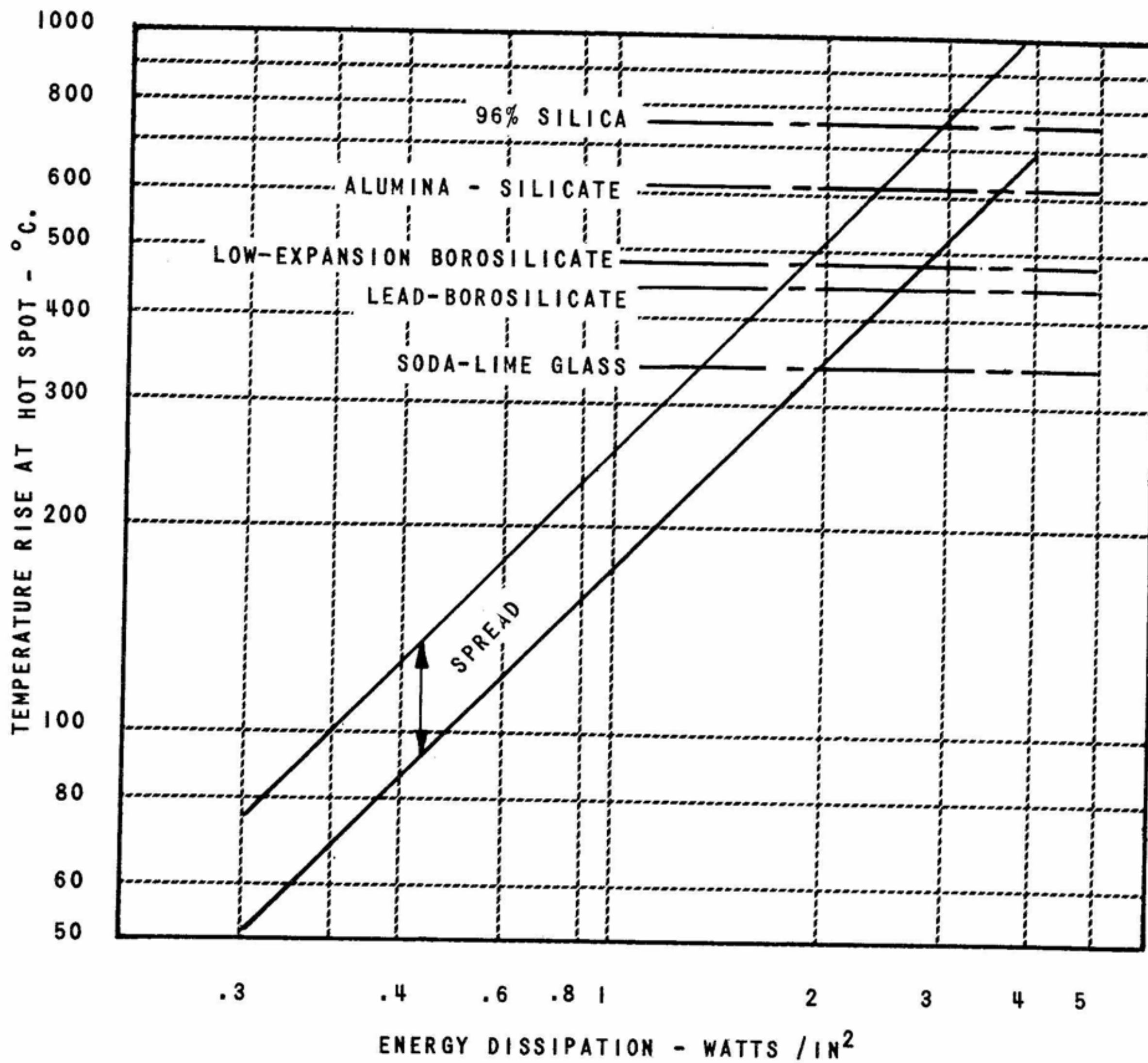


FIG. 5-3 TEMPERATURE RISE FOR GLASS TYPES AT HOT SPOT OF INCANDESCENT LAMPS AS A FUNCTION OF INTERNAL LOSS PER UNIT SURFACE AREA, LIMITING RISE FOR GLASS TYPES - 25°C AMBIENT TEMPERATURE.

(Courtesy Corning Glass Works)

(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.

b. Vacuum Tube Elements

(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 malfunctioning of circuitry.

(2.) Processing Limitations

Receiving type vacuum tubes are usually outgassed with the internal elements heated to approximately 500°C. and the glass heated to 250 to 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.

(3.) Getters

Vacuum 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.

(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 (see C.A.L. Report #HF-1053-D-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 is not mentioned. Unfortunately, the cathode temperature can vary significantly over a wide range of environmental temperature. Findings to date indicate that a miniature triode, such as a 6C4, 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 (Ref. 57).

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.

(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 (see C.A.L. Report #HF-1053-D-3. Since the plate in a vacuum tube is radiation cooled, the temperature of the plate is directly related to the net temperature of its surroundings. Thus, the life a given tube is related directly to the temperature of its elements, as mentioned earlier, not 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.

(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 is drastically reduced through decreased emission, evaporation of getter and emitting materials, increased interelectrode 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 vacuum 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 (see C.A.L. Report #HF-1053-D-3). The cooling of a tube is the most important consideration in its mounting.

3. Vacuum 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, vacuum 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 vacuum tube reliability more than any other single factor.

It is believed that when tubes must be operated at high environmental temperatures or under conditions which will result in abnormal plate and envelope temperatures, that derating is in order. Exact derating values remain to be determined. However, since plate temperature is an index of the thermal condition of a tube, it is recommended that tubes be derated on the basis of plate temperature. The "Thermatron" should prove a useful aid in such determinations (see C.A.L. Report #HF-1053-D-3).

The development of miniature and subminiature tubes has led to a large reduction in envelope surface area and a large increase in the rated unit heat dissipation. Thus, the maximum envelope temperatures were increased greatly over those of the conventional tube types. Table XIII is representative for several bulb types.

TABLE XIII.

UNIT HEAT DISSIPATIONS OF TYPICAL ENVELOPE STRUCTURES

Bulb Type	Octal T-9	Miniature T-5½	Subminiature T-3
Bulb area, sq. in.	10.5	4.1	1.7
Maximum rated dissipation of tube in watts	18.7	16.8	7.8
Unit heat dissipation, watts/sq.in.	1.78	4.1	4.6
Unit heat dissipation ratio (with respect to T-3)	.44	.85	1.
Hot spot bulb temperature in free air at 23°C.	160°C.	225°C.	280°C.

Typical envelope temperatures for sea level and at 23°C. ambient temperature conditions are presented in Table XIV as an indication of the temperatures obtained with a single tube in free air.

TABLE XIV.

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 XV.

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.

Figure 5-4 presents the relative temperatures of various tube types. Note that at maximum ratings, excessive vacuum tube temperatures can be obtained even in free air. When tubes are used in equipment, these free air ambient conditions seldom

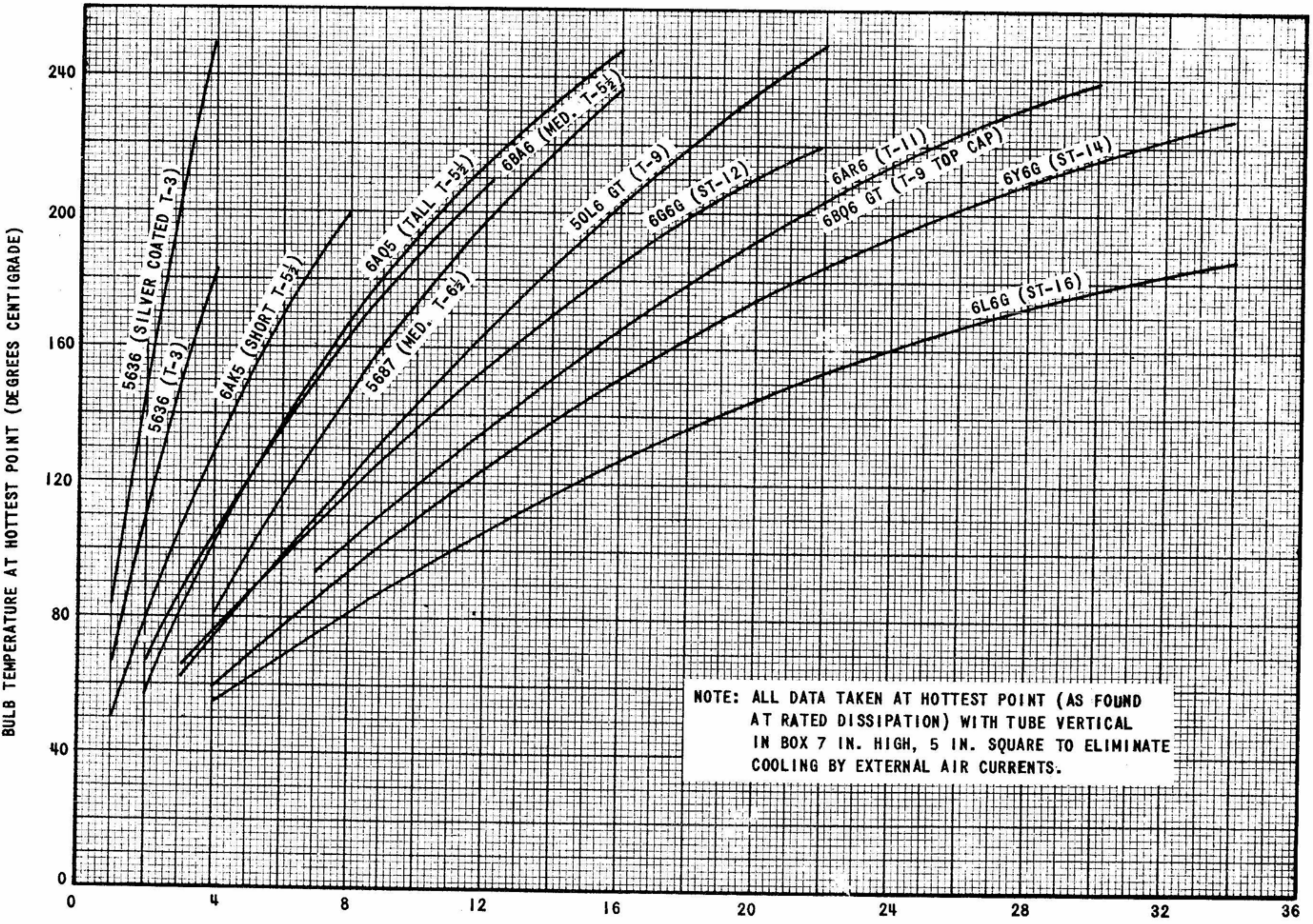


FIG. 5-4 RELATIVE TEMPERATURES OF TYPICAL TUBES

(Courtesy Tung-Sol Electric Corp.)

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. C.A.L. Report #HF-1053-D-3 discusses plate temperature measurement and C.A.L. Report #HF-845-D-2 (NAVSHIPS 900,187) - "Manual of Standard Temperature Measuring Techniques, Units and Terminology" presents methods for bulb temperature measurements.

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 it should not be used to increase the internal element dissipation of tubes beyond their normal rated values. Excessive dissipation is hazardous and over-cooling should not be used in order to exceed the maximum rated power level of any tube.

4. Methods of Cooling Vacuum Tubes

a. Unshielded Tubes

The major mode of heat transfer from a bare vacuum tube in free air is radiation. A smaller 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 an air-tight container, then free convection becomes ineffective and heat will be transferred by gaseous conduction, but still to a smaller degree than that by radiation.

Vacuum tubes larger than the subminiature types can be cooled satisfactorily without shields if the ambient 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 1½ 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.

b. Shields for Cooling Tubes

(1.) General

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. One method which has found some use is to apply silicone grease between the tube and the shield. Unfortunately, this method is not usually suited to the maintenance techniques of the Armed Services. In general, the most practical method is to provide some flexibility in the shield 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. If heat transfer by radiation to the surroundings from the shield is desired, then the same dull surface should be used on the outer surface of the shield. On the other hand, if temperature sensitive parts and other tubes constitute the surrounding objects (in which case radiant heat transfer to these parts is not desired) then the shield's outer surface should be highly polished. Thus, the surrounding objects influence the design of the shield. In general, it is not recommended that radiant energy be deliberately expended inside an electronic enclosure. The heat is radiated from the source and dispersed to other parts in an uncontrolled fashion.

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 should be of metal. Nothing is gained by mounting tube shields on, for example, a plastic chassis, or on materials of low thermal conductivity. Such materials act as thermal insulators and it is easily possible to overheat a well shielded vacuum tube, even when it is operated well within its dissipation ratings. Ideally, for maximum heat transfer, the tube shield should be soldered, brazed or bonded to the metal surface to obtain a near perfect contact. Riveting or bolting a shield to a metal surface leaves a thin air gap which constitutes a high thermal resistance; probably as much as the resistance of the shield itself. For effective heat removal, this gap must be minimized and preferably eliminated. A poor surface contact may cause a shielded tube to operate hotter than a bare tube, even though other thermal considerations are incorporated in the design. It is not advisable to use a tube shield which is not thermally bonded to a cooler metal surface.

(2.) Miniature Tube Shields

The standard JAN miniature tube shield, A of Figure 5-5, 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.

The following miniature tube shield data has been determined by the "Thermatron" evaluation of tube shields. The techniques involved are discussed in C.A.L. Report #HF-1053-D-3, and only the findings are presented herein. The order of thermal merit of the various miniature tube shields is presented by Table XVI. The lower thermal rating numbers indicate the better shields. All shields were internally blackened to provide improved cooling. The net thermal resistance has been defined as the thermal resistance measured from the inside of the bulb through the shield to the chassis by all modes of heat transfer.

TABLE XVI.

THERMAL RATING OF T-5½ SHIELDS

Thermal Rating	Net Thermal Resistance in °C/Watt			Shield Type Ltr. Designation
	2" Bulb at 10w.	1½" Bulb at 9w.	1" Bulb at 9w.	
1	8.7	9.2	14.6	N
2	9.2	9.8	15.0	K
3	10.1	10.0	15.8	B-L (* **)
4	14.0	---	---	C-L***
5	14.1	14.5	18.8	G-E-M
6	17.2	---	---	F
7	18.4	---	---	J
8	19.5	18.0	23.0	A-D-H

* Same plate temperature as tube in free air.
** With special base fabricated by N.E.L.
*** When a standard JAN base is used, the thermal rating of shield L is 4.

Figures 5-5, 5-6 and 5-7 identify the shields. Shields, type N-K-B-L are superior to the other shields, since they have the lower thermal resistances and conduct the rejected heat into the chassis to the greatest degree. Shield E is actually a tube holder and conducts less heat to the chassis than the other shields. Shield G with the "L" or "M" insert has a resistance at 10.2°C/w. However, any of the shields can be used with thermal derating based upon the net thermal resistance. The derating may be based also upon the plate temperatures and the bulb temperatures shown in C.A.L. Report #HF-1053-D-3. It is strongly recommended that all tube shields be internally blackened. Plate temperature derating values for the thermally inferior shields can also be obtained from C.A.L. Report #HF-1053-D-3. Note that the thermal resistances for the 1½ in. bulbs are in some instances lower than those for the 2 in. bulbs. This is apparently due to the shorter heat flow paths in the bulbs and shields. The 1 in. bulbs have a smaller contact area.

Severe deterioration of tube shield plating, usually cadmium, has been encountered at shield temperatures exceeding 150°C. Care must be exercised in selecting black finishes or paints to make sure that the finish can withstand the peak anticipated temperatures.

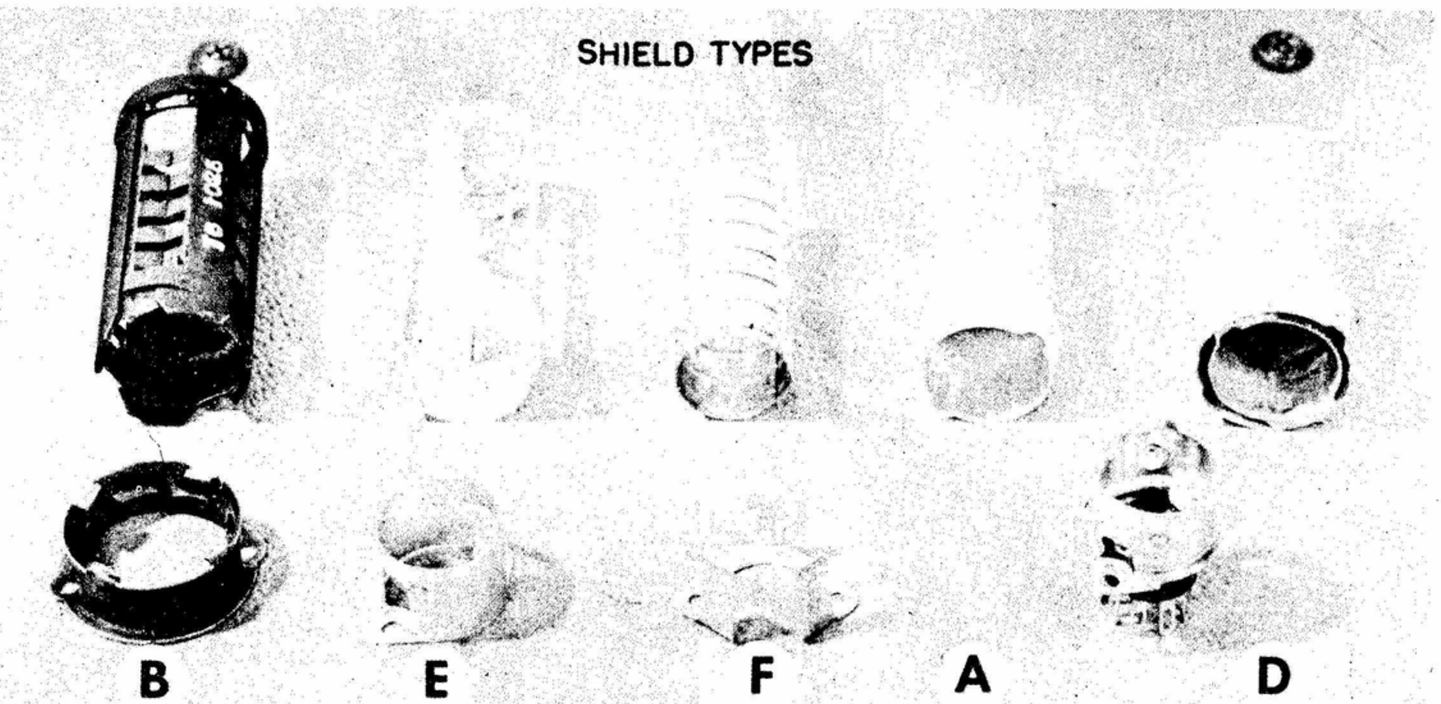


FIG. 5-5 IDENTIFICATION OF MINIATURE TUBE SHIELDS

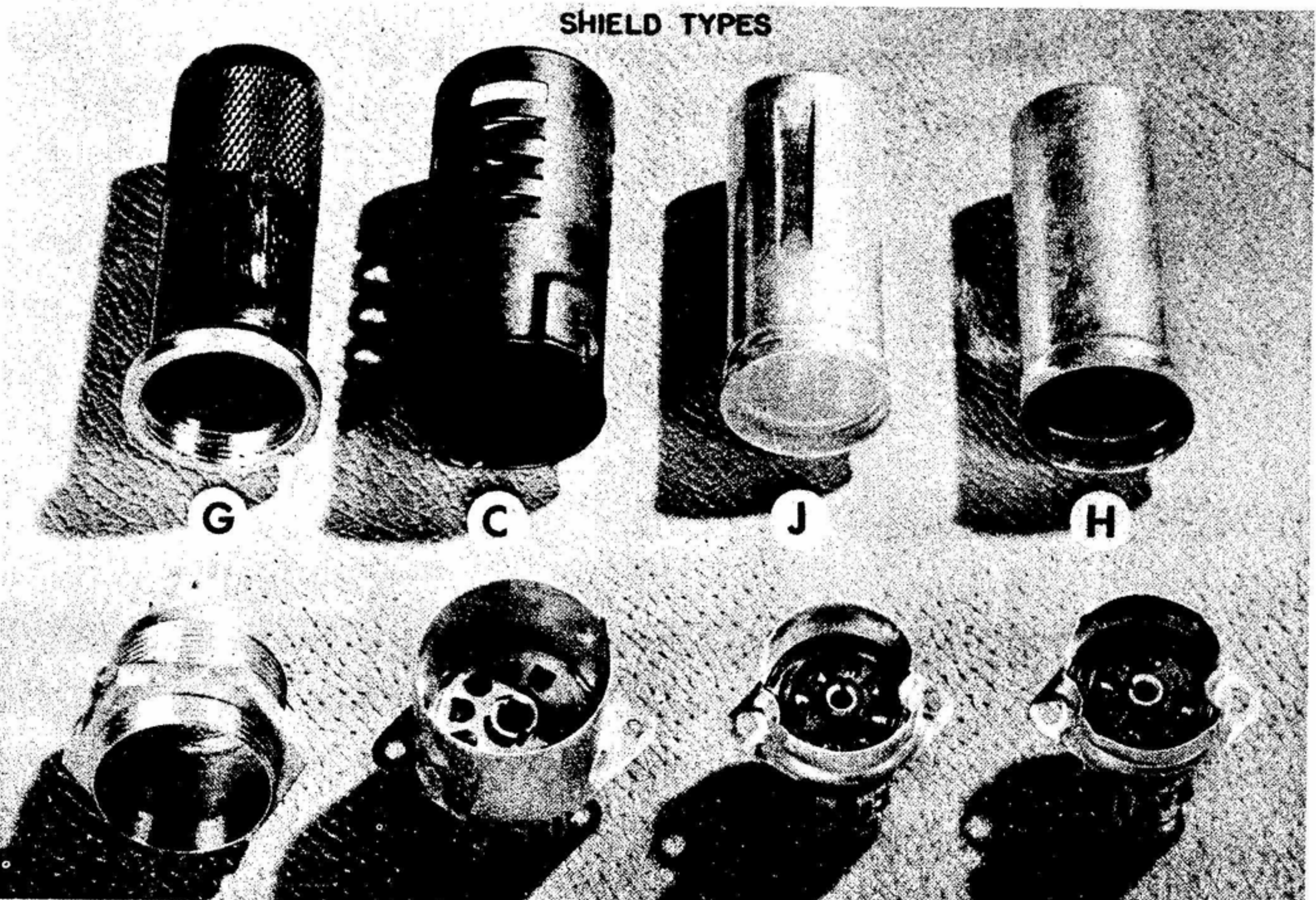


FIG. 5-6 IDENTIFICATION OF MINIATURE TUBE SHIELDS

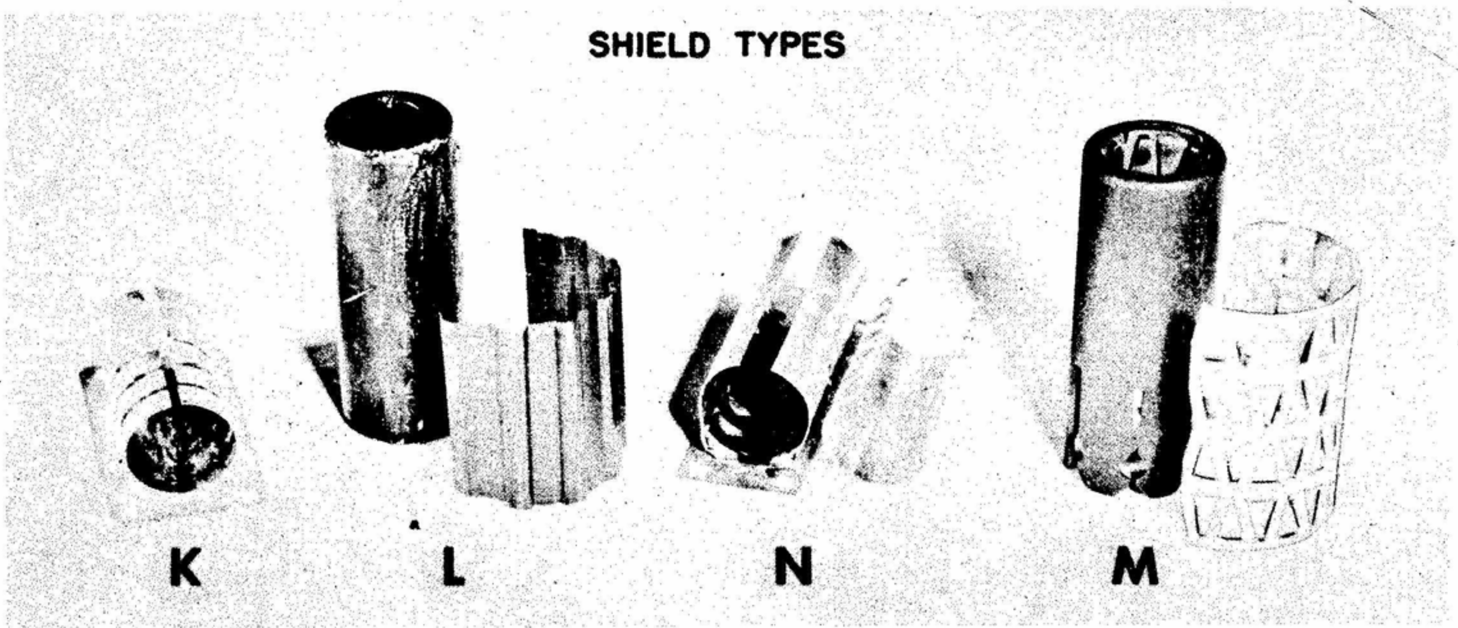


FIG. 5-7 IDENTIFICATION OF MINIATURE TUBE SHIELDS

Special and precious metals are not recommended for tube shields. The thermal gains over steel, aluminum, copper, or copper alloy shields will be insignificant compared to the cost of these expensive materials.

For cylindrical parts surrounded by closely-spaced, blackened shields, the heat-transfer rate may be estimated by the following two equations:

Conduction:

$$q_c = \frac{2.729\ k_{air}\ L\ (t_p - t_s)}{\log_{10} \frac{D_s}{D_p}} \qquad (49) \qquad (D.E.)$$

where:

k is the thermal conductivity of the air or gas within the space.

t_p is the estimated mean temperature of the part.

t_s is the estimated mean temperature of the shield.

D_s and D_p are the diameters of the shield and part respectively.

L is the length of the part.

Radiation:

$$q_r = 0.173\ F_e\ F_a\ A_p \left[\left(\frac{T_p}{100}\right)^4 - \left(\frac{T_s}{100}\right)^4\right] \qquad (50) \qquad (D.E.)$$

where:

F_e is the emissivity factor having a value between 0.85 and 0.95 for most electronic parts including electronic tubes, as long as any metal parts are not bright or polished.

F_a is the configuration factor, being unity for the configuration of a cylinder enclosing a part.

A_p is the area of the part.

T_p and T_s are the absolute temperatures of the part and shield surface respectively.

The total heat-transfer rate is the sum of q_c and q_r.

The tests of Reference (25) show, as could be predicted from heat-transfer theory, that increasing the tube and shield temperatures the same amount results in greater heat transfer by radiation while the heat transfer by gaseous conduction remains the same. Thus, for the same heat dissipation, the temperature difference between the part and the shield is less at elevated temperatures than at lower temperatures. This is due to the increased effectiveness of radiant heat transfer in accordance with the fourth powers of the temperatures.

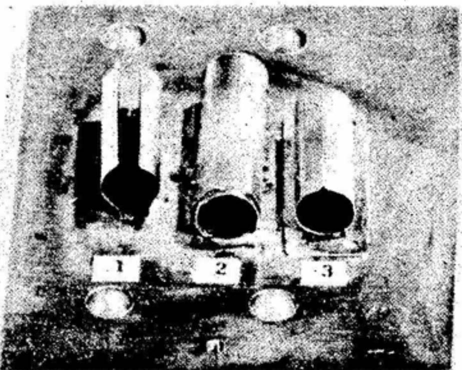
Detailed data on temperature gradients within and on the surfaces of the various tube shields is presented in C.A.L. Report #HF-1053-D-3. This information should be of assistance to those interested in the design of tube shields.

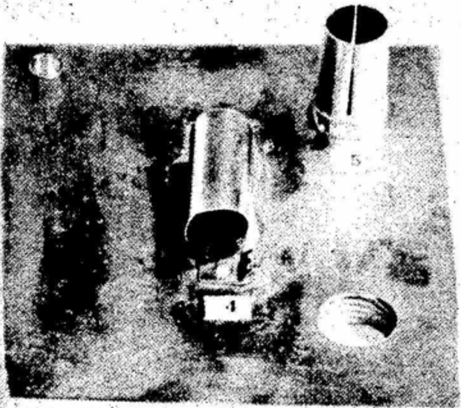
(3.) Subminiature Tube Shields

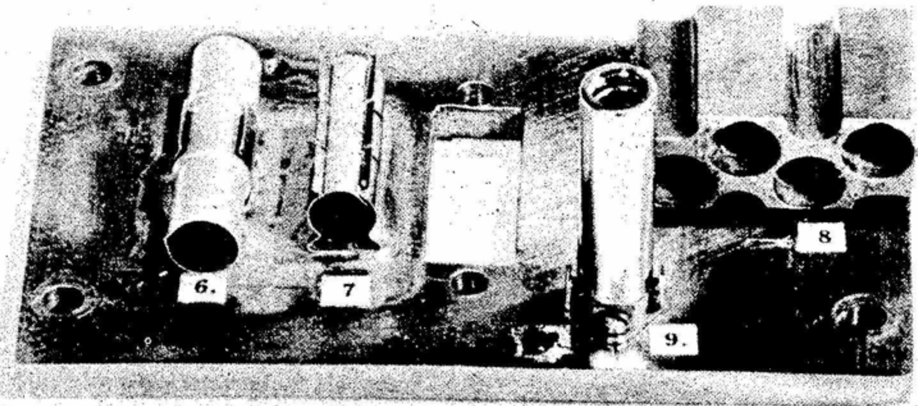
The majority of subminiature tubes are used in radio frequency and voltage amplifier applications wherein about two to three watts per tube are dissipated. At this power level, a single unshielded tube operating in free air at 25°C. can attain an envelope temperature of 130°C. at thermal equilibrium. Approximately 40% of the heat is removed by radiation while the greatest part of the remaining 60% of the heat is removed by free convection. When this same tube is placed in a subassembly with other similar tubes, mutual heating by radiation and convection is increased, and the tube envelope temperature can rise to 225°C. at equilibrium. It is possible to alleviate this condition by using tube shields designed to remove heat from the tube envelope and transfer it into the chassis by metallic conduction.

An assortment of subminiature tube shields designed for conduction cooling have recently become available. There are four general types of subminiature tube shields: the wrap-around shield, the fuse clip type shield, the machined metal shield and the solid metal block type shield. The selection of a given shield depends upon the particular requirements of the electronic subassembly involved.

(a) The wrap-around shield (Shield #4, Fig. 5-8) is the most commonly used in current electronic equipment. The shield consists of a cylinder of springy metal which is formed to fit and wrap around a subminiature tube envelope. Heat is transferred from the glass envelope of the tube by radiation, direct conduction, and gaseous conduction to the shield. The shield in turn forms a conduction path of low thermal resistance to the chassis.







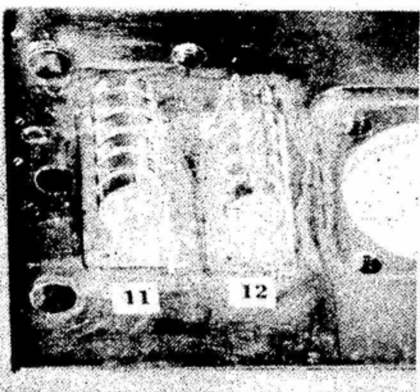


FIG. 5-8 IDENTIFICATION OF SUBMINIATURE TUBE SHIELDS

(b) Another type of subminiature tube mounting utilizes a fuse clip type configuration (Shields #6 & #7, Fig. 5-8) in which a spring metal clip is used to hold the tube in position. The fuse clip is usually riveted to the chassis or support. Some fuse clip mountings are long and cover a good part of the tube surface, thus providing a good conductive heat path away from the tube envelope. Small fuse clips have also been used to hold the tube in a favorable position for free convection and radiation cooling.

(c) The cylindrical shield consists of an aluminum tube shield machined out of a slotted aluminum tube with threads on its base so that it may be screwed into a threaded chassis (See Shield #5, Fig. 5-8).

(d) An alternate method of mounting subminiature tubes has been to insert the tubes in a drilled metal block, the holes (Shield #8, Fig. 5-8) being slightly larger in diameter than the outside diameter of the tubes. Metal tube blocks of this type have been built as an integral part of the equipment cases. Various methods have been used to hold the tubes in the block. In one instance, silastic type rubber was used in the form of a ring around the top and the bottom of the tube envelope to form a shock mounting for the tube. With such a mounting, the primary heat transfer modes from the tube envelope are by radiation and gaseous conduction to the metal block which, in turn, provides a metallic conduction path to the surface of the equipment through the assembly case. However, this arrangement is thermally inferior to another technique which involves wrapping the tube with corrugated aluminum, silver or copper inserts prior to installation into the metal tube block. The air gaps are reduced and conduction is increased.

The following subminiature tube shield data have been determined by a careful evaluation of the thermal properties of tube shields. Since all shields were improved by internal blackening, these ratings are for blackened shields. The lower thermal rating numbers indicate the better shields.

TABLE XVII.

THERMAL RATINGS OF SUBMINIATURE TUBE SHIELDS

Thermal Rating	Shield Type Number Designation (See Fig. 5-8)
1	2
2	1, 7, 8, 11
3	12, 3, 13
4	4, 6
5	5
6	9

Note: Shield #9 permitted excessive temperature rise and is not recommended.

Shield #5 is rated fifth because of large bulb temperature gradients.

Fig. 5-8 identifies the tube shields. Shield #2 is slightly superior to all others in plate temperature reduction. However, this may in part be due to its somewhat greater length. There is not too much thermal difference between shields 1, 2, 7, 8 and 11. The cooling achieved with shields #12 and 3 is somewhat less than that achieved by the higher rated shields. Even so, all of the shields with a thermal rating at 3 (18°C/watt Net Thermal Resistance) or lower are thermally adequate for most applications. Table XVIII presents the detailed thermal data in these matters. Fig. 5-9a shows a subminiature tube holder, identified as Shield #13. The embedment of tubes in silastic or plastic is not recommended. (See Fig. 5-9b.) This technique can only be used if the power dissipation is low.

Any of the above shields, other than #9, can be used with suitable thermal derating. The derating should be based primarily upon the plate temperatures shown in C.A.L. Report #HF-1053-D-3.

TABLE XVIII.

THERMAL CHARACTERISTICS OF T-3 SUBMINIATURE TUBE SHIELDS

DESCRIPTION			*Net Thermal Resistance at 7.0 watts in °C/watt	TEMPERATURES - °C. (Input to Tube - 7.5 Watts)				
Identification No.	Shield	Finish		Amb.	Chassis	Tube Base	Tube Plate	Thermal Merit
1	3/4" Long Spring Clip Type-Silver	Natural Blackened	16.7	30 30	36 33	163 158	503 495	(2)
2	1-5/16" Long (2 piece) Wrap-around soft silver with spring clip	Natural Blackened	14.0	30 30	34 33	140 143	504 492	(1)
3	3/4" long Sleeve-type - silver	Natural Blackened	17.9	30 30	36 35	147 173	492 492	(3)
4	3/4" long Wrap-around - bronze	Natural Blackened	19.1	30 30	34 35	171 171	493 491	(4)
5	3/4" long Slotted Cylinder - Vertical screw base - aluminum without insert	Natural Blackened	19.2	30 30	36 37	183 181	503 495	(5)
6	1-5/8" long - 2 piece Aluminum wrap-around & bronze fuse clip holder	Natural Blackened	19.3	30 30	35 36	183 179	514 500	(4)
7	1-3/16" long - 2 piece Silver wrap-around & steel fuse clip type holder - full length	Natural Blackened	17.2	31 31	34 35	171 168	497 494	(2)
8	Drilled aluminum block with silver wrap insert	Natural Blackened	16.8	31 31	37 36	176 162	504 492	(2)
9	Shield & socket assem. similar to Jan-Min.Std.	Natural	30+	30 Excessive temperature	70	155	540	N.G.
10	Thermatron in Free Air No shield	Vertical	28+	30	--	195	530	--
11	Spring clip - slotted and silver plated-1" lg.	Natural Blackened	17.0	30 30	32 33	180 169	503 409	(2)
12	Open spring clip - slotted and silver plated - 1" long	Natural Blackened	18.1	31 31	34 35	181 175	502 490	(3)
13	3/4" long Spring Clip with sponge copper	Natural Blackened	18.2	--	--	--	--	(3) (3)

*Includes contact resistance when soldered to chassis.

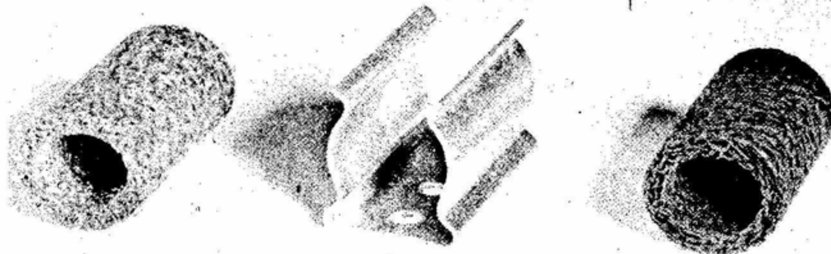


FIG. 5-9a SUBMINIATURE TUBE HOLDER
SHIELD NO. 13

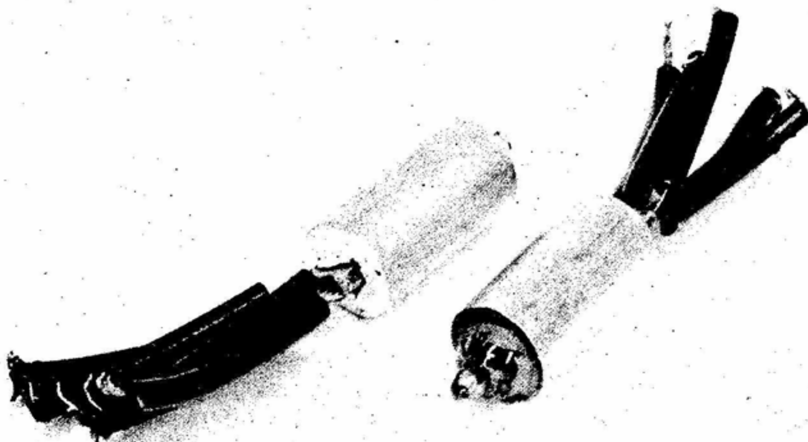


FIG. 5-9b SUBMINIATURE TUBES EMBEDDED IN SILASTIC

Special and precious metals are not recommended for subminiature tube shields. The thermal gains over slightly thicker sections of aluminum, copper, or copper alloys are insignificant compared to the temperature gradients which will exist elsewhere in an equipment.

Wrap-around shields such as shield #4 can be roughly designed based upon

$$\Delta t = \frac{3\pi Pc}{4kLx} \qquad (51)\ (D.E.)$$

where: (see Fig. 5-10)

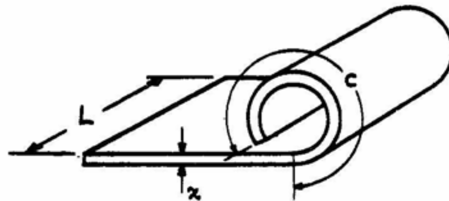


Figure 5-10.

c = that part of the circumference of the tube in contact with the shield.

k = the coefficient of thermal conductivity with respect to silver. (Silver = 1, Copper = .92, Steel = .1, Beryllium Copper = .25.)

L = the length of the shield in inches.

P = the dissipated power in watts.

x = the thickness of the shield in inches.

Δt = the temperature gradient from the edge to the base of the shield.

Fig. 5-11 is a tube shield temperature differential chart. The gradients produced in a wrap-around tube shield for a tube dissipating 4.0 watts (3.75 watts to the shield) are shown as a function the material and its thickness.

(4.) Octal Tube Shields

The selection of shields for octal tubes is somewhat limited, primarily because metal tubes which are inherently self-shielded are usually available as equivalents to the glass

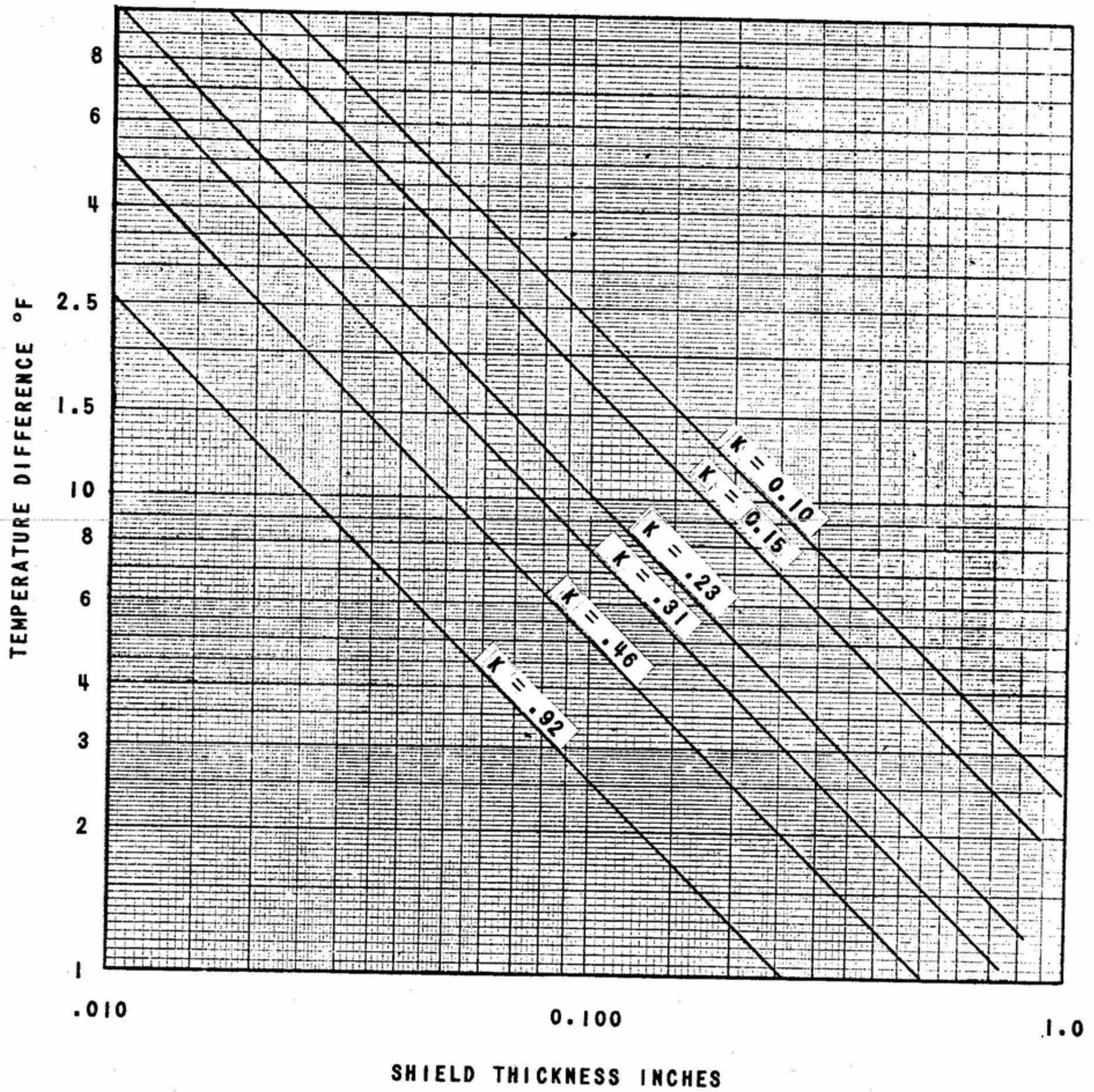


FIG. 5-11 TUBE SHIELD TEMPERATURE DIFFERENTIAL CHART
1-1/4" LENGTH, 0.2" RADIUS $\frac{3\pi}{2}$ ANGLE
3.75 WATTS TO SHIELD

types. Electrical shielding of glass octal tubes has been occasionally provided by shields such as that presented on Fig. 5-12. This shield is made of cardboard with a metal foil lining and is almost the ultimate in inferior tube shields. The shiny metal foil reflects radiation back towards the tube and the cardboard provides good thermal insulation. This is an extremely poor shield, thermally, and is not recommended.

The shield shown on Fig. 5-13 has been studied. It provided essentially the same plate temperatures as those obtained in free air under identical conditions. Consequently, it is considered to be a thermally efficient shield.

NOTE: As a general rule most tube shields increase plate temperatures over those obtained in free air under similar conditions. Only the best shields have exhibited thermal resistances sufficiently low to produce plate temperatures of the order of those obtained in free air. Thus, any shield which does not cause the plate temperatures to rise over that obtained in free air is excellent.

(5.) Optimum Cooling

The point of diminishing returns of cooling vacuum tubes by natural means is of interest. The heat flow from the average well-shielded tube can be analyzed as shown by Fig. 5-14.

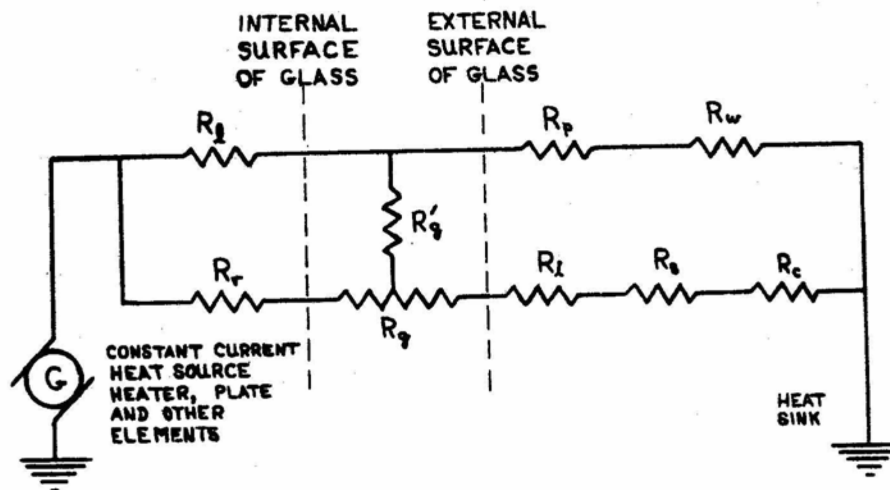


Figure 5-14.
SIMPLIFIED THERMAL ANALOGY OF A VACUUM TUBE

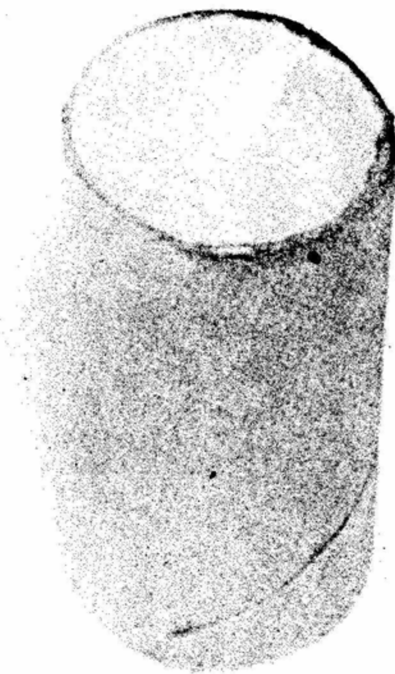
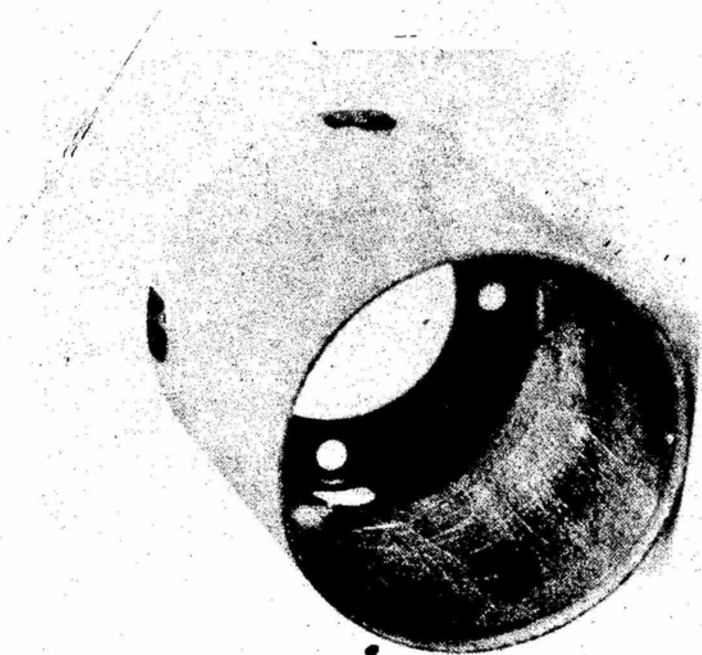


FIG. 5-12 CARDBOARD SHIELD



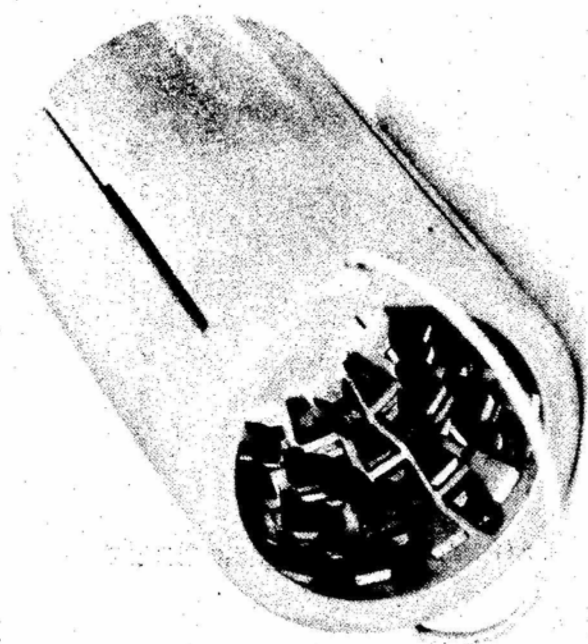


FIG. 5-13 OCTAL SHIELD

where:

R_l = the thermal resistance due to conduction through leads from the tube elements

R_r = the thermal resistance due to radiation through the vacuum

R_g = the thermal resistance of the glass. This includes the attenuation of radiant energy and the thermal conduction through and around the glass. R_g' represents the resistances to heat transfer in the glass from the base to the bulb opposite the plate.

R_p = the thermal resistance of the pins, socket connectors, and the resistance of the joints at the socket

R_i = the thermal resistance of the contact between the shield and the surface of the glass. This includes interface resistance to heat transfer by radiation, conduction and gaseous conduction

R_s = the thermal resistance of the shield

R_w = the thermal resistance of the wiring

R_c = the thermal resistance of the chassis to the sink. This includes the resistance to heat transfer due to conduction, radiation and convection from the chassis of subchassis to the sink.

Resistances R_l, R_r, R_g, and R_g' are built into the tube. Even so, the equipment designer has some control over R_r, since it can be influenced by the external resistances. The optimum cooling or practical point of diminishing returns can be achieved when the thermal resistances external to the tube are small compared to the internal resistances. R_l, R_p, R_g' and R_w are usually insignificant compared to the other resistances; i.e., less than 5 to 10 percent of the heat in a tube is removed through the leads. Thus, R_r, R_g, R_i, R_s and R_c are the important and smaller thermal resistances. R_r within the tube is a function of the difference of the fourth powers of the absolute temperatures of the plate and the glass bulb, and is also a linear function of the absorptivity of inner surface of the shield. Consequently, the inner surface of the shield must have a high absorptivity. Blackening is a practical means of producing this desired surface condition.

R_g and R_g' are within the glass and are usually of small order. Since the temperature drop across the glass bulb is less than 2 to 4°C, R_g is also insignificant. Therefore, the electronic equipment designer can only control R_i, R_s and R_c and indirectly control R_r. If the important resistances are lumped, then the analogy can be further simplified as shown by Fig. 5-15.

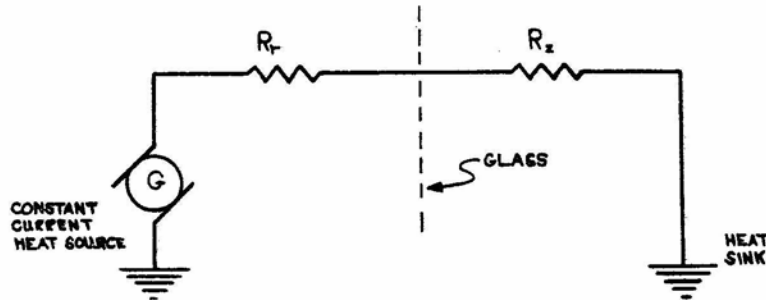


Figure 5-15.
SIGNIFICANT THERMAL RESISTANCES OF A VACUUM TUBE

where:

$R_z = R_i + R_s + R_c$ or the net thermal resistance which can be directly controlled by the equipment designer.

Consequently, the practical point of diminishing returns in vacuum tube cooling is reached when R_z is significantly smaller than R_r. Note that even though R_r is within the tube, its value is indirectly influenced by the temperatures produced by R_z. These relationships can be demonstrated by the application of actual measured values.

For example, for a subminiature tube dissipating 7.5 watts, the heat transmitted from the plate through R_r would be approximately 7.0 watts. The remaining .5 watt is transferred through the leads and radiated from the base of the bulb. Assume unity or equivalent emissivity in all instances.

Case a - For well cooled tube

Plate temperature	=	400°C.
Av. Bulb temperature	=	125°C.
Sink temperature	=	40°C.

Then

$$R_r = \frac{400 - 125}{7} = 39°C/watt$$

$$R_z = \frac{125 - 40}{7} = 12°C/watt$$

Case b - For a tube with acceptable cooling

Plate temperature	=	450°C.
Av. Bulb temperature	=	175°C.
Sink temperature	=	40°C.

Then

$$R_r = \frac{450 - 175}{7} = 39°C/watt$$

$$R_z = \frac{175 - 40}{7} = 19°C/watt$$

Case c - For a poorly cooled tube

Plate temperature	=	500°C.
Av. Bulb temperature	=	250°C.
Sink temperature	=	40°C.

Then

$$R_r = \frac{500 - 250}{7} = 36°C/watt$$

$$R_z = \frac{250 - 40}{7} = 30°C/watt$$

Note that R_r varies from 36 to 39°C/watt and that R_z ranges from 12°C/watt for the well cooled tube to 30°C/watt for a poorly cooled tube. The approximate ratios of $\frac{R_z}{R_r}$ are 1/3 for case a, 1/2 for case b, and unity for case c.

Consequently, it appears that the practical point of diminishing returns in cooling a subminiature tube is reached when the external thermal resistance is of the order of 1/3 the internal thermal resistance of the tube. Acceptable cooling is achieved when the external thermal resistance is in the neighborhood of 1/2 the internal thermal resistance and minimum permissible cooling occurs when the external thermal resistance equals the internal thermal resistance.

In a like fashion miniature tubes can be analyzed.

Case d - For an exceptionally well-cooled tube

Dissipation = 10.5 watts. Assume .5 watts transferred through the leads and the base.

Plate temperature	=	350°C.
Av. Bulb temperature	=	85°C.
Sink temperature	=	40°C.

Then

$$R_r = \frac{350 - 85}{10} = 26.5^{\circ}C/watt$$

$$R_z = \frac{85 - 40}{10} = 4.5^{\circ}C/watt$$

Ratio

$$\frac{R_z}{R_r} = \frac{4.5}{26.5} = 1/6 \text{ approx.}$$

Case e - For good cooling

Plate temperature = 375°C.
Av. Bulb temperature = 125°C.
Sink temperature = 40°C.

Then

$$R_r = \frac{375 - 125}{10} = 25^{\circ}C/watt$$

$$R_z = \frac{125 - 40}{10} = 8.5^{\circ}C/watt$$

Ratio

$$\frac{R_z}{R_r} = \text{approx. } 1/3$$

Case f - For acceptable cooling

Plate temperature = 400°C.
Av. Bulb temperature = 175°C.
Sink temperature = 40°C.

Then

$$R_r = \frac{400 - 175}{10} = 22.5^{\circ}C/watt$$

$$R_z = \frac{175 - 40}{10} = 13.5^{\circ}C/watt$$

The ratio of $\frac{R_z}{R_r}$ = approx. .6

Case g - For poor cooling

Plate temperature = 425°C.
Av. Bulb temperature = 225°C.
Sink temperature = 40°C.

Then

$$R_r = \frac{425 - 225}{10} = 20^{\circ}\text{C/watt}$$

$$R_z = \frac{225 - 40}{10} = 18.5^{\circ}\text{C/watt}$$

The ratio of $\frac{R_z}{R_r}$ = approx. unity

Note that the above ratios for cases e, f, and g are identical to those of cases a, b, and c.

As in the case of the subminiature tubes, miniature tubes can be considered to be well cooled if the ratios of the external to internal thermal resistance are of the order of 1/3. Acceptable cooling is achieved when the external thermal resistance is approximately 1/2 the internal thermal resistance and minimum permissible cooling occurs when the external thermal resistance equals the internal thermal resistance.

Therefore, it is concluded that the above "rules of thumb" are of practical value in determining the point of diminishing returns in cooling most miniature and subminiature vacuum tubes. However, it is possible that unknown exceptions may exist. All of the temperature values used above are those actually measured at this Laboratory. Thus, it appears that in case d, wherein the ratio of $\frac{R_z}{R_r}$ was 1/6, that the tube was cooled beyond the practical point of diminishing returns for this particular operating condition. Incidentally, this cooling situation was obtained utilizing tube shield "B". Further, if the bulb, shield and chassis temperatures and the heat dissipation are known, the designer can determine R_z, the external thermal resistance. R_z can be used as a figure of merit. In general, it should range from 4 to 15°C/watt for acceptable cooling.

The following are typical values of R_r which can be used by designers to determine the degree of cooling if the plate temperature is not known:

TABLE XIX

TYPICAL VALUES OF R_r

FOR TUBES OPERATING NEAR THEIR MAXIMUM DISSIPATION

Tube Type	R_r in ^{o}C/watt (Mean Measured Value)	
T-5½ Miniature (6AQ5)	25 @ 12 watts	35 @ 6 watts
T-6½ Miniature (12BY7)	21 @ 12 watts	30 @ 6 watts
T-3 Subminiature (5902)	37 @ 7 watts	52 @ 3 watts
T-9 Octal (6L6GBY)	12 @ 24 watts	15 @ 10 watts

Note: R_r is not a linear value and varies as the fourth power of the absolute temperatures. Consequently, R_r values can be much larger than indicated at lower plate dissipations.

R_r values for other tubes of similar types can be approximated as a percentage of the relative areas of their plates and bulbs compared to those listed above.

C. SEMI-CONDUCTOR DEVICES

1. General

The application of semi-conductor elements instead of vacuum tubes decreases heat generated in comparable equipments. The electrical efficiency of these devices is usually considerably higher than that of vacuum tubes and consequently the cooling problem may be eased. However, the circuit application, the power level, and the proximity of other heat producing parts can lead to excessive temperatures and subsequent impaired reliability.

Semi-conductor devices are inherently temperature-sensitive and their electronic performance is greatly influenced by their temperatures. Further, the operating temperature ranges are usually limited compared to those of vacuum tubes. Thus, extreme care must be utilized in the thermal design of electronic equipment which incorporates semi-conductor devices.

The following information is that which is available as of this writing. Considerable research and development effort is currently being applied in the field of semi-conductors and new knowledge will continue to accumulate. In certain instances, this may render some of the following data obsolete. Nonetheless, the general cooling techniques should remain essentially similar and future improvements are anticipated to be primarily a matter of degree and materials.

As in vacuum tubes, the largest temperature difference occurs within semi-conductor devices when they are properly cooled i.e., the external thermal resistance is low. Unfortunately, the thermal conductivity of semi-conducting materials is generally low and significant temperature gradients can be produced within them. Most of the present endeavors to produce power transistors and diodes have been directed towards the reduction of internal temperature rise. Further, the maximum temperature which semi-conductors can withstand prior to failure or loss of characteristics is rather low compared to those of other component parts.

2. Transistors

a. Effects of Temperature

In point contact type transistors both the input and output impedances decrease with increasing temperature up to 60°C., while the feedback resistance changes relatively little. The current amplification factor increases rather slowly, changing by perhaps 50% between -50 and +50°C. This usually results in a variation of power gain with temperature to the extent of a few db up to 60°C., the small change resulting from these compensating effects. Temperature cycling often leads to hysteresis effects so that circuit constants may differ by a factor of as much as two up or down following a 25°C cycle. This may be partially due to mechanical shifting of the electrode contacts from expansion due to heating and the physical nature of the assembly of the transistor. At higher temperatures, significant temperature dependent effects may be observed. Thus, prolonged storage or operation at high temperature may accelerate diffusion processes, domain growth in polycrystalline materials or outright failure. At low temperatures the number of charge carriers may be seriously decreased with resultant effects on the operating characteristics (Reference 6). In certain instances, as the temperatures increase, the equivalent circuit resistance decreases to an intolerable value.

The temperature problem in small signal circuits with point contact transistors is alleviated somewhat because the temperature dependence of point contact transistors decreases as germanium resistivity decreases. When germanium having low resistivity is used, alpha cutoff is less dependent upon temperature and the operating range is extended beyond 60°C. Consequently, the temperature characteristics of the device may also be controlled to some extent by the proper selection of germanium resistivity (Reference 7).

In junction transistors, the thermal effects are somewhat different from those exhibited by point contact transistors. Low power alloy junction germanium NPN and PNP transistors show permanent deterioration above 110°C. ambient. They will operate for short intervals at 110°C., but characteristics change (Reference 8). Increased reverse or back current attains a runaway condition unless the stability factor is low (Ref. 9). With NPN grown or fused junction transistors, the current amplification factor does not significantly change with temperature (up to 120°C.) as in PNP junction transistors.

In general, transistors appear to exhibit two types of temperature effects: namely, temperature effects due to changes in ambient temperature, and self-heating effects due primarily to power dissipation at the collector. Heating effects sometimes differ widely from transistor to transistor. Such differences between transistors are believed to be due principally to nonuniformity of germanium melts and to variations in the geometry of the transistor (Reference 10). It should also be noted that thermoelectric effects must be considered in transistor design in the same fashion as contact potential in vacuum tubes.

With point contact transistors, self heating is usually manifested by change in collector resistance. Figure 5-16 shows the self heating effects for five different transistors and Figure 5-17 presents the self heating hysteresis loop. As the collector current increases, the dissipation increases heating the transistor and decreasing the collector resistance, which again causes an increase in current, etc., leading to a runaway condition.

The effects of ambient temperature are similar as shown by Figure 5-18. The cumulative effect can be ascertained from Figure 5-19 which presents the change in output due to temperature for two uncompensated transistorized audio amplifiers. One amplifier utilized selected junction transistors with current amplification factors of .975 and the other amplifier used junction transistors with current amplification factors of .9 to .95. The basic circuits consisted of two

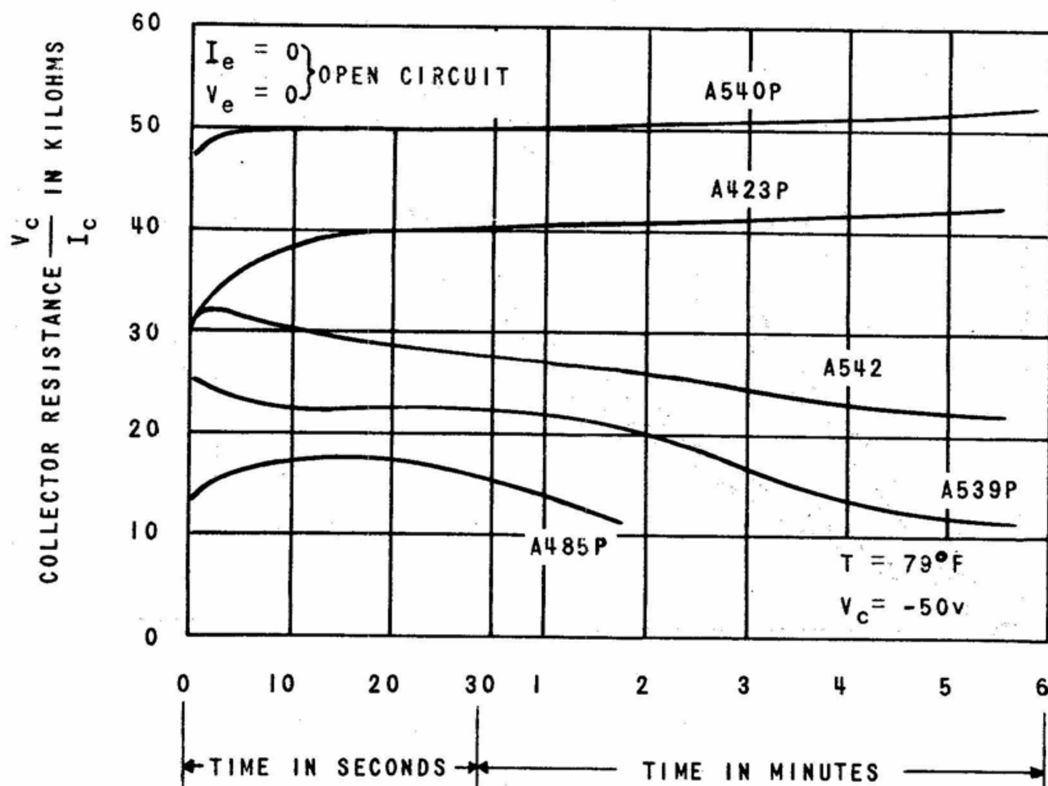


FIG. 5-16 SELF-HEATING EFFECTS OF FIVE DIFFERENT TRANSISTORS
(from Ref. 10)

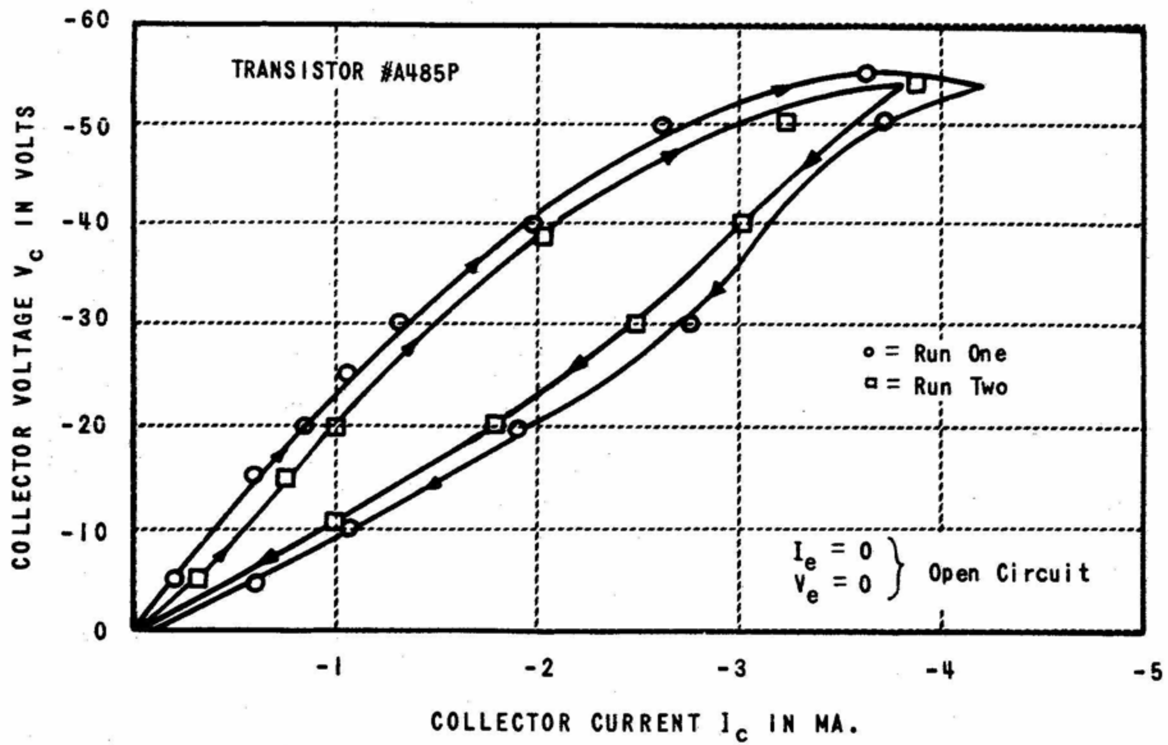


FIG. 5-17 SELF-HEATING "HYSTERESIS" LOOP
(from Ref. 10)

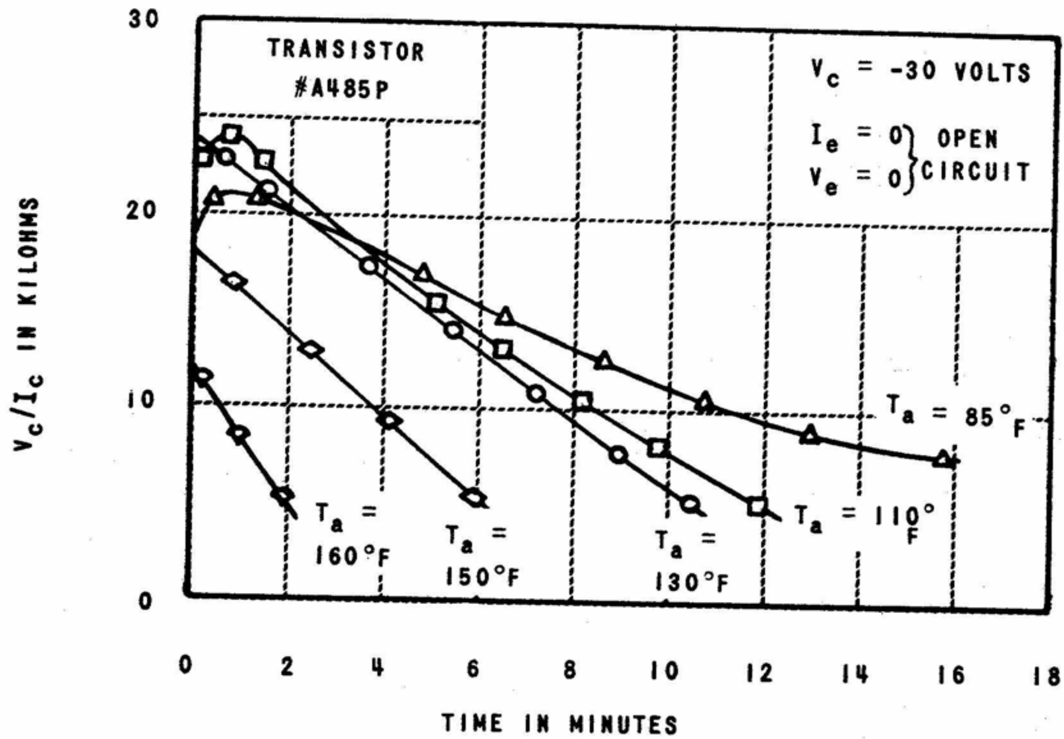


FIG. 5-18 EFFECTS OF AMBIENT TEMPERATURE ON COLLECTOR RESISTANCE
(from Ref. 10)

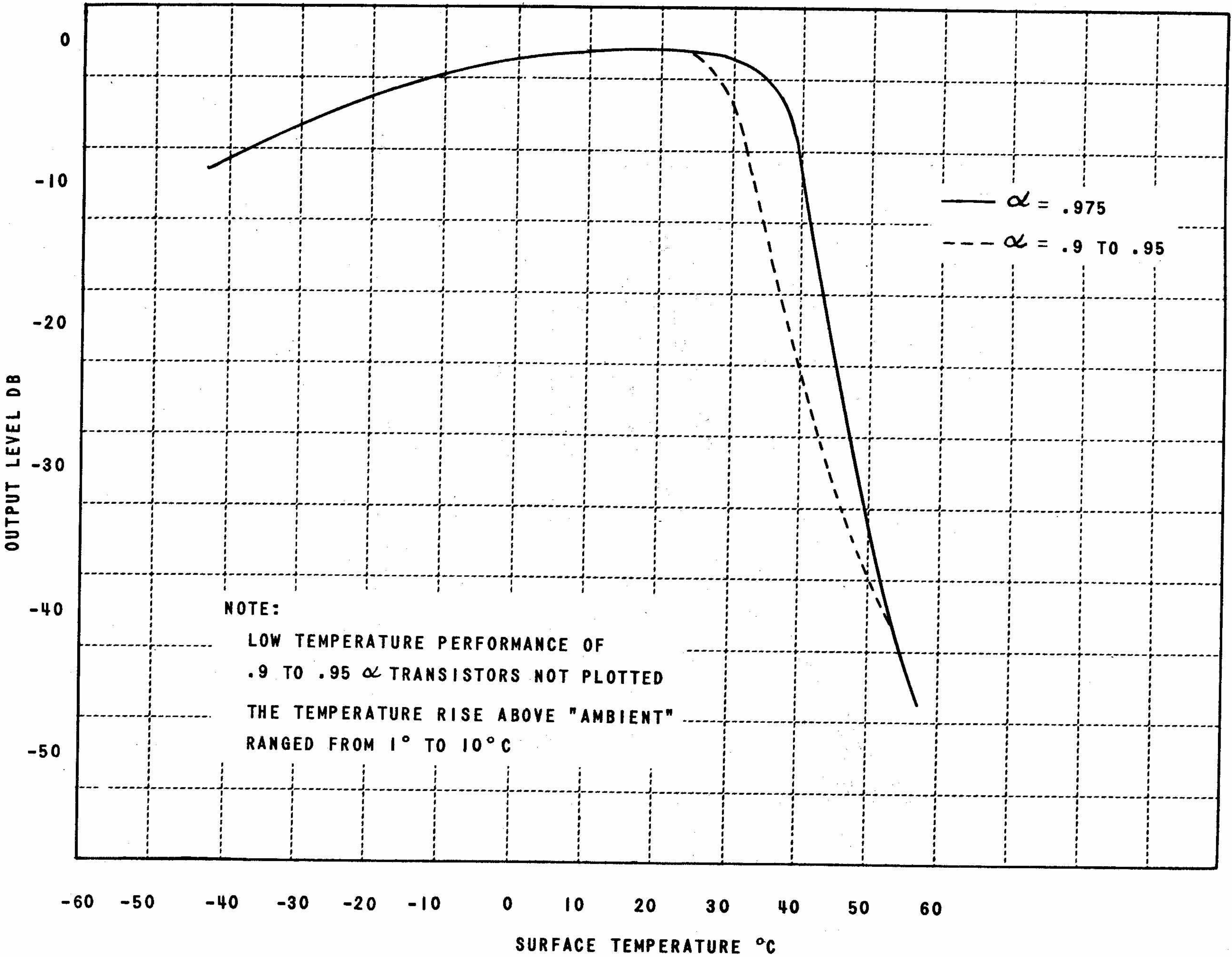


FIG. 5-19 TEMPERATURE SENSITIVITY OF UNCOMPENSATED TRANSISTORIZED AUDIO AMPLIFIER

grounded emitter stages, a grounded collector stage and a push pull class B output stage. In most instances, the transistors were operated at approximately 80% of their maximum level. These examples were deliberately constructed to demonstrate the extremes in performance which can occur due to temperature. Note that the temperature is in terms of surface temperature, not ambient temperature.

b. The Temperature Limitations of Transistors

The practical limit of usefulness of germanium is in the vicinity of 80°C. Certain special processes are now in use which extend this limit to approximately 100°C. At this temperature, intrinsic conduction electrons start to become present in sufficient numbers to seriously modify the rectifying and transistor properties. Indications are that transistor life is significantly shortened when the germanium is operated above 70°C.

Another temperature limiting factor in junction transistors is the low melting point of indium (154°C.) which is currently used at the junctions in germanium transistors. Silicon transistors can be operated to temperatures of the order of 150°C. without malfunctioning. However, the thermal effects on performance are similar but to a lesser degree than in germanium transistors.

c. Modes of Heat Transfer

In transistors the power is usually dissipated in the germanium or the semi-conducting collector and base material and to some extent in the indium, as in the case of junction transistors. The heating occurs locally inside the device and must be removed through conduction in the germanium, indium, the case and the leads. The thermal conductivity of these materials is rather low, being .094 watts/sq.in.-°C/in. at 25°C. for indium and .232 watts/sq.in.°C/in. for germanium at 25°C. decreasing to 20% less at 100°C. This results in a thermal resistance of the order of 20°C/watt within the average junction transistor. With power junction transistors, the internal temperature may be as great as 100°C. at dissipations of the order of 5 watts. Thus, thermal resistances of the order of 15°C/watt are produced. One group of power transistors now commercially available exhibit internal thermal resistances as low as 2°C/watt.

The primary problem in the design of power transistors is that of lowering the internal thermal resistance. Significantly greater powers could be handled if the heat could be more readily removed. Unfortunately, the size of the functioning elements must be minimized in order to permit operation even at relatively low frequencies, and the thermal conductivities

of the semi-conducting materials are such that relatively large temperature differences can be produced. The potentialities of germanium are great. The emission current density in transistors is of the order of 10^5 greater than that of barium oxide cathodes. Theoretically, transistors should be operable to extremely large currents.

d. Methods of Thermally Rating Transistors

The temperature sensitivity of transistors makes accurate thermal rating imperative. In general, there are three methods of rating transistors.

(1.) Surface temperature ratings are commonly used for low power transistors dissipating 100 milliwatts or less. This method is satisfactory only at low power levels wherein the temperature difference between the surface and the internal elements is small.

(2.) Measurement of the collector cutoff current I_{co} of junction transistors under several operating conditions will provide a measure of the internal element temperature (Ref. 9). Thermal instability leads to electrical instability and consequently I_{co} varies with junction temperature. In general, I_{co} doubles approximately every 11°C (Ref. 19). Curves of collector current vs. junction temperature are available from certain manufacturers for specific transistors. From these curves the true junction temperature can be estimated. This method of thermally rating transistors is excellent and is recommended as an alternate to method (3).

(3.) Internal thermal resistance ratings in °C/watt are recommended for transistors.

Such ratings provide an index of the internal junction temperature if a single measurement of the transistor surface temperature can be obtained. Several manufacturers now rate transistors in this fashion. For example, types 2N43, 2N44, and 2N45 are rated at 200°C/watt; types H-3 and H-4 are rated at 3.5°C/watt; and type P-1 is rated at 2°C/watt.

e. Methods of Cooling Transistors

There are two basic cooling philosophies applicable to transistors:

(1.) Provide essentially a constant temperature environment and remove the heat from each transistor so that electrical instability can not occur. Transistors appear to have an optimum internal operating temperature which is

in the neighborhood of 25°C. If maintained at this temperature, stabilization circuitry will not be required and increased performance per transistor can be achieved. Unfortunately, a controlled constant temperature environment is usually only achieved with special refrigeration and heating equipment. Nonetheless, in certain instances, for example, computers utilizing large quantities of transistors, it is desirable in the interests of reliability to provide transistors with an ideal environment.

(2.) Cool the transistors so that their peak internal temperatures do not exceed approximately 80 to 100°C. and provide circuit stabilization to avoid "runaway". At elevated environmental temperatures it will be necessary to keep the collector dissipation low by whatever derating is required. This will reduce the internal temperature rise.

Since the internal thermal resistance of most transistors is relatively high and since the maximum temperature of the elements is relatively low, the external thermal resistance to the sink must be small, especially if the environmental temperature is high or if the transistor is dissipating appreciable power.

It is sometimes difficult to achieve the required low thermal resistances even by conduction. Certain power transistors are rated for their maximum dissipation at 25°C. case temperature. However, with appreciable power dissipation, 25°C. case temperatures are almost impossible to obtain by natural methods of heat removal. Further, the temperature of the ultimate sink in military electronic equipment seldom is as low as 25°C. Even if it were this low or slightly lower, the permissible temperature rise between the sink and the case of a power transistor would be extremely small. Such low thermal resistances could be obtained only at the expense of relatively large and heavy metal conductors.

Unfortunately, most power transistors must be electrically insulated from the cooling conductor which is usually grounded electrically. This insulation can represent a significant thermal resistance external to the transistor. Mica insulators .002 in. thick and 1/2 in. diameter have been used for this purpose. Even so, the thermal resistance of the mica is of the order of 1°C/watt. Consequently, it appears that rating of power transistors for operation at 25°C. case temperature in military electronic equipment is somewhat impractical.

Typical thermal conditions achieved in convection cooling a power transistor were determined. With a 3 in. x 5 in. x 1/4 in. thick dull copper plate as a cooling fin in free air, 14°C. rise was achieved at 2.9 watts dissipation. With a blackened

3/4 in. x 2-1/2 in. x 1/16 in. thick copper plate in free air (see Fig. 5-20), 42°C rise was exhibited at the same dissipation. Conversely, the transistor alone in 26°C. free air had a 43°C rise at only 1.5 watts dissipation.

Conduction cooling was also evaluated. When insulated by a .002 in. thick mica washer and a 1/32 in. thick phenolic insert and attached to a sink at 26°C., 14°C rise was obtained at 2.9 watts dissipation. This is similar to the free air condition above with the 3 in. x 5 in. x 1/4 in. cooling fin. The insulators were removed and the transistor was firmly attached to the 26°C. sink (see Fig. 5-21). At 7.5 watts dissipation, 12°C rise was exhibited and at 8.8 watts the same 14°C rise was achieved. The joints were in the "as finished" condition. The effect of joint discontinuities was displayed when the joints were thoroughly cleaned. The 14°C rise occurred at 12 watts dissipation. Temperature rises of 11 and 13°C were obtained at 8.9 and 10 watts dissipation, respectively. This configuration gave a thermal resistance of approximately 1°C/watt to the hottest spot on the transistor case. It is obvious that metallic conduction cooling is the only natural cooling method which should be used with power transistors.

f. Thermal Stability - (from Ref. 58)

Thermal "runaway" occurs when the rate of heat generation caused by the temperature-dependent component of leakage current and the collector voltage exceeds the capacity of the cooling facility. The criterion of stability is

$$\frac{dP_T}{dT_j} < \frac{1}{R} \qquad (52)$$

where:

P_T = Total transistor power dissipation in watts
T_j = Temperature of the collector junction °C.
R = Thermal resistance for the entire thermal path in °C/w

If it is assumed that gain is independent of temperature, the junction saturation current, which is temperature-dependent, doubles with every nine Centigrade degrees rise in temperature. Stability is maintained if

$$SI_S < \frac{13}{V_C R} \text{ amperes,} \qquad (53)\ (D.E.)$$

where:

I_S = Junction saturation current.
S = Circuit stability factor, as defined in Ref. 59.

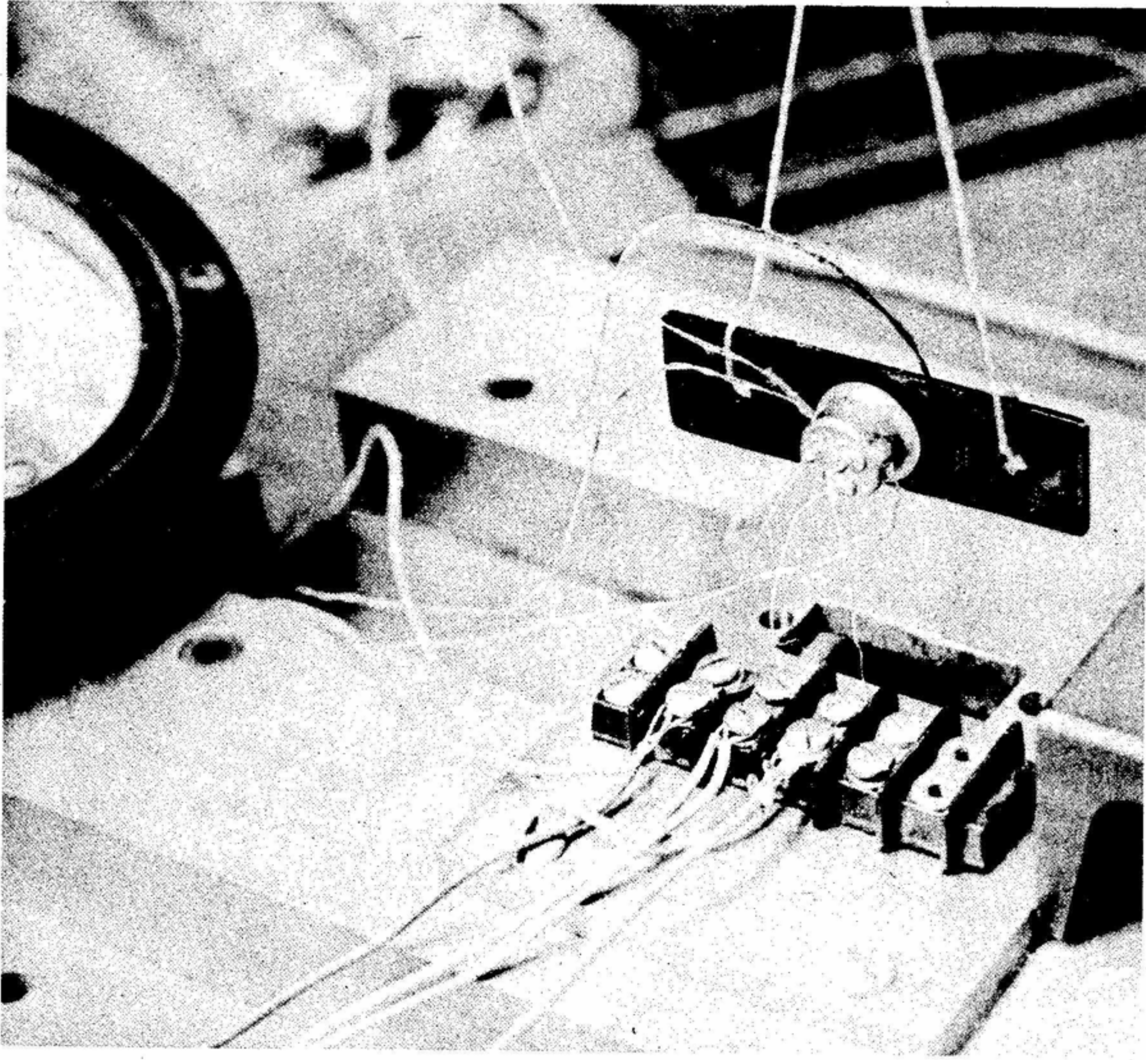


FIG. 5-20 CONVECTION AND RADIATION COOLED POWER TRANSISTOR

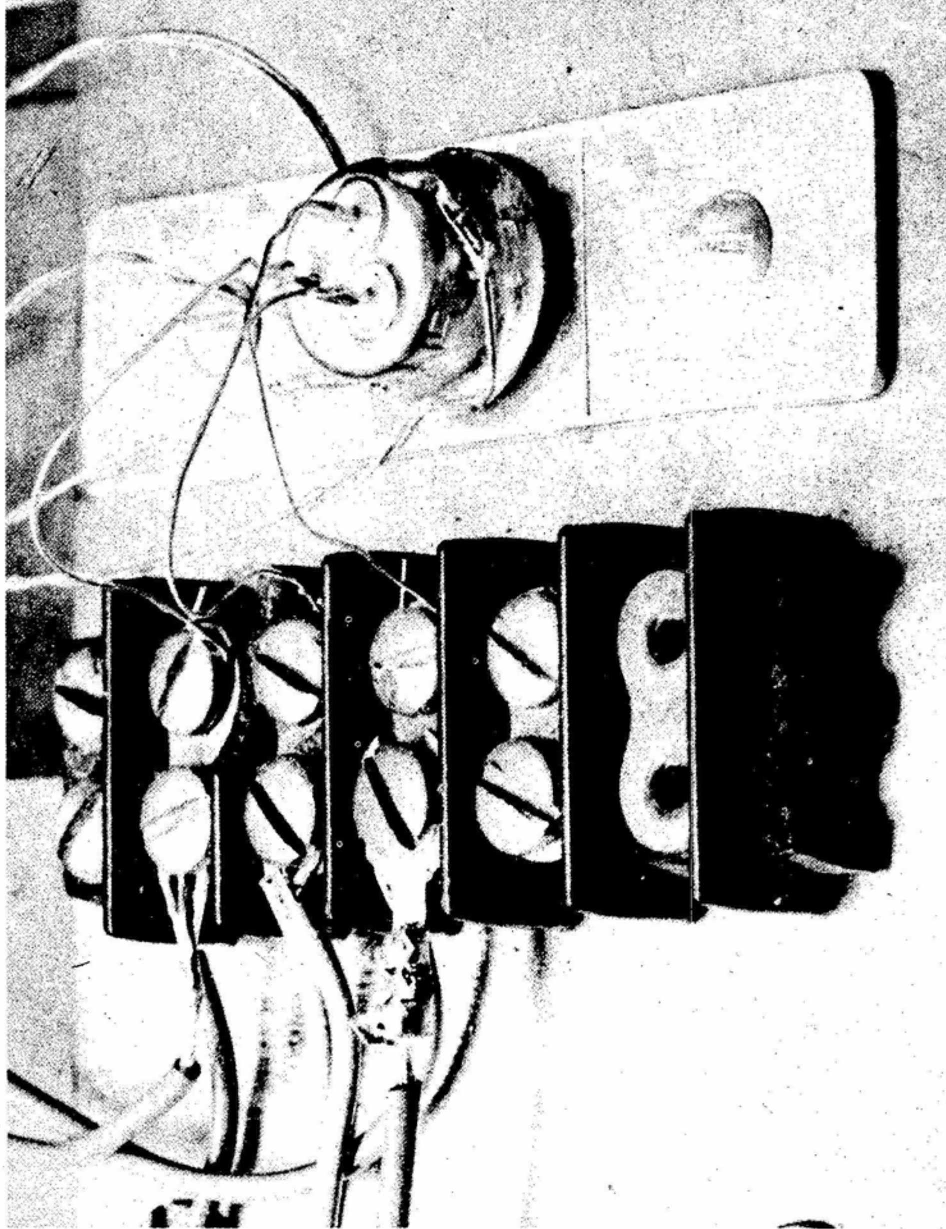


FIG. 5-21 CONDUCTION COOLED POWER TRANSISTOR

Example 8 The Stability Criterion.

The following example illustrates the application of this stability criterion.

Assume:

V_C = 60 volts
R = 5°C/watt
S = 2
T_j = 80°C. At this junction temperature the minimum I_{CBO} (collector junction leakage) current at 2 volts is predicted to be 14 milliamperes.

Under these conditions the circuit would not exhibit thermal runaway, since,

$$2 \times .014 < \frac{13}{60 \times 5} \text{ or } .043.$$

3. Semi-Conductor Diodes

In general, semi-conductor diodes are temperature sensitive similar to transistors. However, since diodes do not exhibit gain, these effects are not amplified and the net electrical effect is not as significant as in transistors.

a. Low Power Diodes

This category includes point contact diodes and small junction diodes which are operated with heat dissipations of the order of milliwatts. Low power diodes seldom exhibit significant self heating and they operate at or near the temperature of their thermal environment. Consequently, the thermal resistance between the surfaces of low power diodes and their immediate environment need not be extremely low.

Point contact diodes are temperature sensitive similar to point contact transistors. Figure 5-22 presents the variation of static characteristics of germanium diodes with temperature. In general, such diodes should not be operated at temperatures exceeding 80°C. dependent upon the permissible deterioration of characteristics. Silicon point contact diodes may be used up to 125°C.

b. Power Rectifiers

Power diodes dissipate appreciable heat and care must be exercised to make sure that the external thermal resistance is low. Finned units for convection cooling should always be mounted in free air and isolated from other heat producing parts.

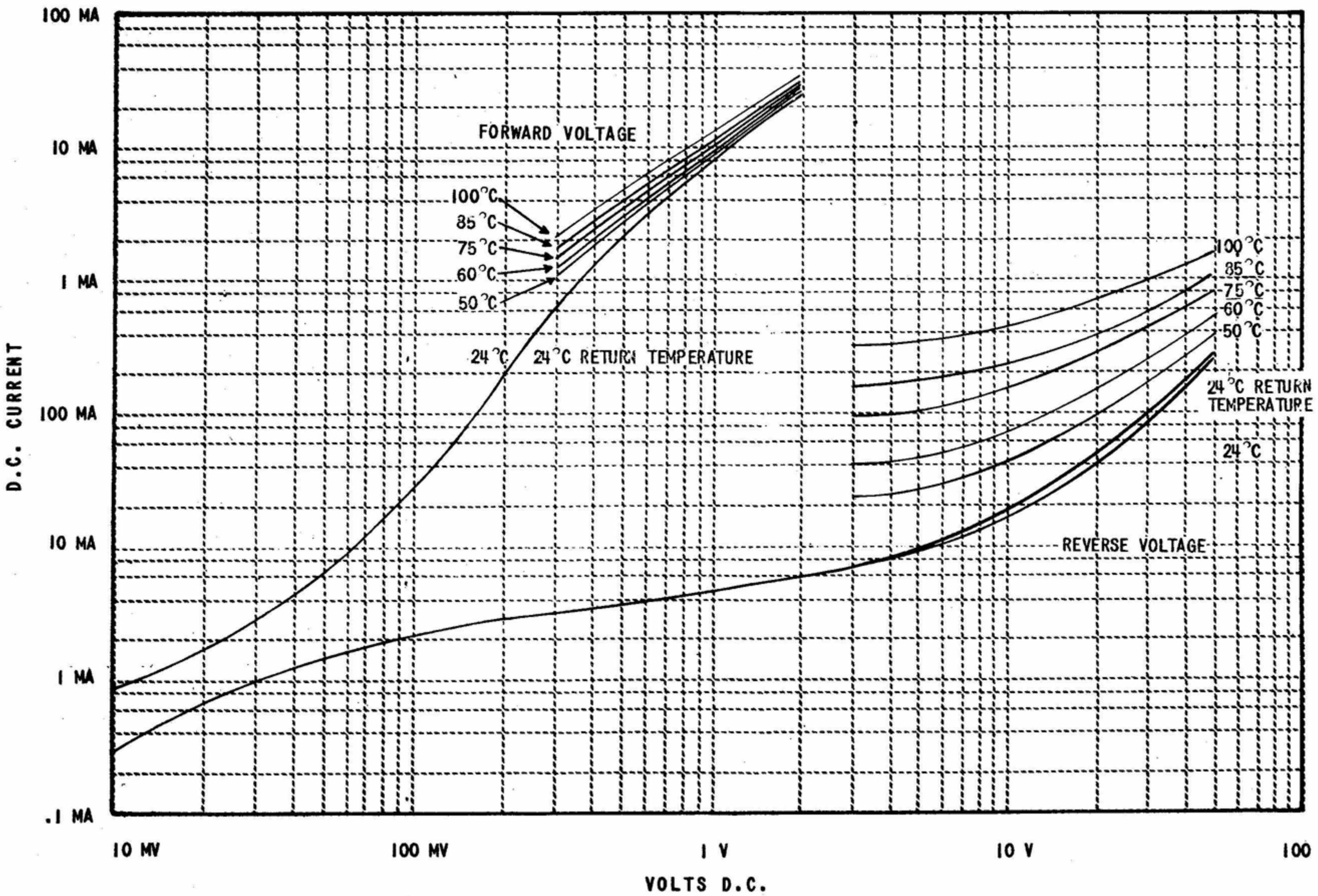


FIG. 5-22 VARIATION OF STATIC CHARACTERISTICS OF POINT CONTACT GERMANIUM DIODES WITH TEMPERATURE.

Courtesy - Sylvania Electric Products. Inc.

(1.) Germanium Diodes

Germanium junction diodes are probably the most temperature sensitive of this class. Both the forward and reverse dissipations should be determined. The forward current power losses are dissipated through the germanium and the reverse current losses are dissipated at the junction. These devices are furnished with or without cooling fins and may be formed into stacks if desired. In general, finned germanium junction diodes should be derated 300 percent at 55°C. environmental temperature. High temperature units can be operated to 85°C. maximum environmental temperature. Diodes without fins should be conduction cooled in the same fashion as power transistors. Silicon junction diodes differ thermally from germanium diodes only in that the permissible operating surface temperatures may be as great as 150°C.

(2.) Selenium Rectifiers

Most selenium rectifiers are designed for convection cooling. In critical applications the reverse and forward power dissipation should be considered. Allow ± 3% variation from stack to stack in determining the forward voltage drop. Further, these units age with time and the forward resistance increases gradually. If the design of equipment having a shelf or operating life of over three years is contemplated, provide cooling for a 100% increase in forward dissipation. Special units have been recently developed which are derated for service at environmental temperatures up to 125°C. maximum. Most selenium rectifiers are rated for 100 percent duty at only 35°C. ambient temperature. They should be derated and cooled so that the maximum cell surface temperature does not exceed 75°C. (See Table XX.). It should be noted that the normal protective coating on selenium rectifiers is not intended for military application. Special coatings are needed (Ref. 19).

TABLE XX.

MAXIMUM FIN OPERATING TEMPERATURES OF SELENIUM RECTIFIERS

Type	Maximum Fin Temperature - °C.			
	Absolute Maximum	700 hr. life	3000 hr. life	10,000 hr. life
Standard	115	100	85	75
Hi. Temp.	135	120	115	105

D. RESISTORS

Most resistors for electronic circuitry have been designed for natural cooling in free air. Resistors differ from vacuum tubes in that almost any power can be dissipated in a given resistor provided adequate cooling is present. Thus, with increased cooling, resistors can be operated successfully at increased ratings. The dissipation rating of a given resistor will therefore vary, dependent upon its thermal environment. Resistor deratings are based on the maximum operating temperature and are published by resistor manufacturers. Section X of C.A.L. Report HF-710-D-16 (NAVSHIPS 900,190) presents the limiting temperatures of resistors.

The average 1/2 watt composition carbon resistor in conventional equipment will reject approximately 40 per cent of its heat by free convection, 10 per cent by radiation, and 50 per cent by conduction through the leads. It will exhibit up to 40°C rise at the surface of the center of the body with respect to the ends and have a net thermal resistance of 125°C/watt with respect to its environment. Similarly, a 2 watt resistor will reject approximately 36 per cent of its heat by free convection, 19 per cent by radiation and 45 per cent by conduction through the leads. The thermal resistance with respect to the environment will be approximately 25°C/watt.

There is not much an equipment designer can do to increase the cooling by radiation and natural convection other than to reduce the environmental temperature. 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°C rise above ambient temperature was obtained with zero length leads connected to a sink (Ref. 31). 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 thermally grounded to the chassis.

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.

Typical conduction-cooled resistors were studied. Figs. 5-23 and 5-24 show the measured thermal performance. Note that the slope of the curves approaches straight-line linearity, indicating that conduction predominates over the other non-linear modes of cooling. The resistor in Fig. 5-23, which is rated at 3 watts dissipation with conduction cooling, is a conventional 1 watt resistor. Similarly, the resistor in Fig. 5-24, rated at 4 watts dissipation, is normally rated at 2 watts.

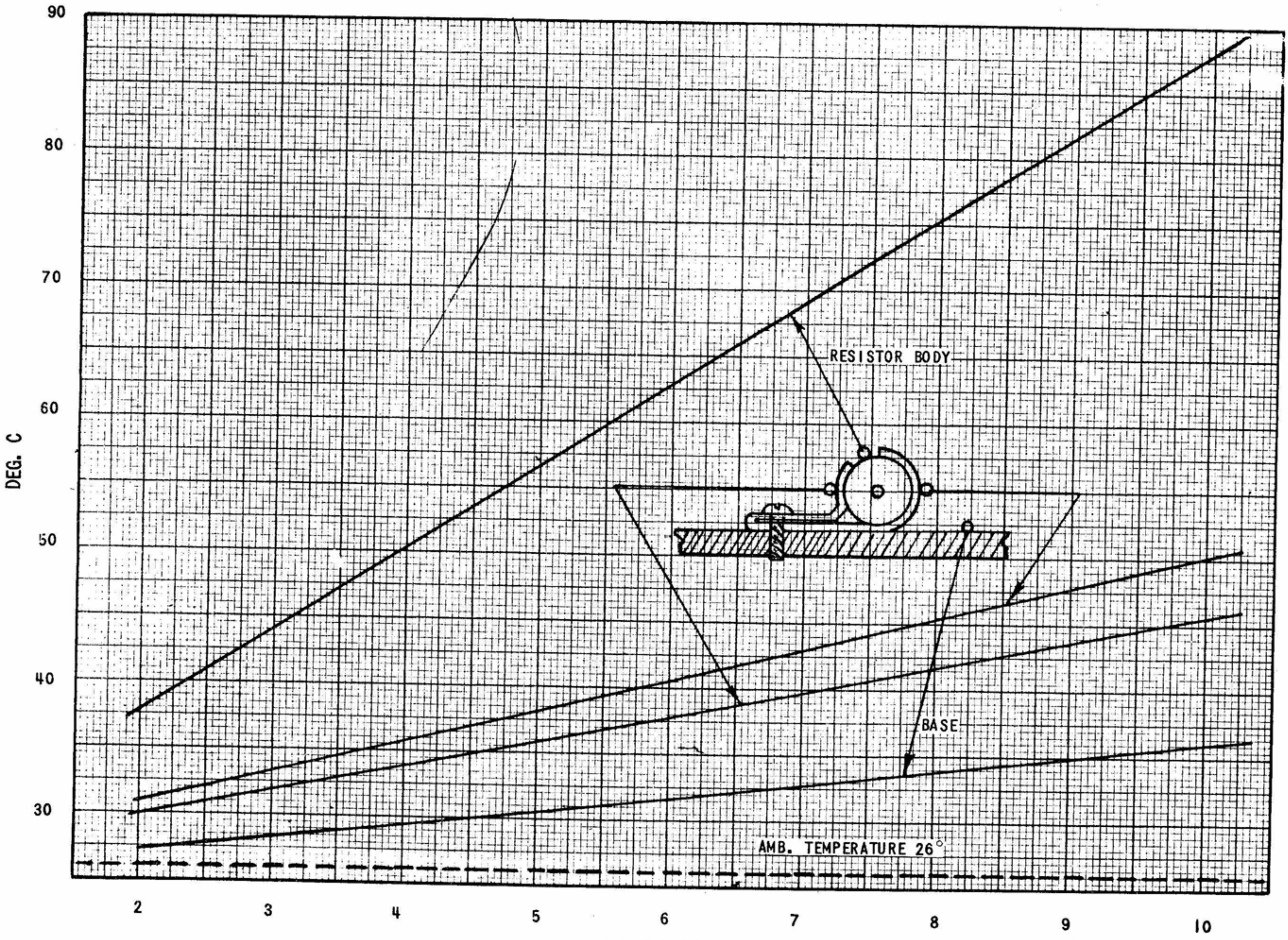


FIG. 5-23 THERMAL PERFORMANCE OF CONDUCTION COOLED A.B. RESISTOR.
TYPE GM - RATED 3 WATTS

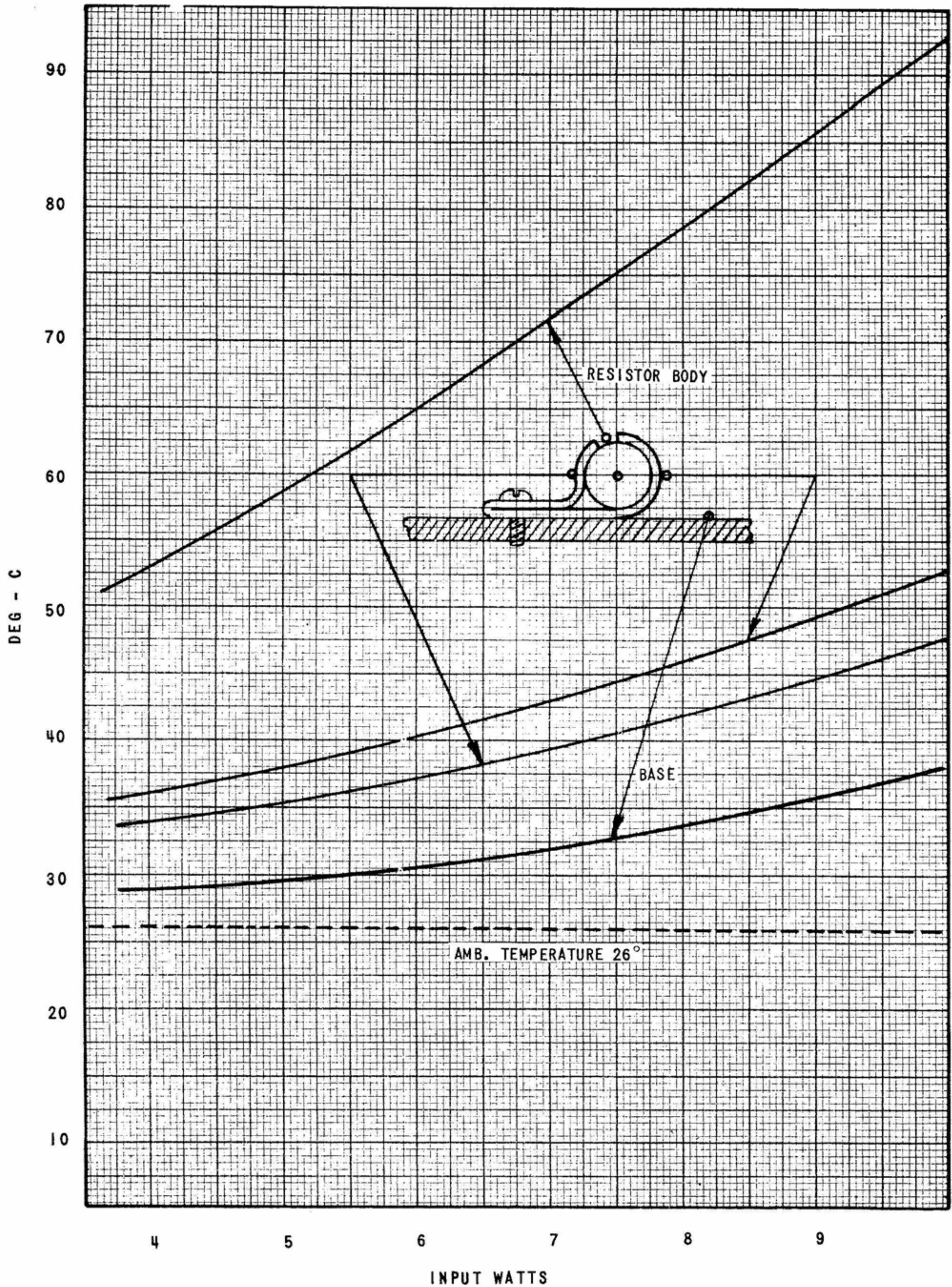


FIG. 5-24 THERMAL PERFORMANCE OF CONDUCTION COOLED A.B. RESISTOR
TYPE HM - RATED 4 WATTS

Large wire-wound resistors dissipate considerable heat and must be mounted not only for adequate cooling, but also so as to minimize heating of other parts by radiation. Power resistors usually operate at high temperatures and much of their heat is rejected by this mode. Chapter VI describes techniques of mounting resistors in groups. 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 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.

E. IRON CORE INDUCTORS

1. General

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 usually due to core losses and I^2R' losses in the windings. Chokes and inductors generally have only I^2R' losses.

Well designed and operated inductors are normally very reliable. Most failures are due to the breakdown of the insulation. The thermal ratings of the various insulations are outlined in C.A.L. Report No. HF-710-D-16. Fig. 5-25 shows the thermal performance of transformers larger than 100 VA (Ref. 39).

2. Heat Flow in Transformers

The thermal analogy of a transformer is displayed on Fig. 5-26. The heat flows out of a transformer structure to the environment as follows: In the usual power transformer, the coil and core are close fitting, and together present a more or less unified exterior surface which is composed of the exposed surface of the coil plus the exposed surface of the core. Heat flow to these surfaces from the interior of the transformer is primarily by conduction. The losses in the core and coil are not generated at a point but are distributed uniformly. This causes the interior flow paths to be quite complicated. However, with regard to the circuit external to the core and coil, the paths may be represented as single resistances connecting the source of heat with the transformer surface. Included in the circuit is provision for conduction between the core and coil. If an

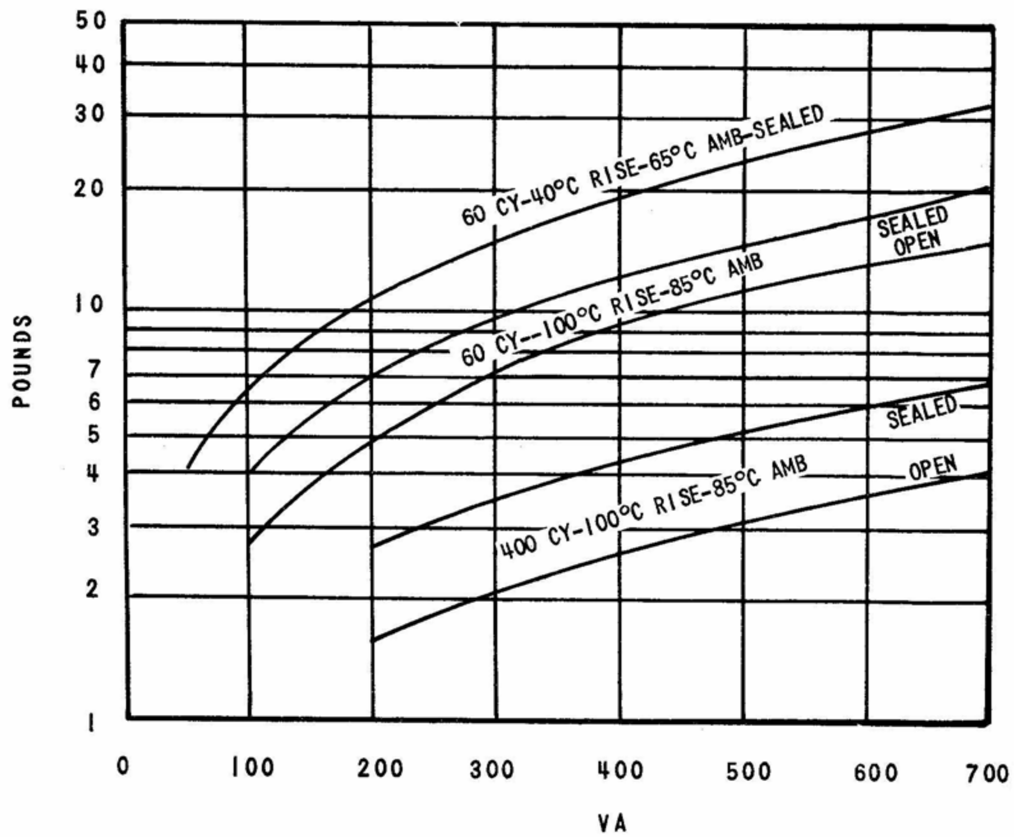


FIG. 5-25 TEMPERATURE RISE OF TRANSFORMERS

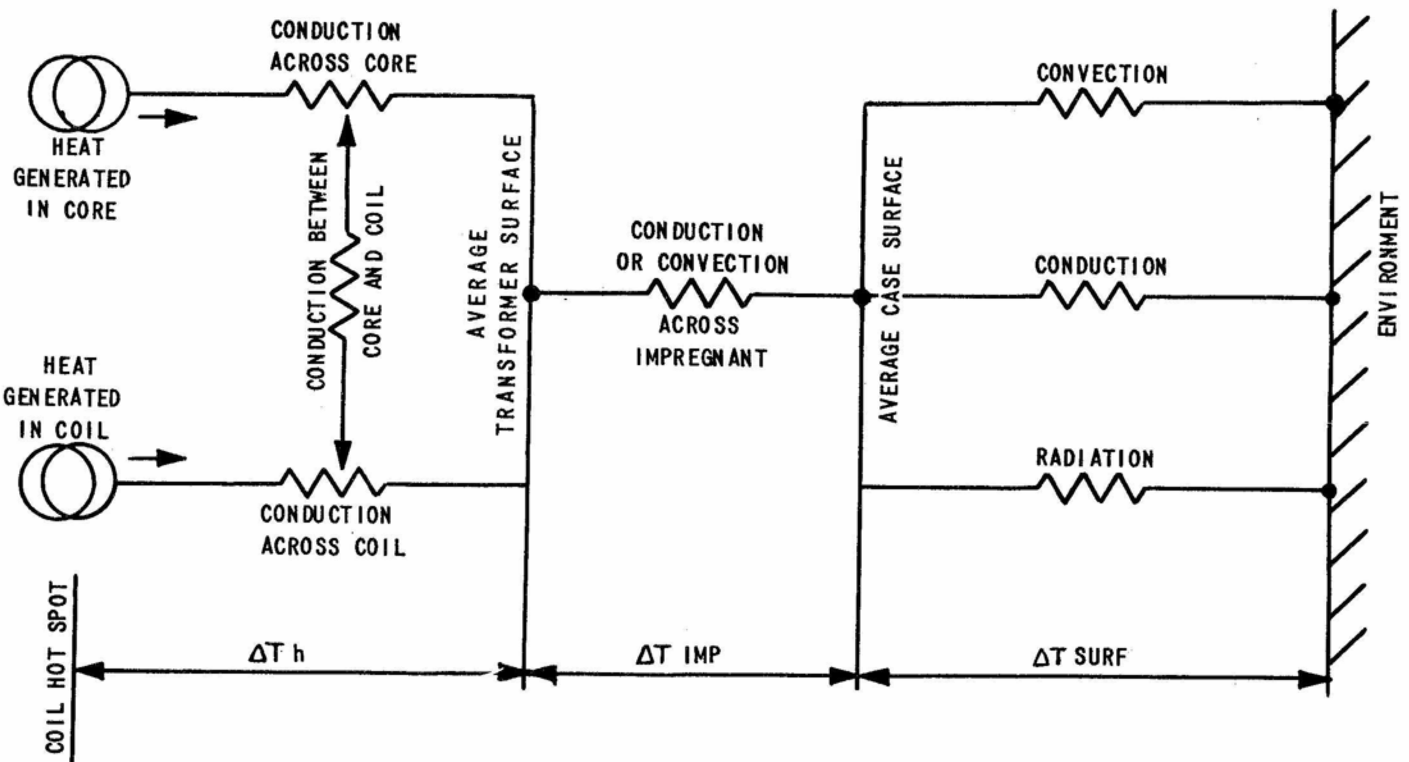


FIG. 5-26 HEAT FLOW ANALOGUE OF A TRANSFORMER

impregnant is used, another thermal resistance is added to the curcuit. If the impregnant is a solid such as compound, wax or plastic, the heat transfer takes place only by conduction. If the impregnant is a liquid such as oil, the heat transfer takes place by conduction and convection. However, only one resistance can be used for this path, in either instance. If the transformer is an open type with no case or encapsulation, the resistance representing impregnant drop can be replaced by a direct connection (Ref. 40).

From the outside surface, heat transfer to the environment can take place by all three modes, radiation, convection, and conduction through the mountings. Heat transfer by conduction is sometimes small. Conduction is recommended as the primary mode of heat transfer from a transformer. It can be greatly increased by securely bonding the core surfaces to a chassis designed to remove the rejected heat. If the transformer is uncased, the mounting feet and core surfaces must be smoothly machined and fitted to the chassis to form a good thermal bond. With enclosed transformers, the case may be thermally bonded to the chassis. At least the case will be cooled by conduction. Should the transformer be oil filled, considerable heat may be transferred from the core into the case for conduction to the chassis. In the event that the core cannot be thermally bonded, then, at least, the mounting lugs should be bonded to the chassis. In those instances wherein significant conduction cooling cannot be obtained, then the total heat flow will divide between conduction and radiation with resultant increased operating temperatures.

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.

3. Calculation of Temperature Rise.

Conduction from the core to the chassis should be utilized wherever possible. Chapter III of this Manual presents pertinent design data. It appears that the conductive thermal resistance between the core, the mounting structure and the chassis can be predicted by measurement of the electrical resistance as mentioned in item B7 of Chapter III. Item B3 of Chapter VI presents typical temperature gradients which can be obtained by good thermal bonding to the chassis. In general, it is recommended that the thickness of the impregnant between core laminations be minimized and that all mating surfaces in the heat flow path, such as those between the laminations and frame and the mounting feet and the chassis, be smooth and clean. All impregnant should be removed from these surfaces, since it constitutes a high resistance to heat transfer.

The heat removed by convection and radiation from iron core inductors can be predicted by utilizing the nomographs presented in Chapter III. 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. Radiationwise, the emissivities of the surfaces of insulating materials range from .8 to .9. In general, most paints, enamels and varnishes range from .88 to .93 surface emissivity between 50 and 200°C.

The surface temperature rise may be calculated based upon the above and the recommendations of Chapter III. The internal thermal resistance within the windings of transformers and inductors is, of course, within the control of the transformer designer and cannot be significantly modified by the user. Ref. 40 presents information in this matter.

VI THE THERMAL DESIGN OF EQUIPMENT FOR NATURAL COOLING

A. GENERAL

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 means are recommended for use within most miniaturized electronic assemblies. Natural cooling is frequently the only practical means of heat transfer within equipment. Hermetic sealing and the dense packaging of parts can prevent the application of other techniques. Further, forced cooling requires the addition of energy to the system to move coolants and this increases the total power consumption and the total heat to be rejected.

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. Control of the heat flow paths may be difficult. Convection currents frequently transfer the heat into other areas which in turn will require increased cooling. Modern equipment is usually so densely packaged that much of the air cooling is by gaseous conduction rather than convection. Note that, even though heat transfer by conduction through metals inherently permits better control of the heat flow paths than the other modes, it is not always easy to direct and control heat flow by conduction. Consequently, the thermal design must be such that conductors having predetermined cross-sectional areas and short heat paths are provided at specified locations in order to prevent the flow of heat into unwanted paths.

The designer should make sure that the heat dissipated by his equipment is minimized. Considerable thermal difficulties can be avoided initially if the electrical efficiency of the equipment is increased. Further, the size, weight, and cost of the equipment will be decreased, together with a probable increase in reliability.

B. THE PLACEMENT AND MOUNTING OF PARTS

1. General

Major thermal benefits can be achieved through the judicious placement of electronic parts within assemblies. In general, it is desirable to locate the heat sources as near as possible to the coolest surface. This will provide the shortest thermal

path from the source to the sink, together with the minimum thermal gradient. In order to obtain maximum heat transfer, heat-producing parts cooled by radiation and conduction should always be mounted with their major axes parallel to cooled surfaces. When convection cooling is mandated for electrical reasons, heat-producing parts should be mounted with their longer dimensions vertical. For example, resistors should be vertical because, in general, they will operate cooler than when mounted horizontally. However, resistors exceeding 4 to 6 inches in length should be mounted horizontally (See Chapter V).

2. Thermal Interaction

When heat producing electronic parts, not cooled by metallic conduction, are mounted in closely spaced groups, convection will be minimized and much mutual heat transfer can occur by radiation and gaseous conduction. Under these conditions, excessive operating temperatures can easily be obtained. Consequently, the parts in such arrangements must be carefully oriented. Heat transfer by radiation can be minimized by providing "shiny" surface radiation shields. In general, it is advisable to "ground" thermally the shields to reduce radiant heat transfer into the parts to be protected.

If it is necessary to group heat-producing parts together, metallic conduction cooling is recommended. Conduction cooling will tend to reduce the thermal interaction and will usually permit any practical spacing desired. If metallic conduction cannot be provided, then the parts should be arranged horizontally to form a "bank" of minimum height. If vertical stacking is necessary, then the parts must be staggered. Avoid mounting convection cooled heat sources directly above each other.

a. Resistors

Resistors must be derated when mounted in groups. This is especially applicable to resistors dissipating appreciable power. Fig. 6-1 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 now available and are generally recommended. The thermal characteristics of resistors with dissipation ratings ranging up to two watts are discussed in detail in Chapter V. Only the thermal interactions are mentioned in this chapter.

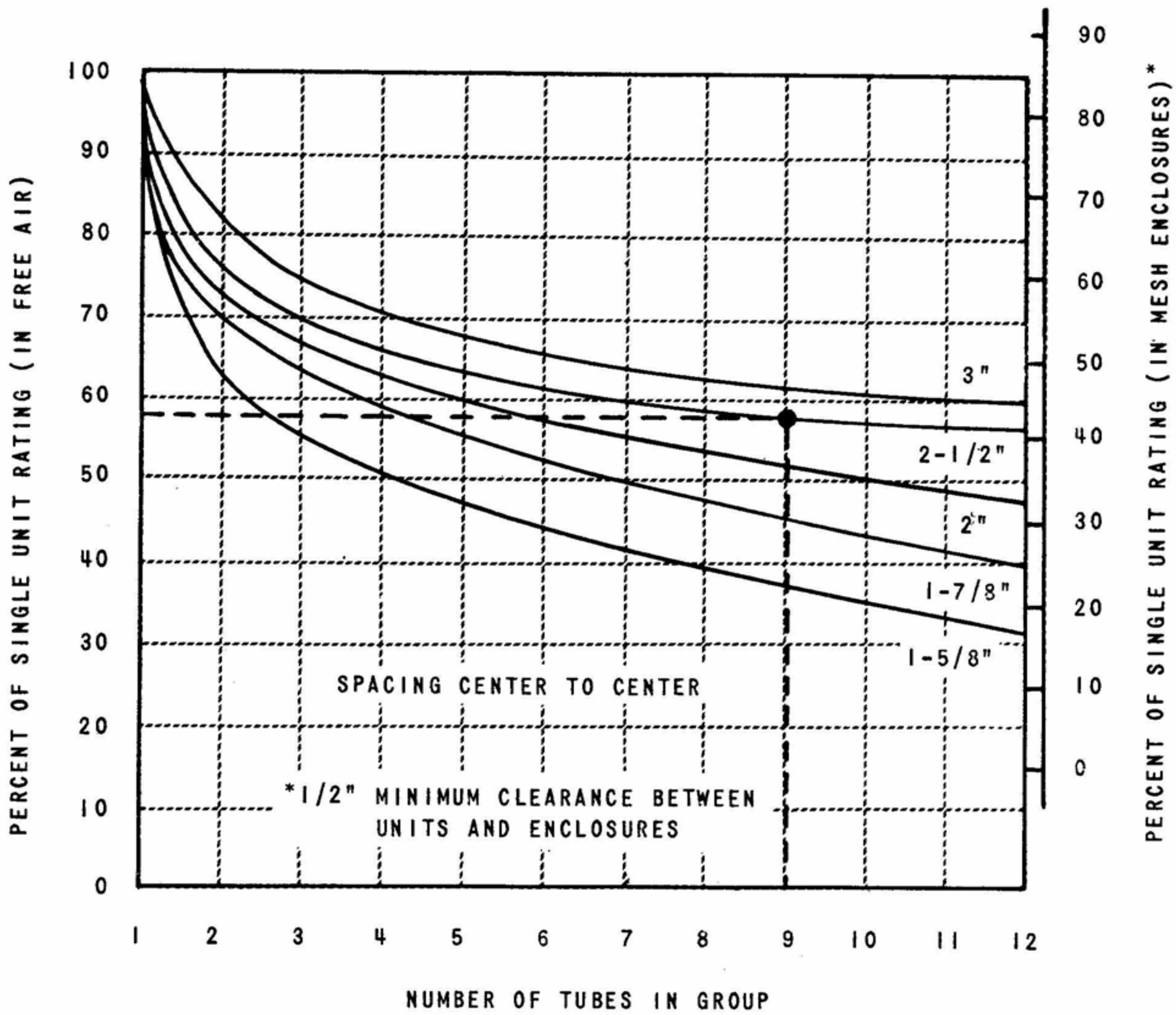


FIG. 6-1 GROUP MOUNTING OF POWER RESISTORS

When composition carbon and other similarly shaped resistors are mounted within 1/8 in. 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 in. 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 in. 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 in., each resistor will increase from 10 to 15°C in surface temperature (See Chapter V). Fuse clip-type mountings thermally bonded to cooled metal surfaces are 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.

b. Vacuum Tubes

Mutual heat transfer between vacuum 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 in. and preferably $1\frac{1}{2}$ bulb diameters, when the bulb diameter exceeds 1/2 in. The greater the separation the better, especially if the tubes are dissipating appreciable power (See Chapter V).

c. Other Heat Sources

In general, because of their lower operating temperatures, iron core reactors and other heat sources will not exhibit as much mutual heating as resistors and vacuum tubes. However, the part-to-part spacing should exceed at least 1/2 in. to minimize mutual heating. In those instances wherein the temperatures of, for example, transformers are relatively high, then the same thermal interaction will occur as with the higher temperature heat sources.

d. Design Considerations

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. Alternately, the parts can be mounted on separate heat conductors.

The main point is that if 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.

Non-heat-producing parts may be temperature sensitive and it is generally recommended that such parts 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 Fig. 6-2). This arrangement is recommended. Fig. 6-3 presents an alternate configuration which, under equivalent conditions, will result in somewhat higher temperatures than obtained with the arrangement of Fig. 6-2.

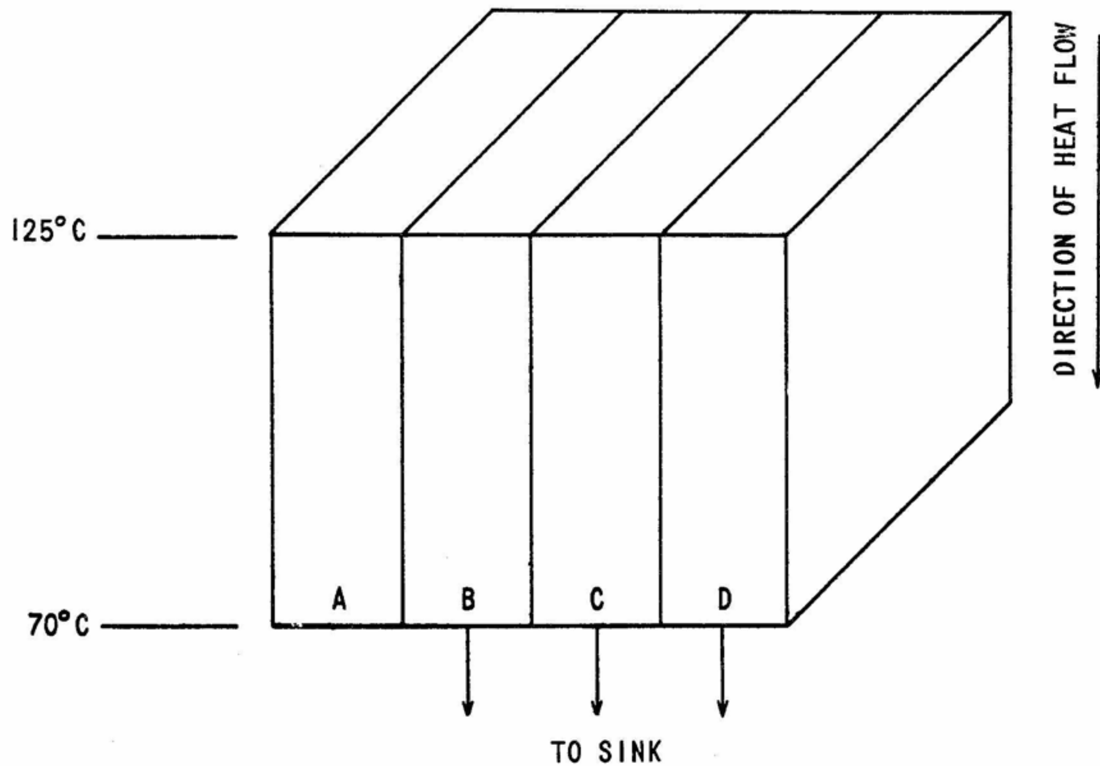
3. The Thermal Effect of Joints

The contact resistance of joints between metal conductors is discussed in Chapter III. Some practical effects of such discontinuities are presented herein.

a. Tube Shields

A conduction-type miniature tube shield (Shield "B" - see Chapter V) was mounted on a constant-temperature (water-cooled) metal chassis to demonstrate the effects of variation in contact resistance. Initially, the tube shield was firmly attached to a brass chassis with both mating surfaces in the "as furnished" condition. The resulting temperatures are shown by Fig. 6-4. Note that the temperature rise at 12 watts between the chassis and the shield mounting base is 27°C and that between the bottom of the shield and the shield mounting base is 23°C. The thermal resistance between the shield and the mounting base is 1.92°C/watt.

The surface of the chassis and the surface of the bottom of the shield base were carefully machined and lapped to provide intimate contact between the surfaces and a minimum thermal resistance. The resulting thermal performance is presented on Fig. 6-5. Note that the temperature rise between the shield base and the chassis is reduced to 12°C at 12 watts. This represents a thermal resistance of 1°C/watt. The temperature rise between the bottom of the shield and the shield mounting base is unchanged, as it should be, and the thermal resistance of this portion of the heat flow path is still 1.92°C/watt.



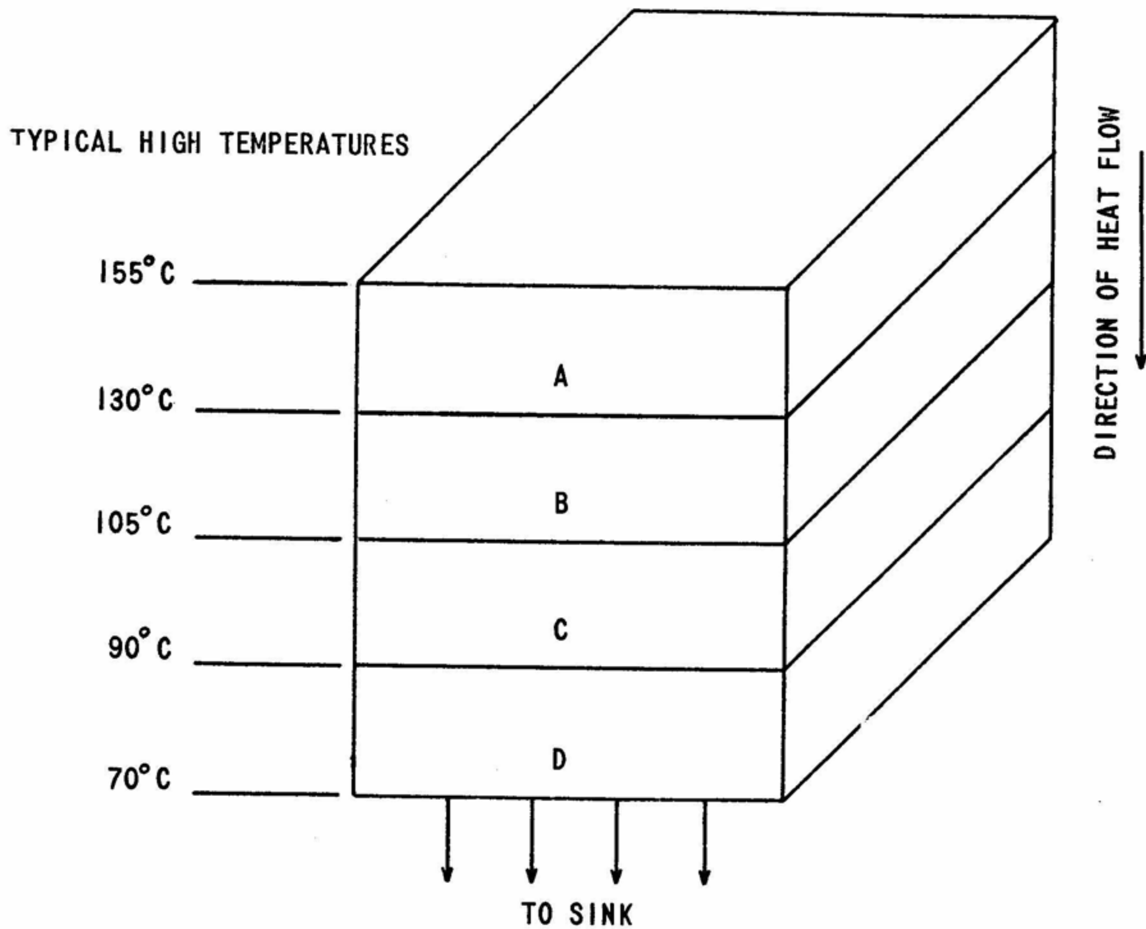
Region A - High temperature region with parts having large heat concentration

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

FIG. 6-2 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

FIG. 6-3 ALTERNATE METHOD OF LOCATING PARTS

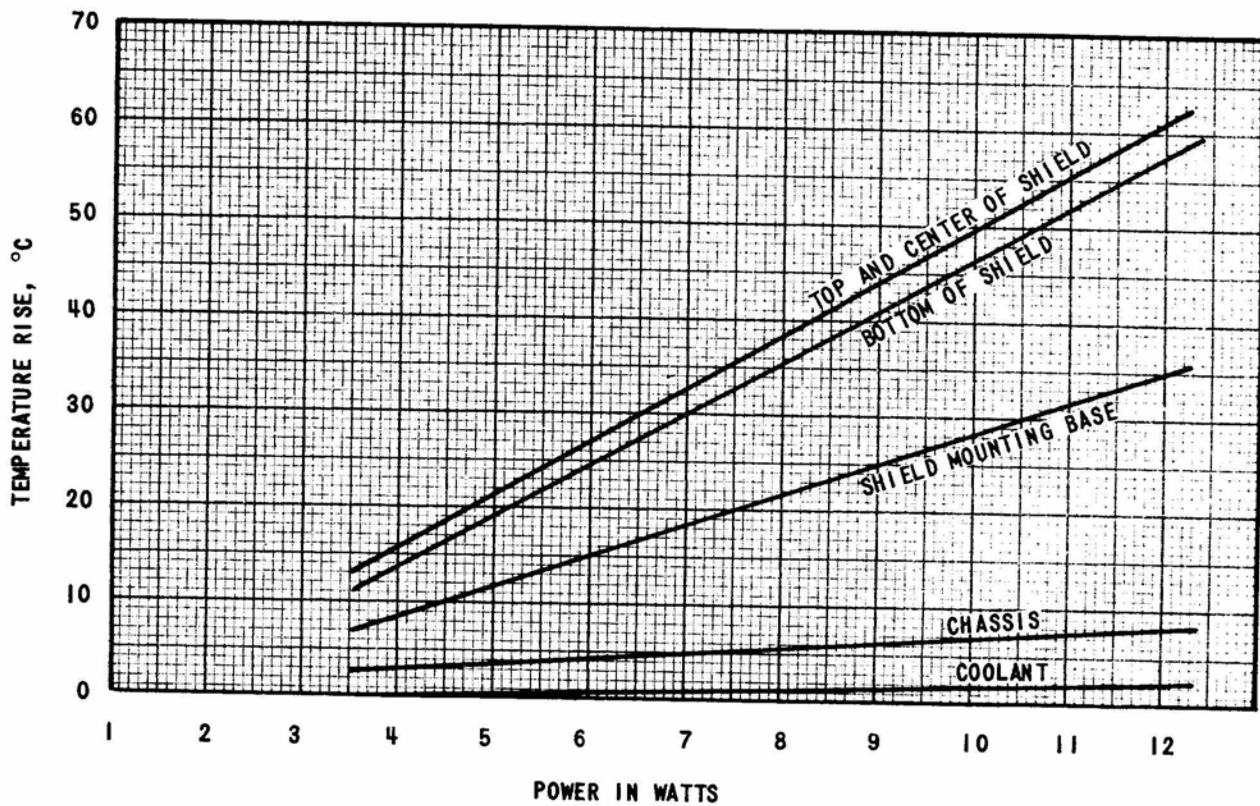


FIG. 6-4 TEMPERATURE GRADIENTS AT BASE OF TUBE SHIELD WITH SURFACES IN AS FURNISHED CONDITION

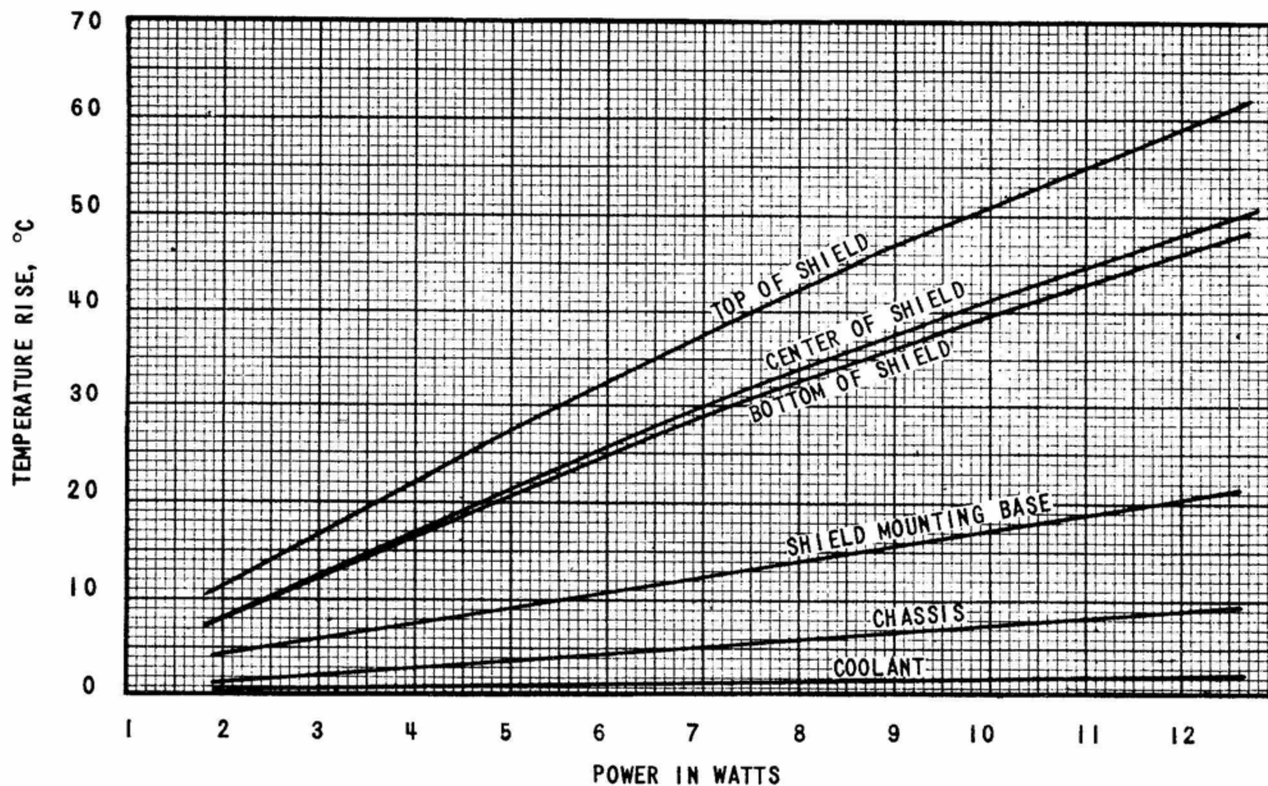


FIG. 6-5 TEMPERATURE GRADIENTS AT BASE OF MINIATURE TUBE SHIELD IN GOOD THERMAL CONTACT WITH CHASSIS

Reduction of the contact resistance between the chassis and the shield from 2.25 to 1.0°C/watt represents an improvement of 1.25°C/watt. This can be a significant saving, dependent upon the energy to be transferred across the joint and the relationship of the temperatures involved to the maximum operating temperature of the parts. The largest resistance to heat transfer beyond the inner surface of the shield is now that between the shield and its mounting base. If possible, shield manufacturers should reduce this resistance, especially since other shield types than that used in this instance have even higher contact resistances.

The above thermal resistances were also determined by measurement of the electrical resistance (see Chapter III) and acceptable correlation was obtained. It is to be noted that the plate temperature of the tube used in this demonstration was reduced only 1.5°C and the bulb temperature decreased 10°C when the contact resistance was minimized. It appears that the plate temperature was leveling off, i.e., "plateau condition" (See Chapter V).

b. Iron Core Reactors

The effects of contact resistance in the mounting of a conventional iron core reactor were demonstrated with a 60-cycle filament transformer. This transformer had an efficiency of 66 per cent, i.e., at 61 watts output, the primary input was 92 watts and the transformer losses were 31 watts. In free air, at 26°C., the iron core temperature was 72°C. and the internal winding temperature (at a random location) was 91°C. When this transformer was mounted on a chassis at 34°C. and the mounting surfaces were in the as furnished condition, the temperature at the top of the core was reduced to 59°C., the middle and lower portions of the core were at 53°C., and the internal winding temperature was 69°C. The thermal resistance from the iron core to the air in the free air cooled instance was 1.5°C/watt and in the conduction-cooled condition (using average lamination temperature) it was lowered to .68°C/watt. This represents an additional (shunt) thermal resistance of 1.25°C/watt by conduction.

When the mounting flanges of the transformer were cleaned and flattened to provide a lower contact resistance to the chassis, the temperature at the top of the core was reduced to 51°C., the middle was at 42°C., the lower section was at 40°C., and the internal winding temperature was 61°C. The thermal resistance from the core to the chassis was .33°C/watt. This is equivalent to adding another parallel heat flow path of .65°C/watt and, since it is approximately one-half of the 1.25°C/watt thermal resistance previously obtained, it can be concluded that the thermal resistance by conduction to the chassis was reduced to less than half of its former value by providing an intimate contact at the joint.

4. Embedment

In general, due to the relatively high thermal resistance of plastics, parts dissipating appreciable power should not be potted. Non-heat-producing parts and certain heat-producing parts having small heat concentrations can be successfully encapsulated. This is exemplified by Figs. 6-6, 6-7 and 6-8. Ref. 55 contains similar information. Note that the thermal gain, even at the lower power levels, is not too great. 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 Fig. 6-8 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 wave-length radiation, but the highly reflective interface between the plastic and the resistor also tends to minimize heat transfer by radiation. Section C, which follows, discusses plastic embedment in further detail.

C. THE THERMAL DESIGN OF ASSEMBLIES

1. General

Each assembly should be designed to reject its heat into 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 dissipations. Consequently, assemblies are severely penalized if they are required to act as their own intermediate thermal sinks to a specified "ambient" temperature and, consequently, connection 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 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.

It is recommended that each assembly be provided with a thermal connection (sink connector) which matches the thermal resistance of the cooling system. Each assembly could then be rated for thermal performance in terms of "sink connector" temperature. This would provide an accurate method of 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.

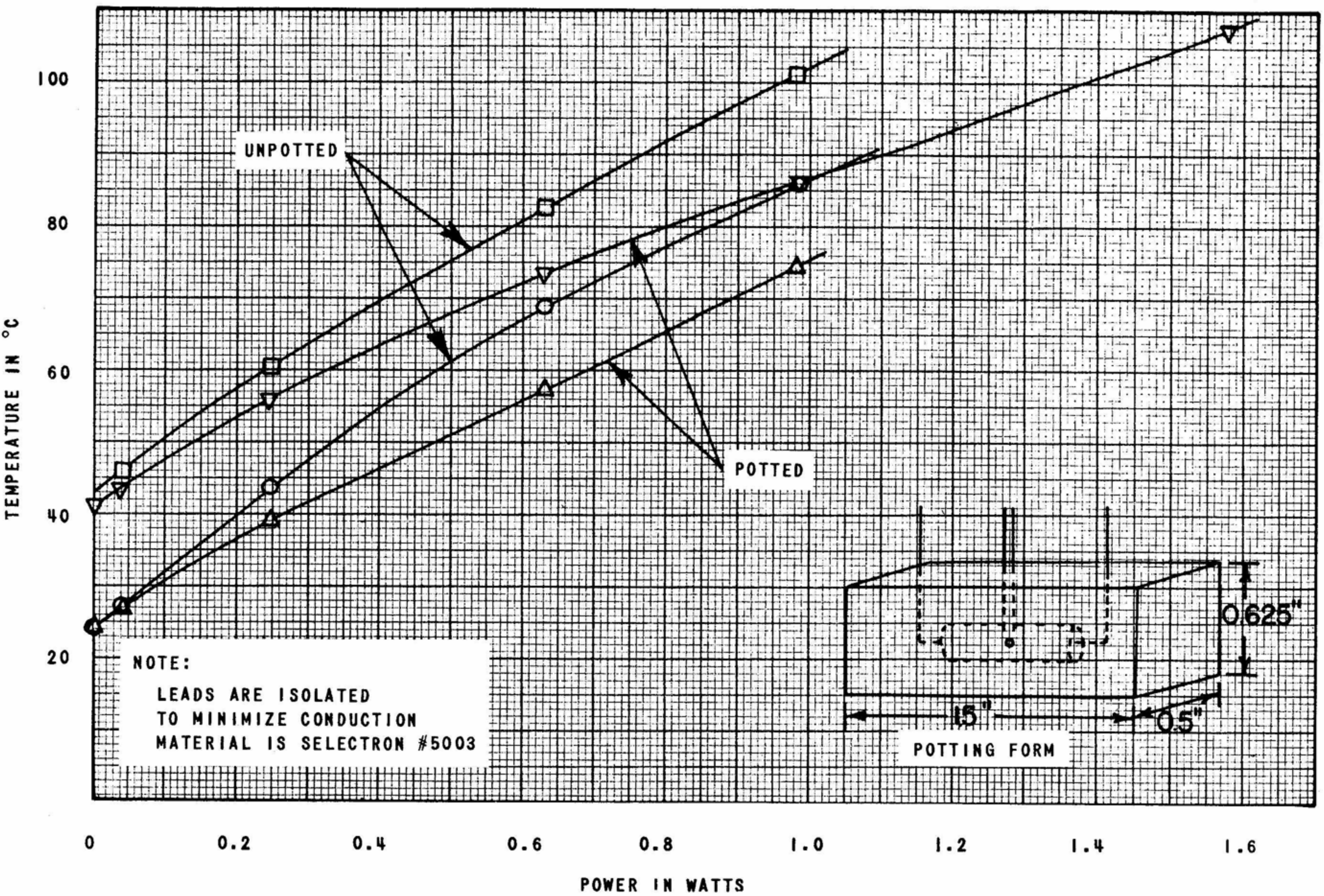


FIG. 6-6 BODY SURFACE TEMPERATURE OF AN ALLEN-BRADLEY 5600-OHM, 1WATT COMPOSITION RESISTOR AS A FUNCTION OF THE POWER DISSIPATED

(Courtesy Bell Aircraft)

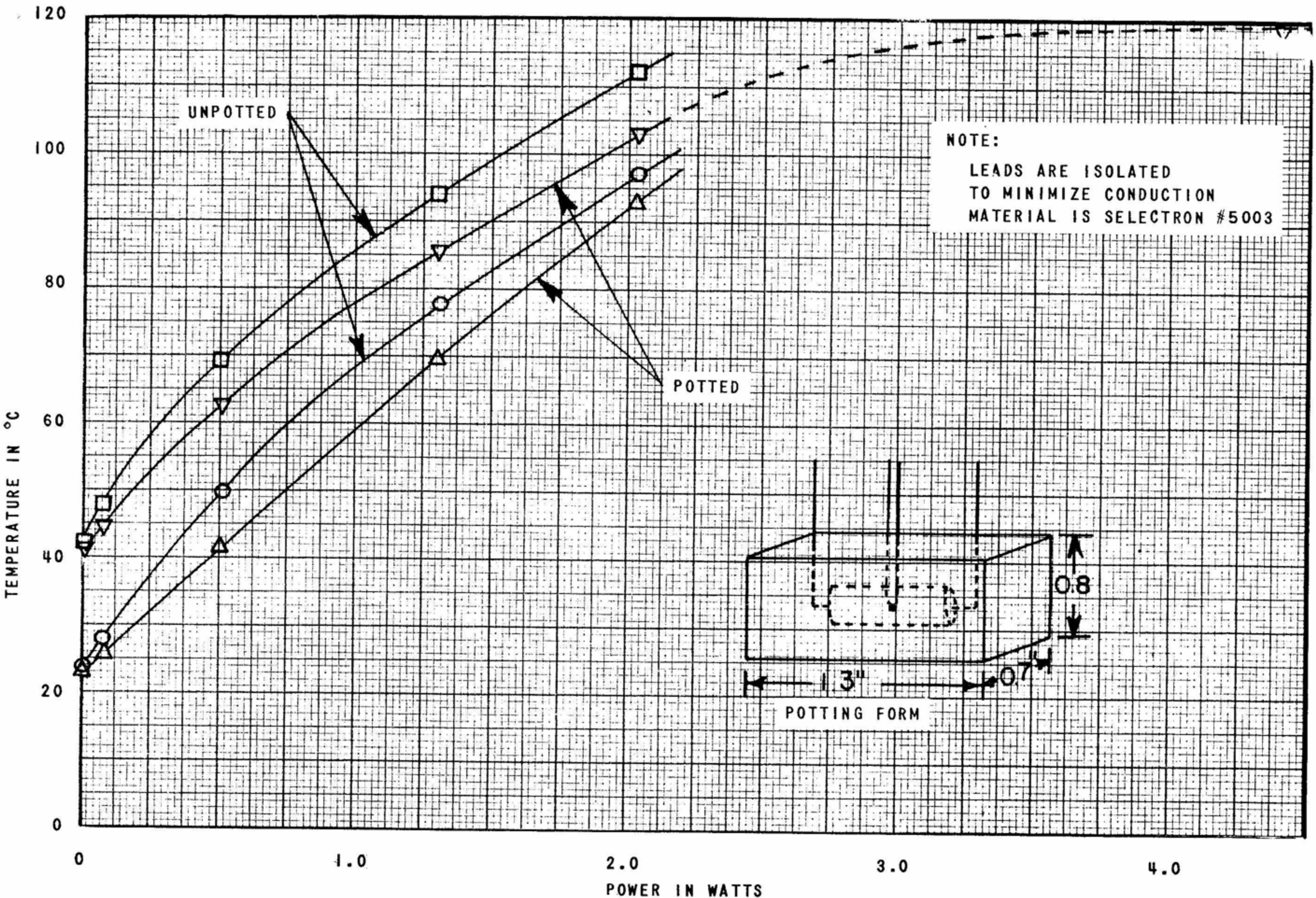


FIG. 6-7 BODY SURFACE TEMPERATURE OF AN ALLEN-BRADLEY 5600 Ω, 2 WATT, COMPOSITION RESISTOR AS A FUNCTION OF THE POWER DISSIPATED (Courtesy Bell Aircraft)

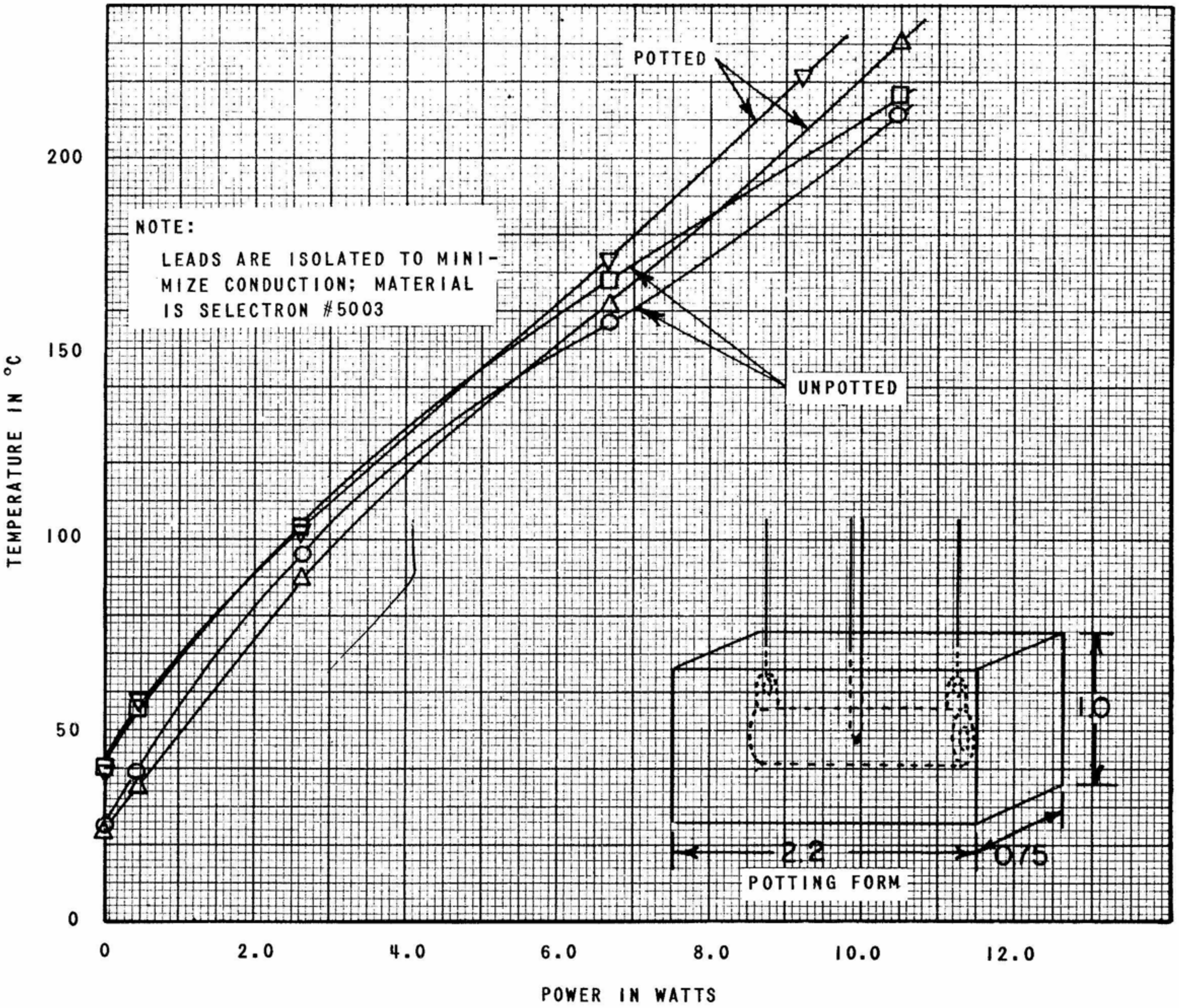


FIG. 6-8 BODY SURFACE TEMPERATURE OF AN I.T.E. 5000-OHM, 10 WATT, WIREWOUND RESISTOR AS A FUNCTION OF THE POWER DISSIPATED
(Courtesy Bell Aircraft)

2. Metallic Conduction Cooling

a. Determination of the Thermal Resistance of an Assembly

The thermal resistances between the heat sources and the sink connectors or heat transfer surfaces of a typical conduction-cooled assembly are outlined in Fig. 6-9. 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 within the tubes, between the tubes and the shields, and between the shields and the subchassis can be determined from the data presented in Chapter V and from the earlier discussion. Also, if the shield and its mount have not been analyzed, the thermal resistance in the metal can be predicted by measuring the electrical resistance (See Chapter III). The degree of cooling of the tube and the tube to tube shield thermal resistances can be determined, if necessary, by utilizing a "Thermatron" (See C.A.L. Report #HF-1053-D-2). The thermal resistances of most resistors are given earlier in this chapter and in Chapter V.

The point of the above is that when the dissipated power, the maximum operating temperatures of the heat sources and the sink connector temperatures are known, the intermediate thermal resistances can be determined by a simple "Ohm's Law" analogy. The physical size of the metallic heat conductors necessary to obtain the desired thermal resistance in $^{\circ}C$/watt can be computed, based upon the information in Chapter III.

<u>Example 9</u>: Design Example with a High Heat Concentration

<u>Given:</u>

An assembly wherein convection and radiation are to be minimized is to be designed to operate reliably at 55°C. sink connector temperature with the following heat sources:

2 - T-3 subminiature tubes dissipating 4 watts each

1 - T-5½ miniature tube with a 2" bulb dissipating 10 watts

12 - ½ watt resistors mounted in the conventional manner and dissipating two watts total. The maximum surface temperature of the resistors is 125°C.

<u>Solution:</u>

The total power to be dissipated is

$2 \times 4 + 10 + 2 = 20$ watts

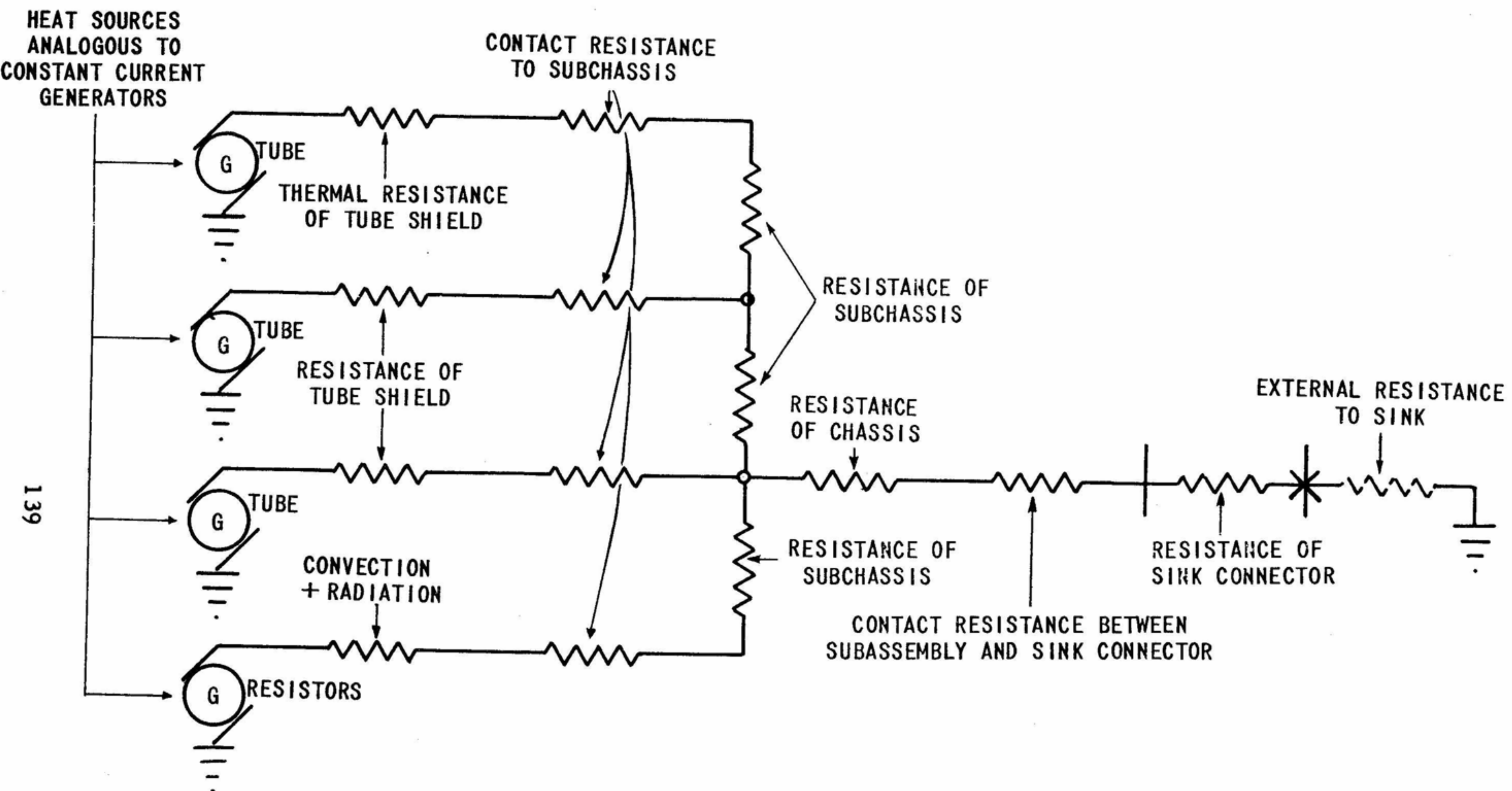


FIG. 6-9 MAJOR THERMAL RESISTANCES IN A CONDUCTION COOLED ASSEMBLY

Fig. 6-10 presents a schematic diagram of the heat flow in the assembly.

Where:

R_z = the total thermal resistance external to the tubes

R_r = the internal thermal resistance of the tubes

R_1 = the thermal resistance of the tube shield

R_2 = the contact resistance to the subchassis

R_3 = the thermal resistance of the subchassis

R_4 = the thermal resistance of the chassis

R_5 = the thermal resistance of the sink connector

R_x = the thermal resistances associated with the resistors

(1) Resistors

Since the $\frac{1}{2}$ watt resistors are mounted in the conventional fashion and none are dissipating significant power, it appears that adequate cooling can be provided by normal radiation, conduction, and convection within the assembly. The net thermal resistance of such resistors is approximately 125°C/watt (from Chapter V) and if each is dissipating 2/12 = .17 watts and has a maximum temperature rating of 125°C., the maximum thermal resistance permissible for each, if conventionally mounted in an environment near the sink connector temperature, is:

125 - 55 = 70°C rise

The maximum thermal resistance $R_x = \frac{70}{2} =$ 35°C/watt for all 12 resistors or

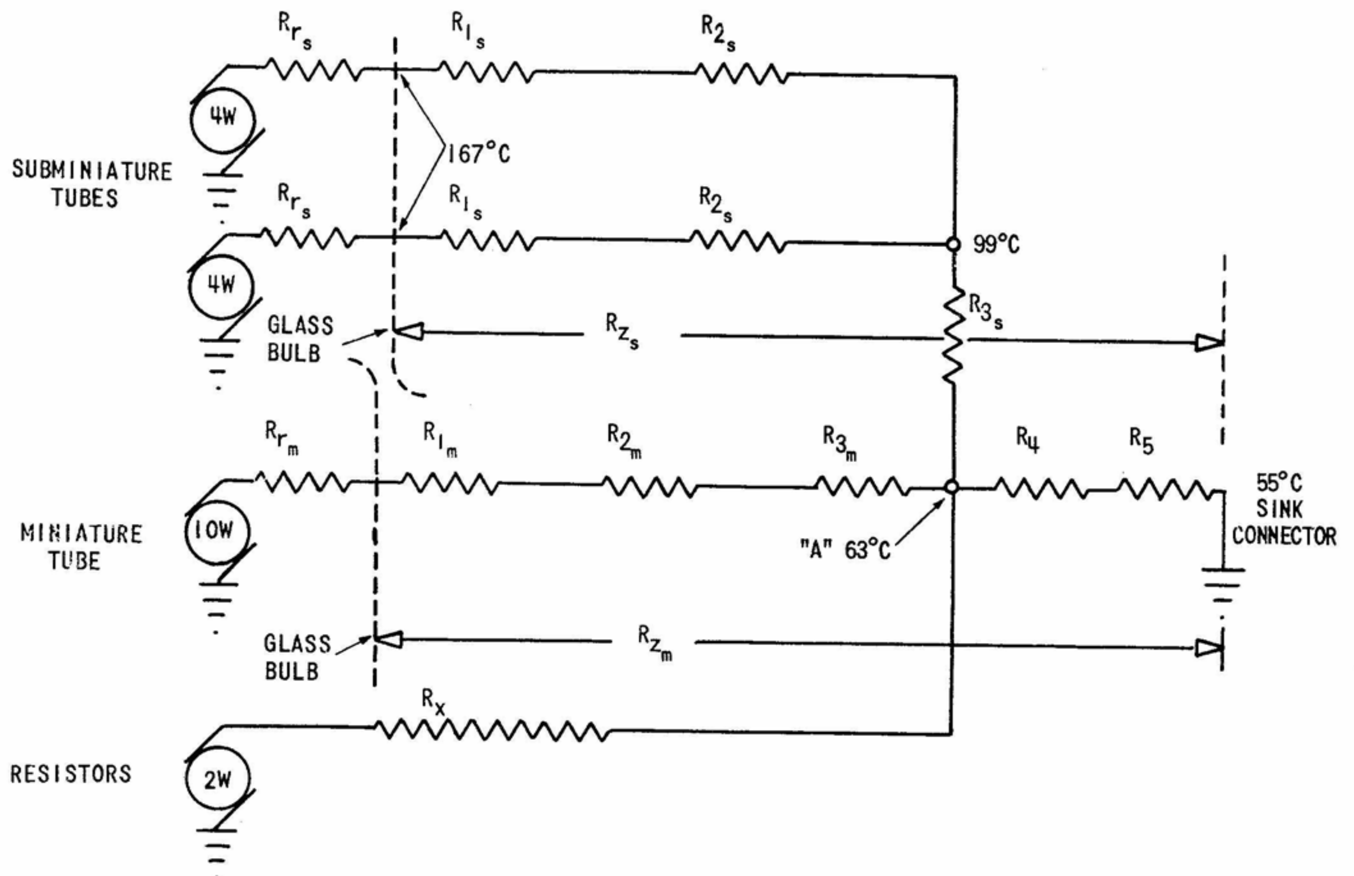
12 x 35 = 420°C/watt for each resistor.

Conversely, at 125°C/watt and a dissipation of .17 watts, the temperature rise will be

.17 x 125 = 21°C.

Note:

This required thermal resistance is considerably greater than the nominal 125°C/watt value obtained in a typical instance. A good thermal design should exhibit much lower values of thermal resistance. If the resistors were dissipating appreciable power, conduction cooling could be provided by fuse-clip mounting to the subchassis.



R_z = THE TOTAL THERMAL RESISTANCE EXTERNAL TO THE TUBES

R_r = INTERNAL THERMAL RESISTANCE OF TUBE

R_1 = THERMAL RESISTANCE OF TUBE SHIELD

R_2 = CONTACT RESISTANCE TO SUBCHASSIS

R_3 = THERMAL RESISTANCE OF SUBCHASSIS

R_4 = THERMAL RESISTANCE OF CHASSIS

R_5 = RESISTANCE OF SINK CONNECTOR

R_x = THERMAL RESISTANCES ASSOCIATED WITH THE SMALL RESISTORS

FIG. 6-10 SCHEMATIC OF HEAT FLOW IN A CONDUCTION COOLED ASSEMBLY

Thus, the thermal resistance, R_x, will be approximately 21/2 or 11°C/watt, if the resistors are judiciously mounted. The net ambient temperature for the resistors is probably 1/3 of the rise between the tube shields and point "A". Since the outer surfaces of the tube shields generally have of the order of 40°C rise above the sink, the net ambient temperature "seen" by the resistors will be:

$$T_n = \frac{40}{3} + 55 = 68^oC.$$

With a 21°C rise, the resistors will operate at 68 + 21 = 89°C. surface temperature.

(2) Electron Tubes

The necessary thermal resistance for cooling the tubes can be determined by several methods. Although hot spot bulb temperature is not an ideal index of the thermal condition of a tube (See Chapter V), it will be used initially in this example. The net thermal resistances of tube shields, presented in Chapter V, which are based upon the relative bulb and plate temperatures and provide a better index, will also be used and the degree of cooling will be determined as design verification.

The maximum reliable hot spot bulb temperature of the electron tubes is assumed to be 175°C. Consequently, the temperature rise must not exceed 175 - 55 = 120°C.

For the miniature tube at 10 watts:

$$R_{zm} = \frac{120}{10} = 12^oC/watt.$$

A tube shield with a thermal resistance less than this value must be used. Shield B (See Chapter V) with a thermal resistance of 10.1°C/watt is selected. This value includes the contact resistance and is the sum of $R1_m$ and $R2_m$ on Fig. 6-10.

The thermal resistance of the subchassis, chassis, and sink connector must, for this tube alone, be less than 12.0 - 10.1 = 1.9°C/watt.

the rise through $R3_m$, R_4, and R_5 = 1.9 x 10 = 19°C

the rise through $R1_m$ and $R2_m$ = 10.1 x 10 = 101°C

Total rise 120°C

For the subminiature tubes, each dissipating 4 watts,

Maximum temperature rise = 120°C

$$R_{zs} = \frac{120}{4} = 30^oC/watt$$

A tube shield with a thermal resistance less than this value must be used. Shield #1 (see Chapter V) with a thermal resistance of 17°C/watt is selected. This value includes the contact resistance when soldered to the subchassis. In this instance, the shield will be riveted and the additional contact resistance is estimated to be 2°C/watt (See Chapter III). Thus, the sum of $R1_s$ and $R2_s$ = 17 + 2 = 19°C/watt for each tube.

(3) Chassis and Subchassis:

The thermal resistance of the subchassis, chassis, and sink connector must, for each tube, be less than 30 - 19 = 11°C/watt. Since R_4 and R_5 will be less than 1.9°C/watt for the miniature tube, an R_3 value of 9°C/watt for each subminiature tube appears to be in order. Therefore, $R3_s$ = 9/2 = 4.5°C/watt for both tubes (in parallel thermally),

the rise through $R_{3s} = \dfrac{4.5^\circ C}{w} \times 8w = 36^\circ C$

the rise through R_{1s} and $R_{2s} = \dfrac{19^\circ C}{w} \times 4w = \underline{76^\circ C}$

Rise, excluding R_4 and R_5 = 112°C

120 - 112 = 8°C = the allowable rise through R_4 and R_5

Thus, point "A" on Fig. 6-10 = 55 + 8 = 63°C.

Assume R_{3s} is to have an average heat flow path length of 2 inches and is to be made for 1/16 inch thick chrome copper strip having a thermal conductivity of 8.25 watt-in/°C-in.^{2}

Since: $$A = \frac{L}{kR} \qquad (54) \quad (D.E.)$$

Then: $$A = 2 \text{ in.} \times \frac{1}{4.5}\,\frac{\text{watt}}{^\circ C} \times \frac{1\ ^\circ C\text{-in.}^2}{8.25 \text{ watt-in.}} = .054 \text{ sq.in.}$$

$$\frac{.054}{.0625} = .86 \text{ in. wide}$$

Thus, the subchassis supporting the subminiature tube can be 7/8 in. wide and soldered or welded to the chassis.

Since point "A" is at 63°C., which is 8°C above the sink connector temperature, then the thermal resistance of $R_4 + R_5$ must be such as to limit the rise to 8°C at 20 watts. Thus,

$$R_4 + R_5 = \frac{8}{20} = .4^\circ C/\text{watt}.$$

Assume that the thermal resistance R_5 of the sink connector is one-half of this total or .2°C/watt. If two studs are used, having an effective length of 1/8 in., then each will have a thermal resistance of .4°C/watt and the thermal resistance per unit length will be 3.2°C/watt-in.

Chrome copper having a thermal conductivity of 8.25 $\frac{\text{watts-in.}}{\text{in.}^2\text{-}^\circ\text{C}}$ will be used:

Thermal resistivity of chrome copper = $\frac{1}{8.25} = .121 \frac{^\circ\text{C-in.}^2}{\text{watt-in.}}$

Required conductor area = $\frac{.12}{3.2}$ = .038 sq. in.

$\frac{\pi d^2}{4} = .038$ $\qquad$ d = .22 in. dia.

Use two 1/4 in. dia. studs! The contact resistance between the studs and the chassis must be minimized because of the requirement for only 8°C rise in both the chassis and studs. Therefore, it will be desirable to weld or solder the studs to the chassis essentially to eliminate this contact resistance.

The chassis size can now be determined. Assume the subchassis for the tubes are to be mounted so that the heat paths from the sink connectors to the chrome copper subchassis are 1/2 inch long and the chassis is to be $2\frac{1}{2}$ inches wide.

$R_4 = R_5 = .2^\circ$C/watt.

Required conductor area = $\frac{.121}{.2 \times 2}$ = .3 sq. in.

The chassis thickness = $\frac{.3}{2.5}$ = .12 = 1/8 in.

Assume that the chrome copper subchassis for the miniature tube will have an average heat flow path length of 1.5 inches.

The thermal resistance of R_{3_m} = 1.9 - (.4 x 2) = 1.1°C/watt or .66°C/watt-in.

Required conductor area = $\frac{.121}{.66}$ = .184 sq. in.

Make the subchassis for the miniature tube 1.5 in. wide x 1/8 in. thick and solder or weld it to the chassis.

The temperatures can be predicted:

55 + 8 + 36 + 76 = 175°C. net bulb temp. of the subminiature tubes, as originally established.

55 + 8 + 11 + 101 = 175°C. net bulb temp. of the miniature tube, as originally established.

(4) Check of Thermal Index of Tubes:

"Good" cooling of the tubes is required for operation at such a high sink temperature. Consequently, the ratio of external to internal thermal resistance for the tubes should be of the order of .5 (See Chapter V).

The temperature of the ultimate sink is estimated to be 25°C., either that of the atmosphere or the sea. The thermal resistance from the ultimate sink to the sink connector is:

$55 - 25 = 30°C$ rise

$$R = \frac{30}{20} = 1.5°C/\text{watt}$$

R_{z_m} for the miniature tube $= \frac{175 - 55}{10} = 12°C/\text{watt}$

Total $R_z = 12 + 1.5 = 13.5°C/\text{watt}$

R_{r_m} from Chapter V = 25°C/watt at 12 watts

$\frac{R_2}{R_{r_m}} = \frac{13.5}{25} = .54$, i.e., the cooling index is acceptable

R_{z_s} for the subminiature tubes $= \frac{175 - 55}{4} = 30°C/\text{watt}$

R_{r_s}, from Chapter V = 52°C/watt at 3 watts

$\frac{R_{z_s}}{R_{r_s}} = \frac{30}{52} = .57$, i.e., the cooling index is acceptable

Note that consideration has not been given to heat transfer from the tubes and their shields by radiation and convection. The heat transferred by these modes will be small compared to that by conduction through the low thermal resistance paths. Thus, the thermal contributions of radiation and convection can constitute a design safety factor.

b. Typical Case Histories

(1) Metal Block Chassis:

This type of construction (see Fig. 6-11) involves a die-cast or a machined metal block chassis with the tubes inserted in cylindrical holes near the cooled surface. The metal block forms the outside of the case of the assembly. The non-heat-producing parts may be mounted in the center of the block or they may be placed at an edge of the block, if desired.

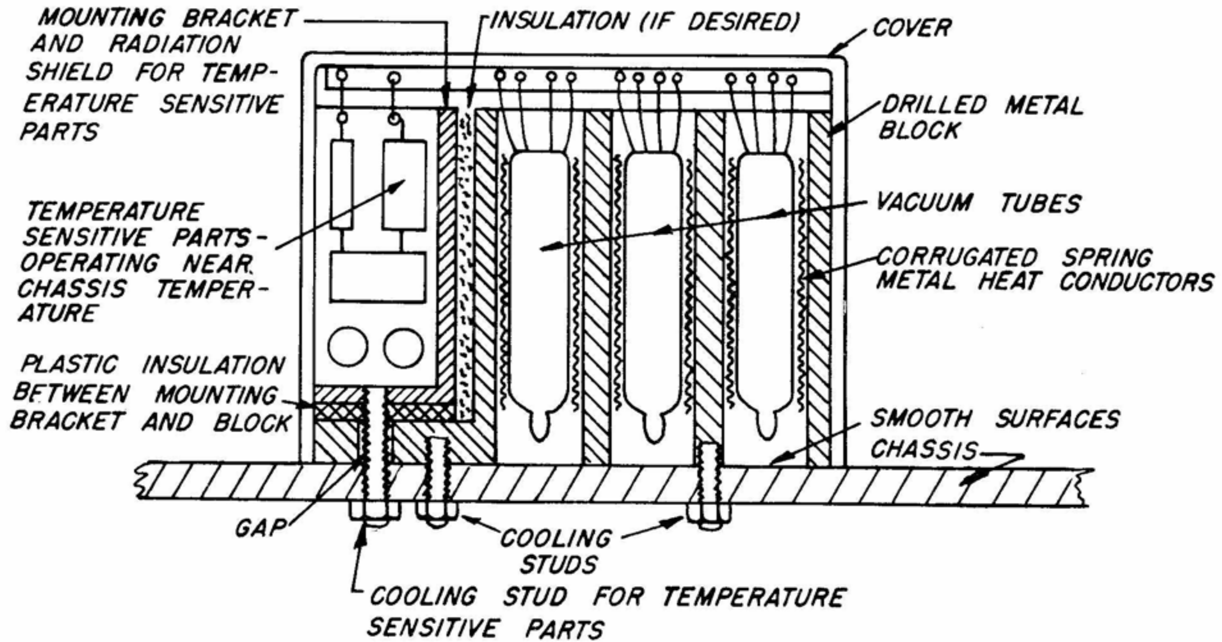


Fig. 6-11a RECOMMENDED CONSTRUCTION PRACTICES

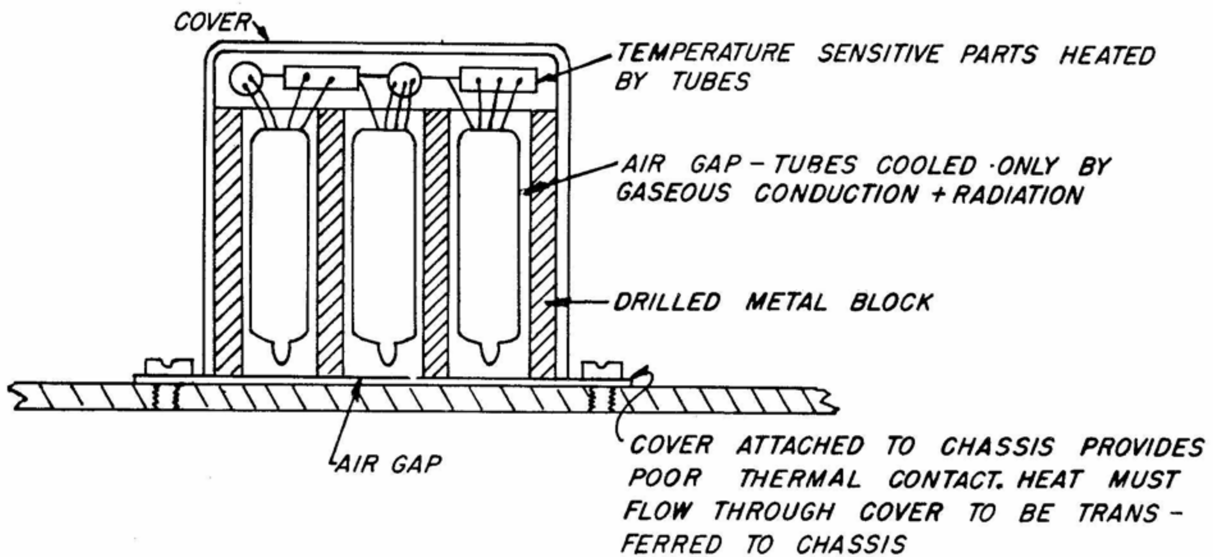


Fig. 6-11b POOR DESIGN

Fig. 6-11 DRILLED METAL BLOCK CONSTRUCTION

The assemblies, Ref. 46, developed for operation at high temperatures are representative of metal block chassis construction. In general, these units closely resembled the assembly shown in Fig. 6-11.

Case (a)

One assembly, which was 5 in. x 1 in. x 2 in., dissipated 12 watts in 8 tubes and exhibited a 15°C rise to the top of the case. The hot spot at the base of the tubes was 56°C above the sink connector temperature.

Case (b)

Another assembly, developed by NEL, utilized a cast aluminum case and tube holder (See Fig. 6-12). This unit was 4 1/2 in. x 3 3/4 in. x 2 in. Originally, the heat from the tubes was transferred only by radiation and gaseous conduction into the top of the case. Consequently, a large thermal gradient was obtained between the bulbs and the case. For example, at 30 watts dissipation, the bulb temperature rise was 155°C above that of the bottom of the case and the plate temperature exceeded 450°C. The addition of metal inserts or braid between the tube bulbs and the top of the case will significantly reduce this gradient. The thermal resistance between the top and bottom of the case through the rubber gasket is approximately .63°C/watt. This is acceptable unless high heat concentrations or high environmental temperatures are anticipated. Under these conditions it would be advisable to incorporate a metal gasket in place of the rubber gasket between the top and bottom of the case and also make sure that the matching surfaces are smooth and flat. The thermal resistance between the bottom of the assembly and the cold chassis (sink) was .33°C/watt.

(2) Metal Chassis Construction:

This type of construction is frequently used. Case wall thicknesses of the order of .125 in. are commonly encountered and thicknesses of .25 in. have been utilized in equipments of high heat concentration.

Case (a)

A typical example is presented in Fig. 6-13a. These assemblies (Ref. 48) utilized .03-in.-thick copper cases and .02-in.-thick beryllium copper subchassis which were spring loaded against the inside of the cases to accommodate expansion. The tube shields were bolted to the subchassis near the case so that the heat flow path was as short as possible.

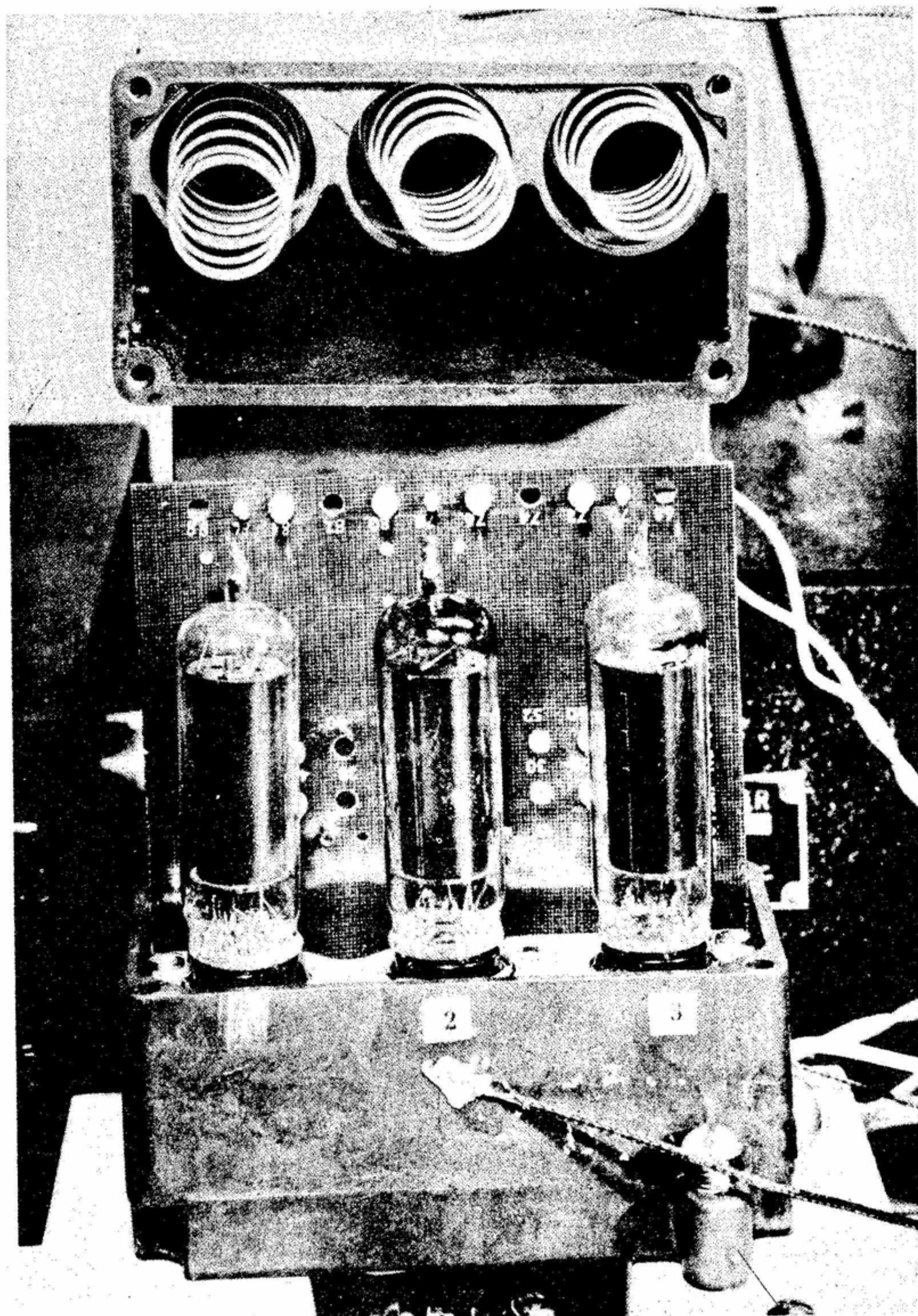
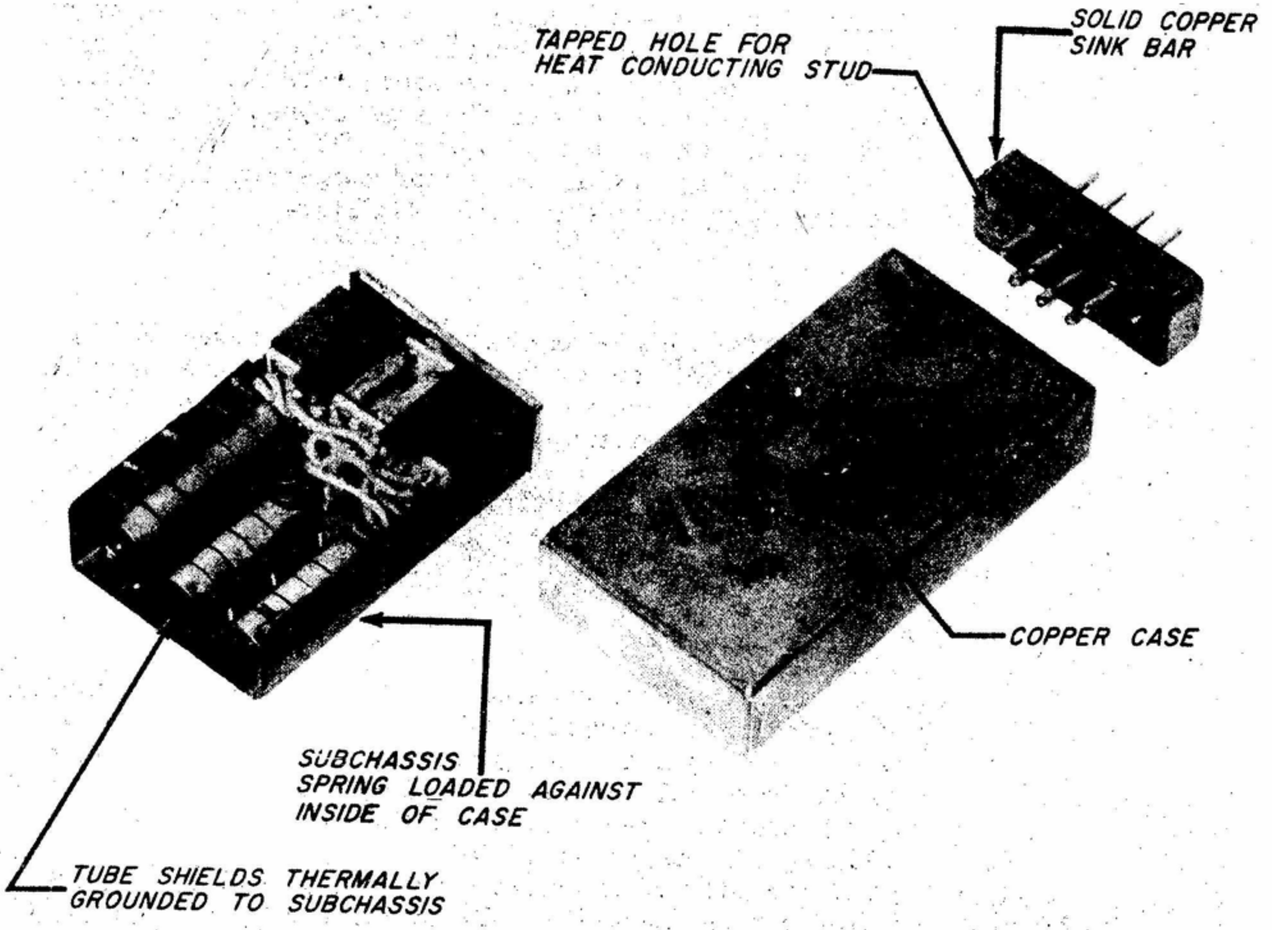
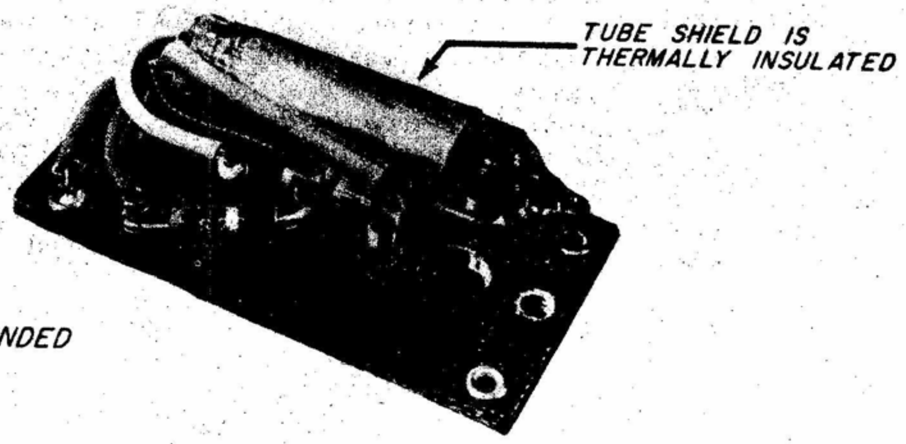


FIG. 6-12 NEL PACKAGE WITH THERMATRONS

a) RECOMMENDED CONSTRUCTION PRACTICE – METALLIC CONDUCTION COOLING TO CHASSIS





b) NOT RECOMMENDED

Fig. 6-13
CONSTRUCTION METHODS

With a heat concentration of 2.1 watts/cu. in., i.e., a total dissipation of 10 watts in 4.72 cu. in.; the net thermal resistance of a subminiature tube dissipating 5.5 watts was 13°C/watt to the sink connector. Tubes dissipating 2.25 watts each had a net thermal resistance of 11°C/watt to the sink connector. The thermal resistance through the chassis and subchassis was .3°C/watt and the resistance between the sink connector and the cold chassis (sink) was .32°C/watt. These data were obtained at 113°C. sink connector temperature, the maximum rated temperature for the assemblies.

Case (b)

A miniaturized high-temperature 60-cycle power supply associated with the above assemblies dissipated approximately 10 watts and was rated for operation at 110°C. sink connector temperature. It exhibited 15°C/watt thermal resistance to the hot spot on the rectifier tubes, each dissipating 3.5 watts. The laminations of the filter choke had a 9°C rise and laminations of the power transformer exhibited a 11°C rise above the sink connector.

Case (c)

The conduction-cooled assembly of Fig. 6-14 was constructed to demonstrate the design techniques mentioned herein. The over-all dimensions were 3 in. x 5/8 in. x 2 in. and the chassis material was .06-in.-thick brass. Note that the sink connectors are welded and that the tube shields are soldered to the chassis. Considerably greater power than would normally be dissipated in such an assembly was injected during the tests. Table XXI presents the thermal performance. Mounting to a "sink" made a significant improvement, in spite of the use of relatively long and thin sink connectors made of a material with a thermal conductivity one-fourth that of copper. Even so, the thermal resistance of the sink connectors, including contact resistance to the sink, was 2.4°C/watt. The point is that the thermal conductivity of the heat-conducting material must receive careful consideration when significant power is being transferred, i.e., at 23.6 watts dissipation, a 59°C gradient existed across the sink connectors.

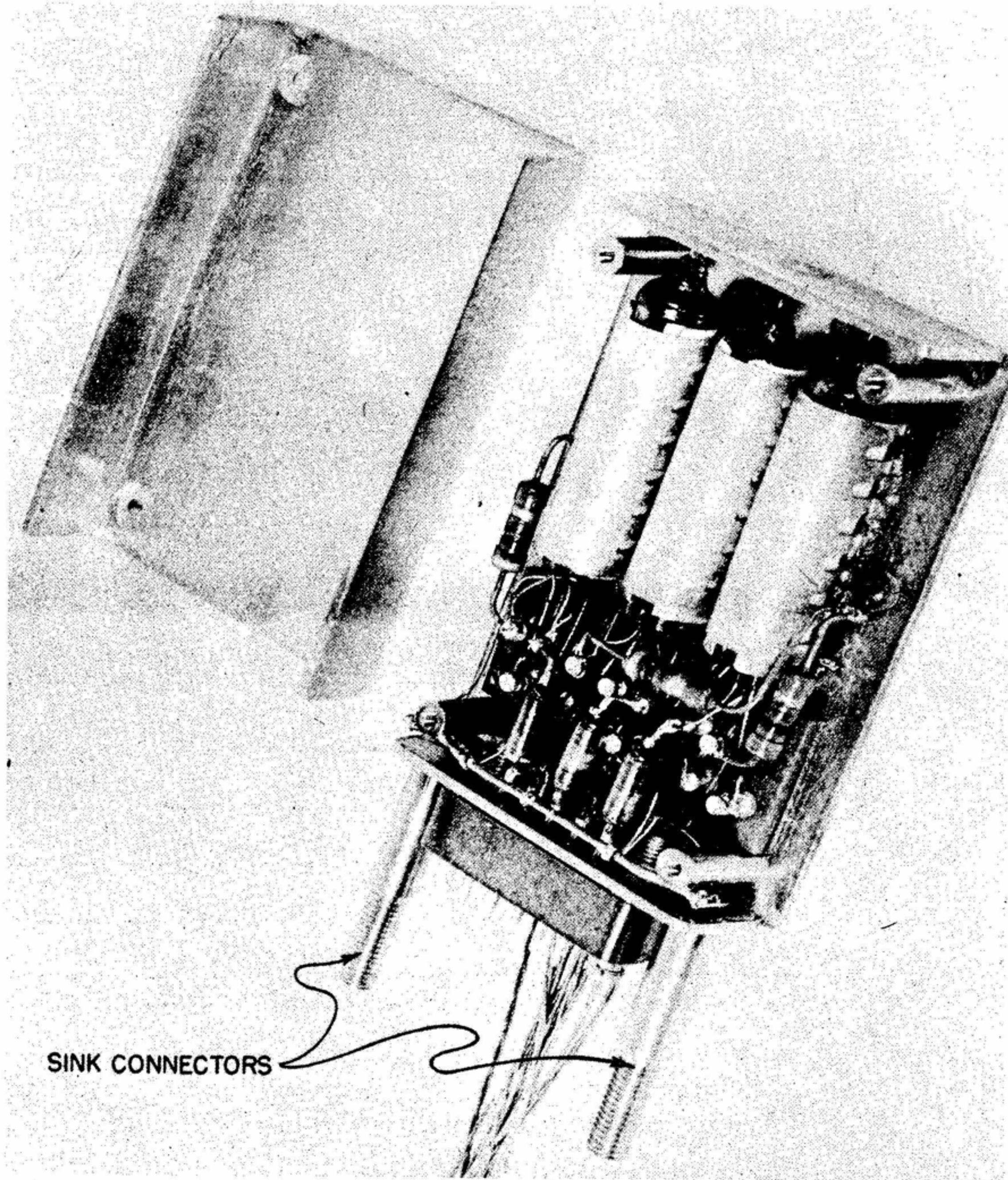


FIG. 6-14 CONDUCTION COOLED ASSEMBLY

TABLE XXI.

THERMAL PERFORMANCE OF AN EXPERIMENTAL ASSEMBLY

In Free Air at 26°C.

Location	Temperature Rise Above "Ambient" in °C		
	8 Watts	15.6 Watts	23.6 Watts
Case	57	109	152
Tube Bases	101	158	216
Tube Shields	62	114	170
Terminal Board	63	117	165

Mounted on 40°C. Sink

Location	Temperature Rise Above Sink in °C		
	8 Watts	15.6 Watts	23.6 Watts
Top of Case	22	57	94
Bottom of Case	11	37	59
Tube Bases	74	119	159
Tube Shields	24	61	94
Terminal Board	28	64	96

Several examples of thermal designs which are not recommended follow:

Case (d)

The single-tube assembly of Fig. 6-15 is well designed electrically. The thermal design omits a metallic heat path to the sink. Further, the tube shield is shiny and should be blackened. Fig. 6-16 shows the effects of blackening only the inside of the shield. Note the significant reduction of plate temperature. The temperature of the tube base at 10 watts dissipation was 175°C. with the shiny shield and 160°C. when the inside of the shield

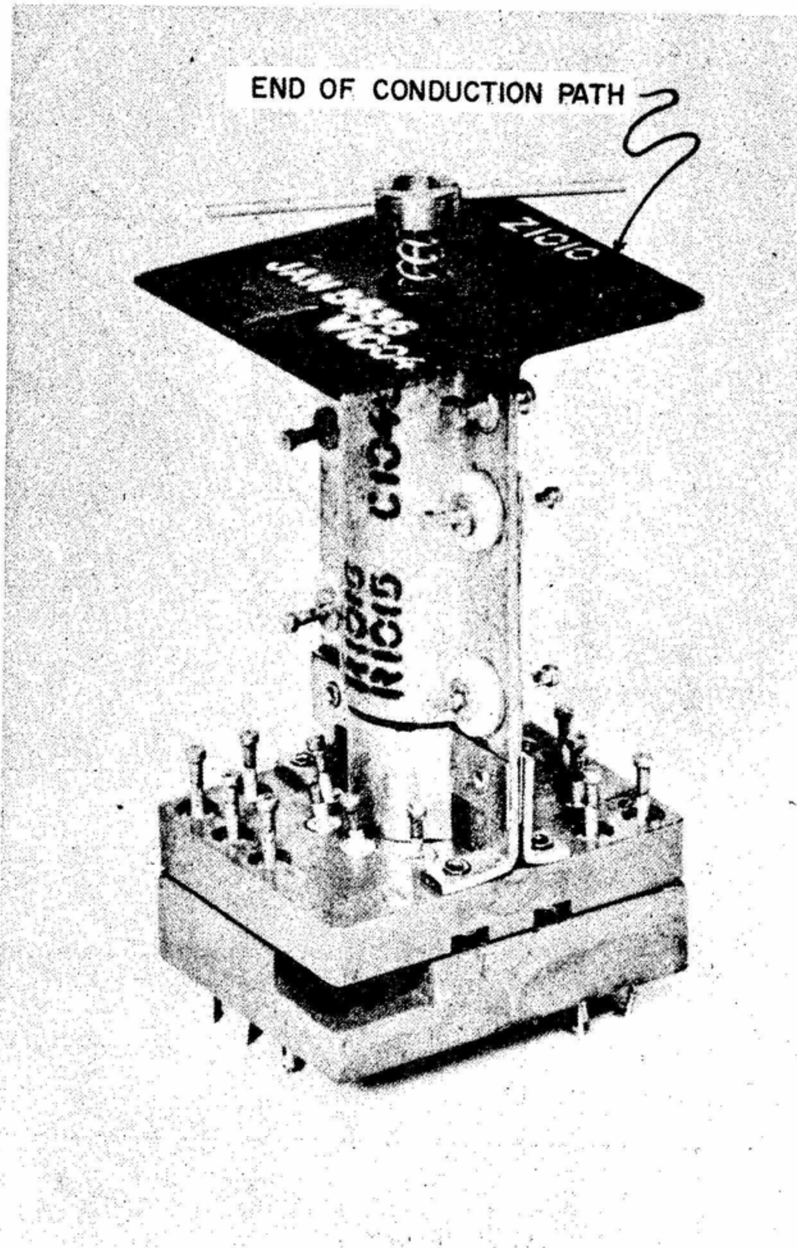


FIG. 6-15 SINGLE TUBE ASSEMBLY

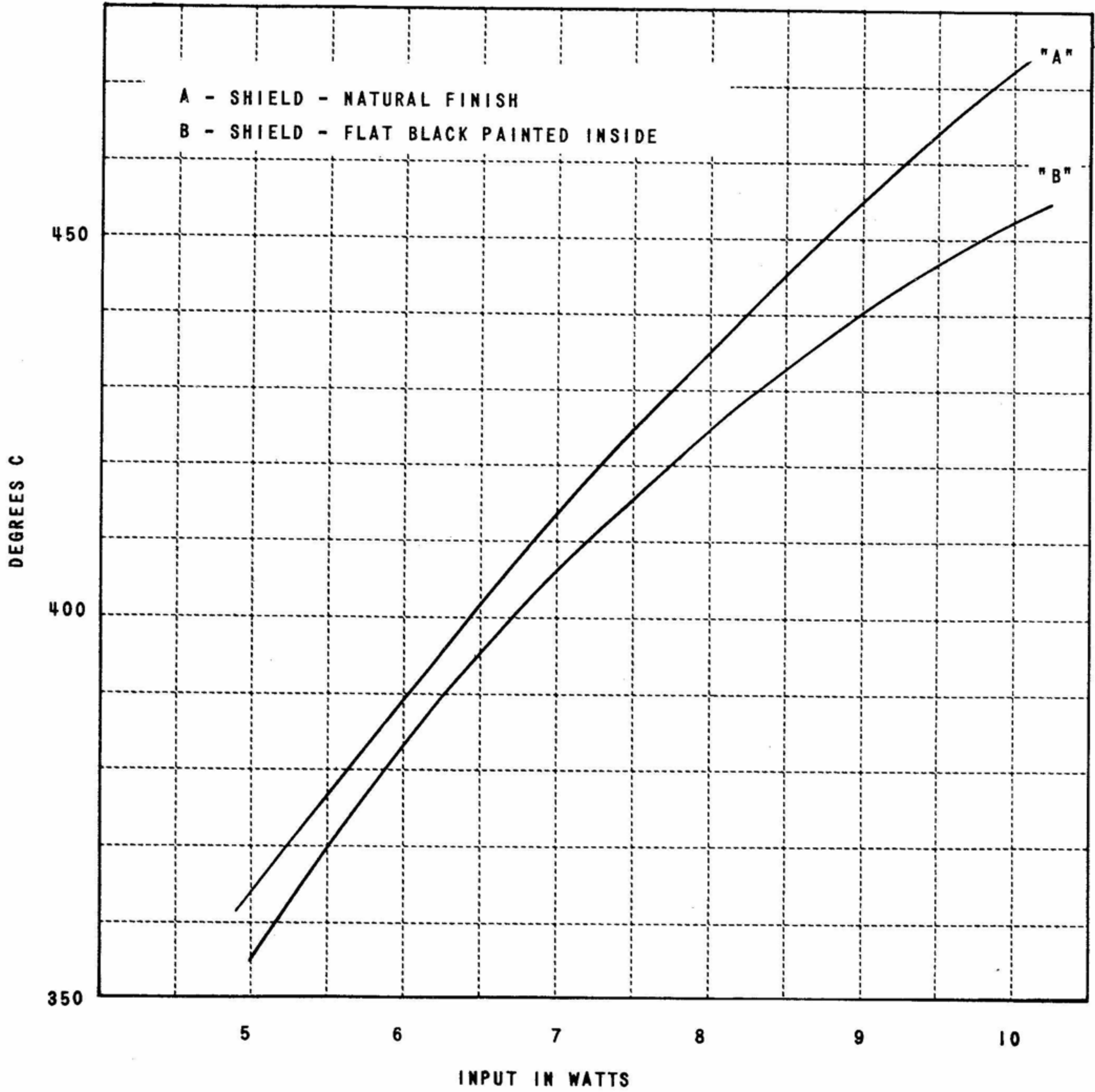


FIG. 6-16 PLATE TEMPERATURE TYPE 5902 THERMATRON TUBE IN JAN. #5636 ASSEMBLY

was blackened. This assembly was designed to reject its heat by radiation and convection from the blackened plate at its top. It is futile to attempt to incorporate a local heat sink as part of an assembly unless the heat concentration is extremely low and the thermal environment is at a low temperature. Even so, the local sink plate probably would reject most of its heat by radiation and require a relatively high temperature difference for reasonable heat transfer. It appears that the above-measured temperatures would be considerably higher than indicated in an actual installation, since these data were obtained in free air at 25°C. The thermal performance of this assembly could be made satisfactory by blackening the shield and by providing a metallic heat path from the subchassis and shield to a supporting metal structure, which is thermally attached to the ultimate sink.

Case (e)

The modular assembly shown on Fig. 6-17 is not recommended. Note that the heat from the tube will be transferred into the adjacent electronic components. Further, the tube can only reject heat by radiation and convection. Such a configuration will normally lead to overheating and subsequent impairment of reliability.

Case (f)

Another assembly which is thermally similar to that of Fig. 6-15 is presented on Figs. 6-18 and 6-19. This unit incorporates a single miniature or four subminiature tubes, each provided with a shield and heat conductor to the local sink at the top of the assembly. As in the aforementioned instance, the sink at the top of the assembly is inadequate for proper cooling. The bases of the four subminiature tubes were at 234°C. at a total dissipation of 20 watts and the plastic case became scorched. Fortunately, the thermal resistance from the tubes to the local sink is low, being of the order of 1.5°C/watt. If the center post was connected to a path to the ultimate heat sink instead of the local sink, the temperatures would be drastically reduced and the thermal design would be good. The point is that, due to the omission of a connection to the ultimate sink, a good thermal design was made ineffectual.

3. Plastic Embedment

Heat transfer in embedded assemblies is primarily by conduction through the plastic in conjunction with some metallic conduction in the wiring. A number of 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

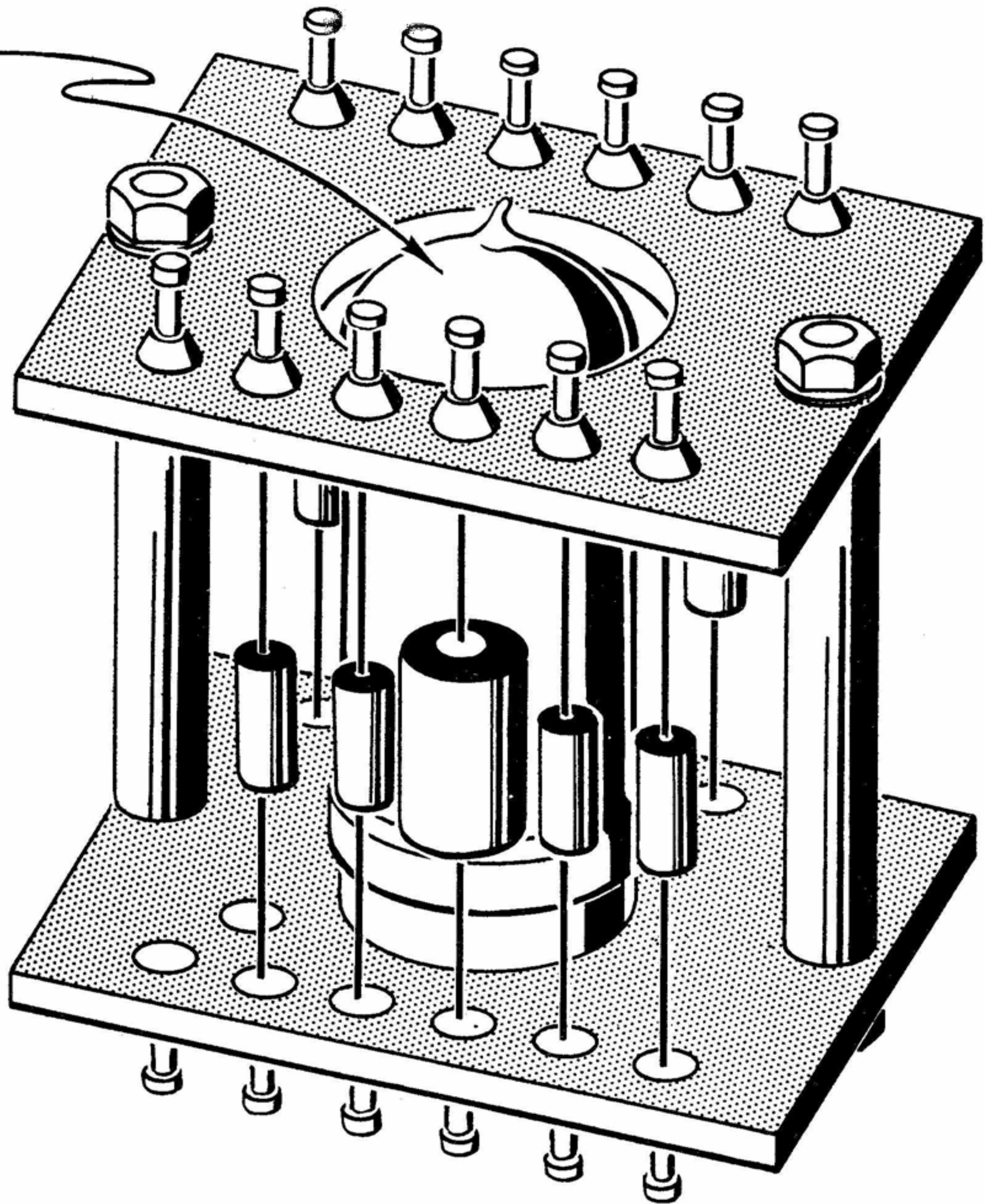


FIG. 6-17 MODULAR MOUNTINGS

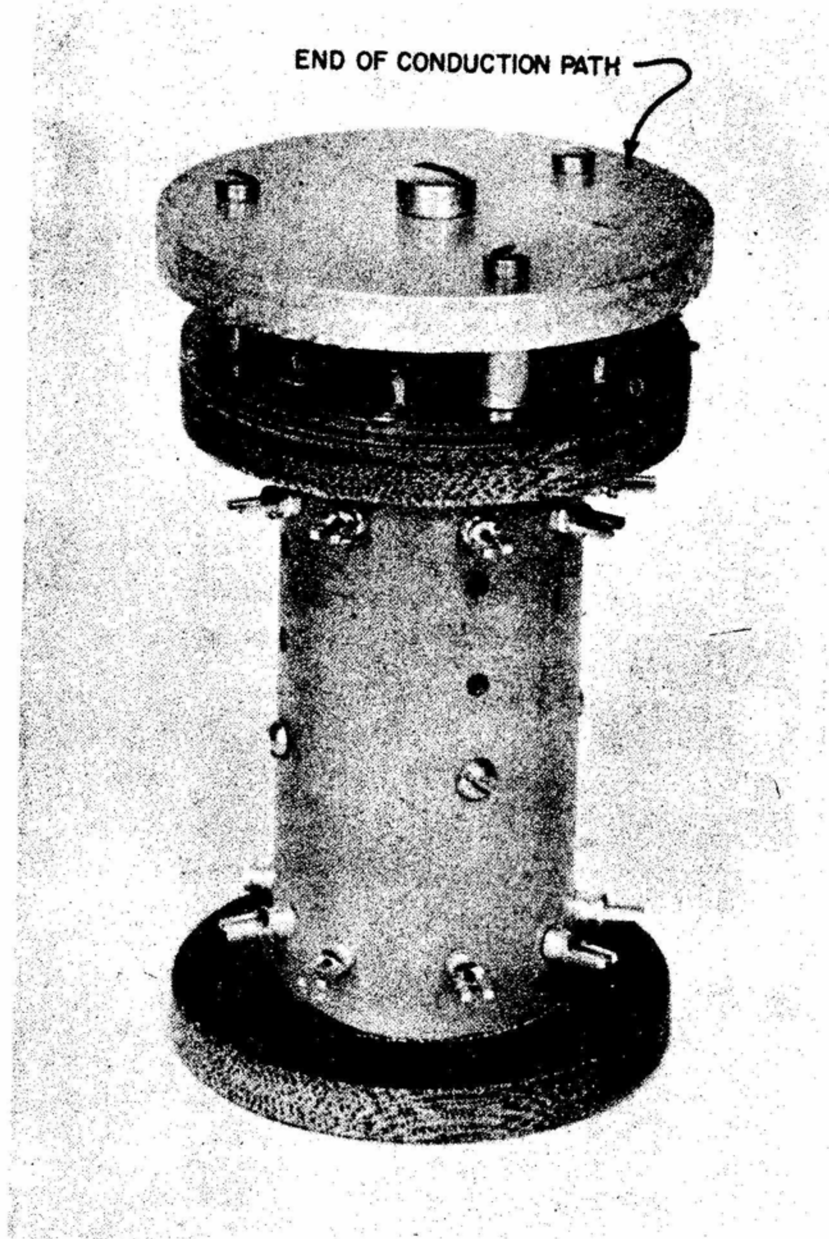


FIG. 6-18 MODULAR ASSEMBLY

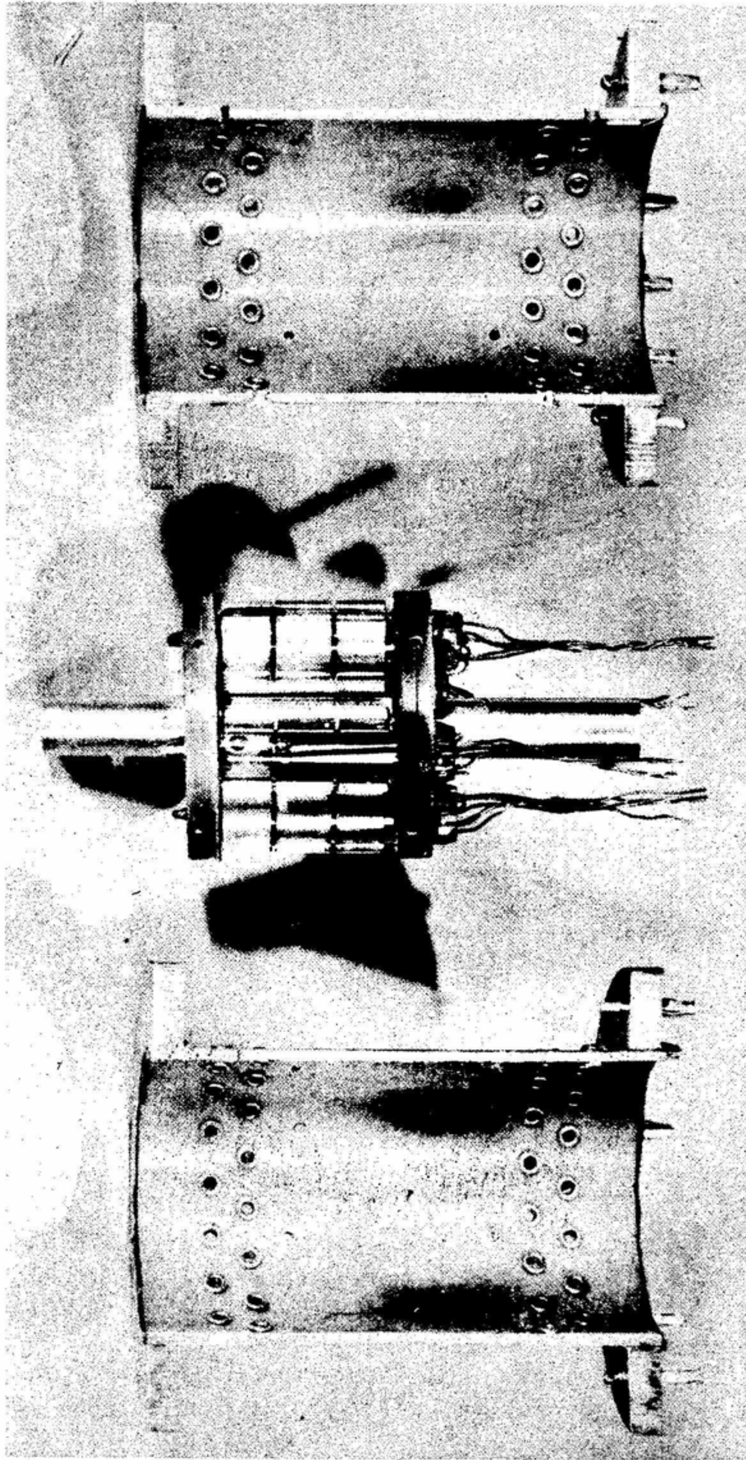


FIG. 6-19 INTERNAL VIEW OF CYLINDRICAL ASSEMBLY

construction method should be used above 0.25 watts/cu.in. The following is limited to those types which rely on heat conduction through the plastic as the primary cooling mode.

In general, due to the poor thermal conductivity of plastics (see Appendix), 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.

In a typical instance, four type T-2 diodes were embedded with casting resin to form a cylinder 5/8 in. diameter x 1 3/4 in. long (Ref. 51). Difficulty was experienced with cracked and overheated resin at 3 watts total heat dissipation. It was necessary to incorporate aluminum particles in the resin 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. However, no metal-filled material has yet been qualified under MIL-I-16923.

Extreme care must be utilized in the embedment of vacuum tubes since excessive temperatures can easily be obtained even at relatively low heat concentrations. Ref. 54 describes some of the related problems. The thermal conductivity of plastics is generally of the same order of magnitude of that of "fire brick" and, without due care, potted assemblies can readily become overheated.

4. Convection and Radiation-Cooled Assemblies

This type of assembly is not normally recommended, especially for miniaturized devices, since a low heat concentration is mandated in order to prevent excessive temperature rise. In general, unit heat dissipations should not exceed .25 watts/sq.in. With special care and high temperature component parts, unit heat dissipations approaching .5 watts/sq.in. can be obtained. Unfortunately, assemblies which reject their heat by only radiation and convection will, if closely packaged, exhibit considerable thermal interaction and the allowable unit heat dissipations may be lower than mentioned above. In such instances, only those surfaces which do not "see" other assemblies should be considered as the pertinent heat-rejecting surfaces, and design calculations should be predicated on this assumption.

The nomographs of Chapter III can be used to predict the surface temperatures of convection and radiation-cooled assemblies. Fins may be added, if desired, to decrease the unit heat dissipatio... The major axis of the fins should be vertical. However, deep cooling ribs or fins lead to increased air friction and tend to reduce the natural convection air-flow rates. A "good" rule of thumb is to consider the fins in calculations as having only one-half of their actual area. The vertical rather than the horizontal surfaces of the assemblies should be finned. If the fins are longer than one inch in length, they should be broken or staggered so as to have an effective

length of approximately one inch. This tends to increase the air-flow rates and, under some conditions, turbulence will be increased.

D. THE MOUNTING OF ASSEMBLIES

Chassis must be designed to accommodate the rejected heat from the associated assemblies and thereby provide a low-resistance thermal path between the assemblies and the sink. It is not recommended that the assembly chassis be used as a heat sink. Rather, the metal chassis should only form a heat flow path.

Assembly chassis will generally be capable of providing a low thermal resistance by metallic conduction because the metal required for structural reasons will usually have a large cross-sectional area. However, the thermal resistance should be determined for design purposes and, if necessary, the cross-sectional area should be increased to provide the required thermal resistance. Thermal difficulties will usually be encountered at the ends of the heat flow path through the chassis rather than in the chassis itself.

1. Methods of Attaching Assemblies

Chassis should be provided with features necessary to thermally match the assemblies. The thermal connection can be made with studs or bolts of the proper thermal size or the common surfaces of the chassis and assembly can be used provided that they are smooth, clean, and under a reasonable mounting pressure. The important point is that the contact resistance between the heat-conducting chassis and the assemblies must be reduced to the desired value. This thermal resistance can be determined based upon the data presented in Chapter III and item C of this chapter. If the surfaces are not extremely smooth, it is possible to reduce the thermal resistance by utilizing thin flexible metal gaskets.

2. Methods of Removing Heat from the Chassis

The thermal resistance from the chassis to the sink must be extremely low, since the total heat from all of the assemblies in the unit is involved. Consequently, when the chassis is mounted in, for example, a rack or a cabinet, considerable attention must be given to the transfer of heat in the enclosure. Metallic conduction is again recommended as the primary heat transfer mode. Since points of attachment of the chassis and the enclosure must serve as the heat flow path, the joints must be thermally adequate for conducting all of the rejected heat. All paint must be removed; large, smooth matching surfaces are in order and plenty of bolts or screws should be used. Fig. 6-20 presents data on such joints. Thus, it is obvious that hinged chassis and sliding drawer construction alone is thermally inadequate for devices having heat concentrations exceeding .25 watts/sq.in. If such mechanisms are necessary, heat-conducting studs and joints must also be provided.

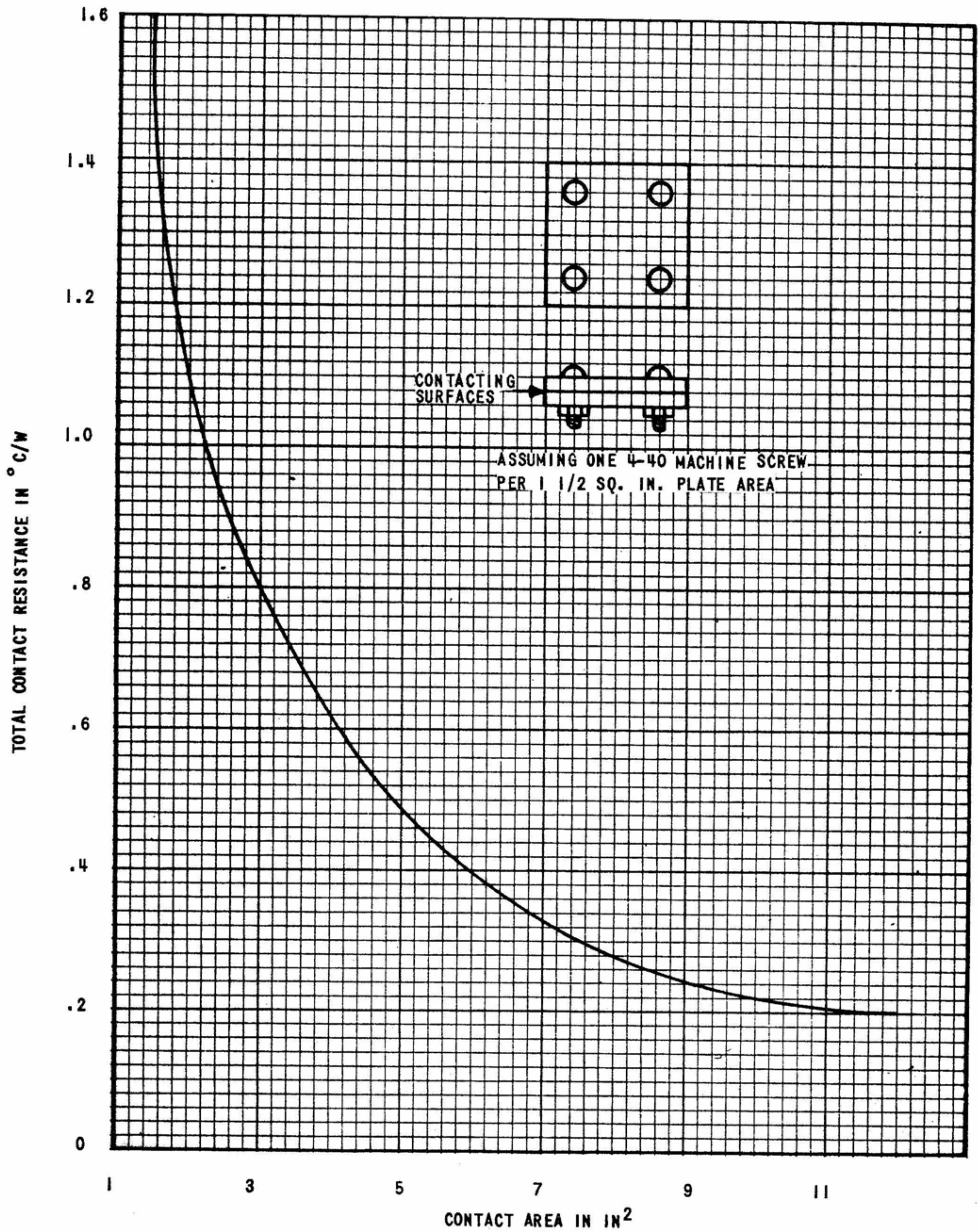


FIG. 6-20 CONTACT AREA VS TOTAL THERMAL CONTACT RESISTANCE FOR TWO METAL PLATES

E. THE THERMAL DESIGN OF EQUIPMENT ENCLOSURES

1. General

Equipment enclosures should be constructed so that a reasonably low thermal resistance is obtained between all surfaces and the chassis. The metal normally utilized for enclosures will usually provide an adequately low thermal resistance. As in chassis, the primary resistance to heat transfer by conduction will be that of the joints. Consequently, all metal-to-metal joints must be clean, smooth, have large contact areas and proper contact pressures. Joints having a low thermal resistance will also tend to minimize leakage of electrical fields from the equipment. Fig. 6-21 presents the heat flow paths in an equipment case.

2. Enclosures as Heat Sinks

Equipment cases can be utilized as sinks to dissipate the internally developed heat by convection and radiation to their environment. The unit heat dissipation must be low and the thermal environment must also exhibit a low temperature, otherwise the largest temperature rise in the heat transfer system will exist between the case and its environment. It is also necessary that the heat-rejecting surfaces be as close as possible to the heat sources. Small temperature gradients can be obtained over appreciable distances only when large metallic heat conductors are used.

The maximum unit heat dissipation for enclosures should not exceed .25 watts/sq.in. Even so, relatively high temperatures can be obtained. Care also must be exercised to make sure that the rejected heat is not introduced into other nearby electronic equipment. Whenever convection or radiation is relied upon for the dissipation of heat from enclosures, consideration should be given to the effects of compartment air temperature and thermal interaction with nearby equipments.

The case temperature rise can be predicted by several methods, i.e., calculation based upon the equations or the nomographs presented in Chapter III, or the curves presented on Figs. 6-22 and 6-23 (Ref. 56). Pertinent sample calculations are presented in Chapter III. The curves of Figs. 6-22 and 6-23 do not always provide accurate solutions because they largely disregard variations of shape factor. Note that the external surface of the case should always have a high emissivity.

Consideration should also be given to providing adequate ventilation for cases cooled by convection and radiation. Equipment, so cooled, cannot normally be installed in confined spaces without excessive temperature rise. Long, slender cases with the large dimension vertical are preferred. Fins, discussed earlier in this section, may be used to decrease the unit heat dissipation. If it appears that convection and radiation will occur internally from the assemblies (and heat sources), ventilation holes or louvers are desirable. In general, the holes should be at least 3/8 inch in diameter and should be located at the top and bottom of the equipment enclosure.

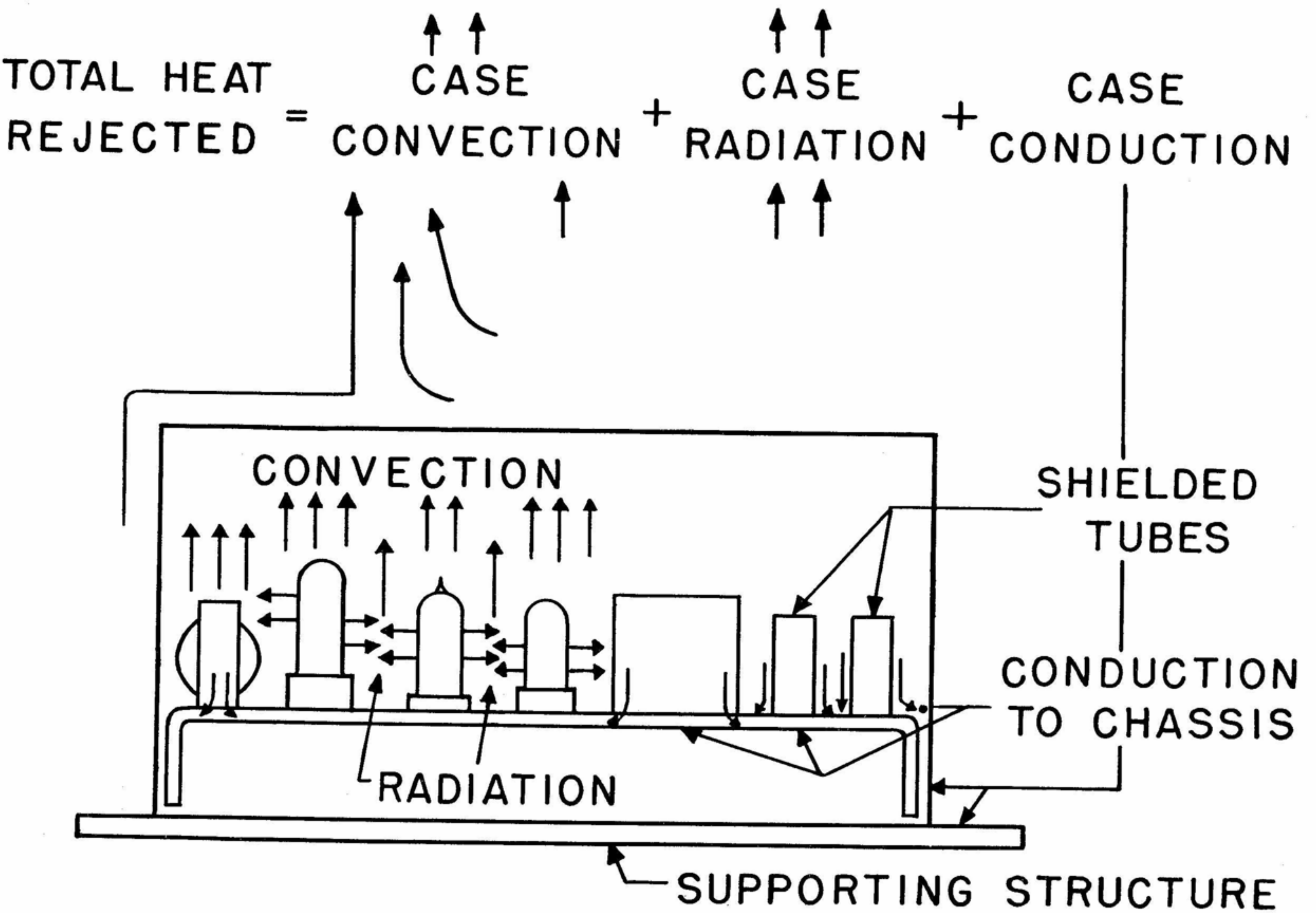


FIG. 6-21 NATURAL HEAT FLOW PATHS IN AN ELECTRONIC EQUIPMENT CASE

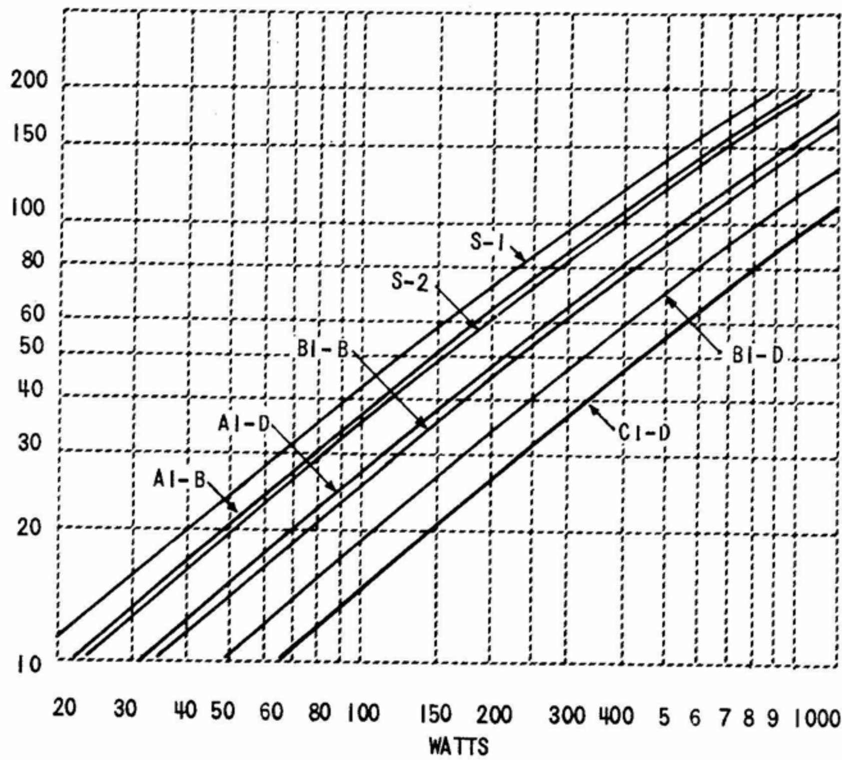


FIG. 6-22 (a) TEMPERATURE RISE OF JAN-C-172A CASES (MAXIMUM HEIGHT)

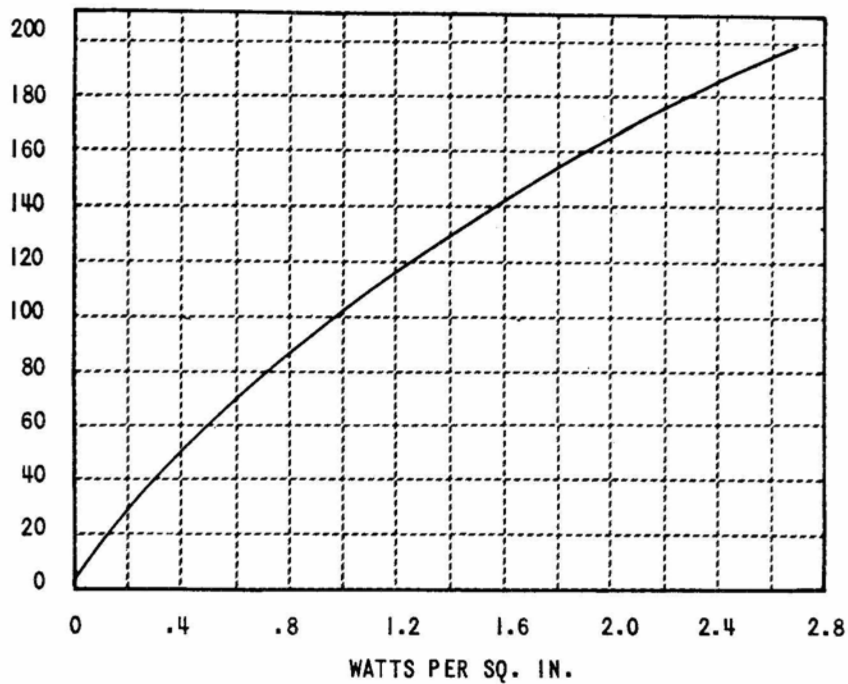


FIG. 6-22 (b) TEMPERATURE RISE OF NONSTANDARD CASES (2 X 2 X 1 INCHES TO 18 X 18 X 18 INCHES)

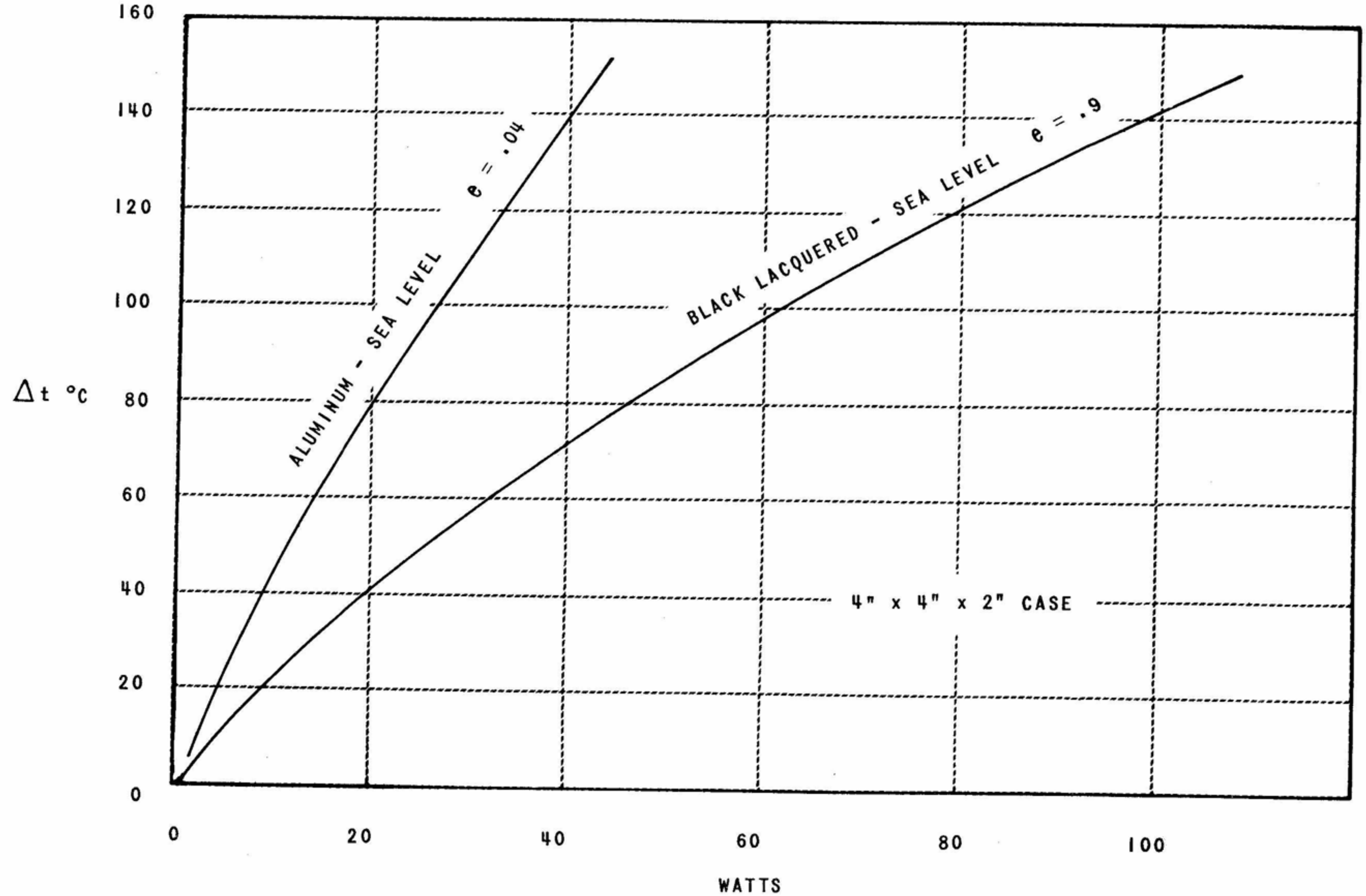


FIG. 6-23 EFFECT OF CASE SURFACE ON TEMPERATURE RISE

The total area of the ventilation holes or simple louvers (either inlet or outlet) can be predicted by:

$$A = .568 \frac{q}{\rho (t_2 - t_1)^{3/2}} \qquad (55) \qquad (D.E.)$$

where:

A = the area of the inlet or outlet openings in sq. in.

q = dissipated power in watts

t_1 = external air temperature in ^{o}C.

t_2 = internal air temperature in ^{o}C.

ρ = density of the internal air = $\frac{22.1}{t_2 + 273}$ lbs./ft.^{3}

Note:

The above does not apply if $t_2 - t_1$ exceeds 35°C. The error for larger temperature difference may become significant and it is recommended that 35°C. be utilized as the maximum value for $t_2 - t_1$. Temperature differences larger than 35°C are not normally recommended, since excessive part temperatures will usually be encountered.

3. Enclosures as Heat Flow Paths to Ultimate Sinks

If possible, equipment enclosures should be utilized to conduct the heat to the ultimate sink. In certain instances, the enclosures can be attached to structural members which are thermally connected to the ultimate sink, for example, the hull of a ship. As mentioned above, the thermal resistance of the joints must be minimized and the heat flow path must be short, otherwise the temperature rise will be large and convection and radiation will be initiated with probable thermal interaction with nearby units. Further, the thermal resistance of the metallic heat flow path should be determined. It appears that the cross-sectional area of some equipment cases will not be adequate to provide the required low thermal resistance. Consequently, it may be necessary to increase the case thickness or utilize metals with large thermal conductivities in order to achieve the required thermal resistance.

VII METHODS OF IMPROVING THE EFFICIENCY OF ELECTRONIC EQUIPMENT

A. GENERAL

It is well known that the electrical efficiency of most electronic circuits is very low, ranging from almost zero to a few per cent. Careful design will, in certain instances, increase the efficiency and help to alleviate the heat removal problem by reducing the dissipated power. It is recommended that each circuit be reanalyzed during its development and, if necessary, be redesigned to obtain the highest practical efficiency prior to initiating the heat removal design.

To design efficient electronic circuitry, it is necessary to examine each electronic stage individually for useless power being dissipated as heat during stand-by and full-output conditions. An analysis will provide an indication of where the greatest quantity of power is being wasted and which of the efficiency measures described in this section should be utilized.

B. REDUCTION OF PLATE AND SCREEN DISSIPATION OF VACUUM TUBES

The problem of heat removal in electronic equipment can be reduced somewhat if one is careful to dissipate the minimum possible wattage in the system at the outset. Since we are concerned with reliable performance, this reduction must only be made if the performance will not be impaired. This means that the heaters will be operated at rated voltages and all of the power savings will be made in plate and screen dissipations.

There are many applications which do not require appreciable power output but where voltage amplification is desired and where the signal level amplitudes to be handled in the plate and grid circuits are quite low. It is in such applications where savings in tube dissipation can be made without affecting reliability, except to improve it. Examination of most electronic equipment will disclose the 'sockets' which meet these requirements and to which these reduced plate and screen voltages may be applied.

The tubes whose characteristics are shown in Tables XXII and XXIII are from the latest "Armed Services Preferred List of Electron Tubes". The types shown in the list include only those which carry a published rating at some other voltage conditions in addition to the 100 volt ratings. Thus, these characteristics are already set by the tube manufacturer and are held in control. In addition to the types shown, there are several on the same tube list which carry only 100-volt ratings, thus making it possible to include almost every function except power stages with 100-volt supplies.

Table XXII indicates several single and double triodes which carry either a 250 or 150-volt rating, in addition to the 100-volt rating.

TABLE XXII.

WATTAGE SAVINGS WITH REDUCED PLATE VOLTAGE OPERATION OF TRIODES

	TUBE TYPE	TOTAL WATTS - 250 VOLTS E_b	TOTAL WATTS - 150 VOLTS E_b	TOTAL WATTS - 100 VOLTS E_b	WATTS REDUCTION	PERCENT REDUCTION	G_m DECREASE MICROMHOS	PERCENT G_m DECREASE	G_m INCREASE MICROMHOS	PERCENT G_m INCREASE
SINGLE TRIODES	6C4W	3.58	----	2.14	1.44	40.0	----	----	900	41.0
	6J4	----	4.77	3.52	1.25	26.0	1000	8.4	---	----
	5718	----	2.90	1.80	1.10	38.0	700	10.8	---	----
	5719	----	1.23	1.02	.21	17.0	600	26.0	---	----
DOUBLE TRIODES	12AT7	6.89	----	2.63	4.26	62.0	1500	27.3	---	----
	5751	2.80	----	2.30	.50	17.1	350	21.9	---	----
	5814	7.46	----	4.56	2.90	39.0	900	41.0	---	----

TABLE XXIII.

WATTAGE SAVINGS WITH REDUCED PLATE AND SCREEN VOLTAGE

	TUBE TYPE	TOTAL WATTS E_b = 250V E_{sg} = 100V	TOTAL WATTS E_b = 250V E_{sg} = 150V	TOTAL WATTS E_b = 100V E_{sg} = 100V	WATTS REDUCTION	PERCENT REDUCTION	G_m DECREASE MICROMHOS	PERCENT G_m DECREASE	G_m INCREASE MICROMHOS	PERCENT G_m INCREASE
PENTODES	6SK7W	----	4.45	3.59	.86	19.3	----	----	350	17.6
	5749	----	5.06	3.41	1.65	32.6	100	2.25	---	----
	6AU6	5.19	----	2.60	2.59	50.0	1300	28.9	---	----

The change in mutual conductance (G_m) when the plate voltage is reduced to 100 volts is also shown, so that a quick calculation will indicate whether the proposed substitution may be made without increasing the number of tubes in the equipment to obtain the desired performance. It is important to realize that this change should not be made if the signal to be applied to the grid of the tube is larger than the grid bias minus approximately 0.6 volts.

Examination of this table will show, as expected, that the savings are least with high mu (amplification factor) tubes. This suggests another possible savings in heat by employing a high mu tube when the input signal conditions warrant.

Table XXIII indicates possible savings for three pentodes when operated with a plate and screen voltage of 100 volts instead of the higher ratings which are also published for these tubes. These tubes include remote and sharp cutoff pentodes, so most any circuit function involving pentodes can be achieved where small signals are involved.

The data supplied here indicate heat reduction possibilities for small signal operation which would contribute to the over-all reliability program by reducing the heat developed in the equipment.

The foregoing data are not complete. Many other tubes can be rerated in similar fashion provided their characteristics at lower voltages are defined by the appropriate military specifications and provided such characteristics are controlled by the manufacturer. Consequently, the above information is not reliable for tubes other than those mentioned. However, it is an indication of what can be accomplished in this direction. It is strongly recommended that most tubes be rerated for lower voltage applications.

C. REDUCTION OF HEATER VOLTAGE

Several authorities have recommended that tube life will be increased and that the dissipated power will be reduced, if tubes are operated at reduced heater or filament voltage during standby. This practice is not generally recommended because:

1. Experience to date has not indicated any significant improvement in reliability from standby operation at reduced heater voltage. Rather, reliability has been impaired on some occasions due to the misapplication of plate and screen voltages when the heater voltages are low. This has led to cathode poisoning by residual gasses and decomposition products.

2. Current surges due to the sudden application of the higher heater voltage will increase the possibility of heater failure.

3. The added circuit complexity necessary to change the heater voltage will increase equipment manufacturing and maintenance costs. Further, added parts will statistically reduce the probable reliability of any device.

The reduction of heater or filament voltage during normal operation is not recommended unless so specifically directed by the tube manufacturer.

D. DESIGN CONSIDERATIONS

It is recommended that the use of high-performance tubes be avoided wherever possible. Tubes with high transconductances (G_m) are inherently less reliable than other types. The internal element spacings and dimensional tolerances are very critical. Relative movement of the elements due to thermal expansion or vibration results in degraded performance to a much greater extent than in lower-performance tube types. Further, due to the small grid to cathode spacings (which produce higher grid temperatures) high-performance tubes have a reduced life expectancy compared to other types, especially at elevated temperatures. The control grids of high-performance tubes are so close to the cathodes that emitter migration and contact potential difficulties are enhanced. It has been predicted (Ref. 60) that the inherent reliability of a tube will be reduced six-fold if the performance is doubled.

APPENDIX A

SYMBOLS AND NOMENCLATURE

Symbol	Definition	Typical Unit
A	Area, surface or cross-sectional	$in.^2$
a	Convection modulus = $\frac{g \beta \rho^2 c_p}{\mu k}$	$\frac{1}{ft.^3 - {}^\circ F}$
B	Boltzman constant = 1.380×10^{-16} $\frac{ergs}{{}^\circ C}$	
C	Thermal conductance	$\frac{watts}{{}^\circ C}$
c_p	Specific heat at constant pressure	$\frac{watt\text{-}min.}{lb.\text{-}{}^\circ C}$
D,d	Diameter	in.
D_e	Equivalent diameter	in.
E	Electromotive force	volts
e	Electronic charge = 1.602×10^{-20} emu (abcoulombs)	
F	Clamping force	lbs.
F_a	Configuration factor (radiation)	
F_e	Emissivity factor (radiation)	
G	Electrical conductance	mhos.
Gr	Grashof number = $\frac{g \beta \Delta T L^3 \rho^2}{\mu^2}$	
g	Acceleration due to gravity = 32.2 $\frac{ft.}{sec.^2}$	
h	Coefficient of heat transfer	$\frac{watts}{in.^2\text{-}{}^\circ C}$
h_c	Coefficient of free convection	$\frac{watts}{in.^2\text{-}{}^\circ C}$
h_r	Coefficient of radiation	$\frac{watts}{in.^2\text{-}{}^\circ C}$
I	Electrical current	amps.
k	Thermal conductivity	$\frac{watts\text{-}in.}{in.^2\text{-}{}^\circ C}$
L	Length (characteristic length)	in.

APPENDIX A (Contd.)

SYMBOLS AND NOMENCLATURE

Symbol	Definition	Typical Unit
N_u	Nusselt number = $\frac{h_c L}{k}$	
P	Power dissipated	watts
P_r	Prandtl number = $\frac{c_p \mu}{k}$	
q	Rate of heat transfer	watts
q_c	Rate of convective heat transfer	watts
q_r	Rate of radiation heat transfer	watts
q_T	Total heat transfer rate	watts
R'	Electrical resistance	ohms.
R	Thermal resistance	$\frac{°C}{watt}$
r	Thermal resistivity	$\frac{°C\text{-in.}^2}{\text{watt-in.}}$
r'	Thermal contact resistivity	$\frac{°C\text{-in.}^2}{\text{watt}}$
S	Circuit stability factor	
T	Absolute temperature	°K.
t	Temperature	°C.
$\Delta t, \Delta T$	Temperature difference	°C
W	Weight	lbs., gms
x,y	Distance, (length)	in.
β	Volumetric expansion coefficient	$\frac{1}{°C}$
ϵ	Emissivity (of a surface)	
σ	Electrical conductivity	$\frac{1}{\text{ohm-cm.}}$
σ_s	Stefan-Boltzman constant = $0.0037 \times 10^{-8} \frac{\text{watt}}{\text{in.}^2 (°K)^4}$	
ρ	Density	$\frac{\text{lbs.}}{\text{ft.}^3}$
ρ_e	Electrical resistivity	ohm-in. or ohm-cm.
μ	Absolute or dynamic viscosity	$\frac{\text{lb.}}{\text{ft.-hr.}}$

APPENDIX B

LIST OF CONVERSION FACTORS

k - thermal conductivity

English system		cgs. system		hybrid system
$1.0 \frac{Btu.-ft.}{hr.-ft.^2-^oF}$	=	$0.004134 \frac{cal.-cm.}{sec.-cm.^2-^oC}$	=	$0.044 \frac{watt-in.}{in.^2-^oC}$

C_p - specific heat

$1.0 \frac{Btu.}{lb.-^oF}$	=	$1.012 \frac{cal.}{gm-^oC}$	=	$31.6 \frac{watt-min.}{lb.-^oC}$

μ - viscosity

		1.0 centipoises	=	$2.42 \frac{lb.}{hr.-ft.}$

$1.0 \frac{Btu.}{hr.}$ = 0.293 watts

$1.0 \frac{Btu.}{min.}$ = 17.58 watts

$1.0 \frac{calories}{hr.}$ = 0.001162 watts

$1.0 \frac{ergs}{hr.}$ = 2.78×10^{-11} watts

$1.0 \frac{joules}{hr.}$ = 2.78×10^{-4} watts

APPENDIX C

LIST OF ASSOCIATED CORNELL AERONAUTICAL LABORATORY REPORTS

Description	Report Number	Date of Issue	BuShips Contract Number
Survey Report of the State of the Art of Heat Transfer in Miniaturized Electronic Equipment	HF-710-D-10 NAVSHIPS 900, 189	3 March 1952	NObsr-49228
Manual of Standard Temperature Measuring Techniques, Units, and Terminology for Miniaturized Electronic Equipment	HF-845-D-2 NAVSHIPS 900, 187	1 June 1953	NObsr-63043
A Guide Manual of Cooling Methods for Electronic Equipment	HF-710-D-16 NAVSHIPS 900, 190	April 1954	NObsr-49228
Design Manual of Methods of Liquid Cooling Electronic Equipment	HF-845-D-9	Scheduled for July, 1957	NObsr-63043
Design Manual of Methods of Forced Air Cooling Electronic Equipment	HF-845-D-19	Scheduled for March, 1957	NObsr-63043
The Thermatron	HF-1053-D-3	Scheduled for January, 1957	NObsr-72531

APPENDIX D

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APPENDIX E

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Mr. Richard Booth, Associate Research Electronics Engineer, correlated the vacuum tube data.

Mr. Richard Wrobel, Junior Electronics Engineer, edited, proofread, and dimensionally analyzed this Manual.

Mr. William Brown, Cornell University Vacuum Tube Laboratory technician, constructed over one-hundred instrumented tubes and Thermatrons.

Mr. Oren Landis, Electronics Technician, instrumented and performed the experiments mentioned herein.

APPENDIX F

TABLE XXIV.

PROPERTIES OF AIR*

Temperature		** c_p Specific Heat, Btu / (lb.) (°F)	*** ρ Density, lb. / cu.ft.	μ Viscosity, lb. / (ft.)(hr.)	k, Thermal Conductivity, Btu / (hr.)(ft.)(°F)	$\frac{c_p \mu}{k}$ Prandtl Number	β Coefficient of Thermal Expansion $\frac{1}{°R}$	*** a x 10^{-6} Free Convection Modulus, $\frac{1}{(cu.ft.)(°F)}$
°F.	°C.							
-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

* Table derived mainly from Ref. 30

** Specific heat at constant pressure

*** Density and convection modulus for atmospheric pressure (29.92 in. Hg)

TABLE XXV
PHYSICAL PROPERTIES OF THE USEFUL METALS

SUBSTANCE	SPECIFIC GRAVITY	SPECIFIC HEAT	MELTING POINT, DEG. F	BOILING POINT, DEG. F	CUBICAL EXPANSION BY HEAT FROM 32 F TO 212 F	HEAT CONDUCTIVITY, SILVER=100	ELECTRICAL CONDUCTIVITY SILVER = 100	TENSILE STRENGTH, LB. PER SQ. IN.
Aluminum	2.67	....	1217	3272	0.0070	48	53	18,000
Antimony	6.76	.050	1166	2624	.027 – .050	4.2	3.5	1,000
Bismuth	9.82	.0301	520	2300	.0040	1.8	1.13	6,400
Brass	7.8 – 8.6	.092	1650±	....	.0057 – .0064	15 – 30	23 – 17	9,000 – 40,000
Bronze	8.52 – 8.96	.086	1650±	....	.0051 – .0057	....	....	3,500 – 25,000
Cadmium	8.65	.0567	609.6	1430	.0094	....	....	......
Cobalt	8.55	.107	2696	....	.0037	....	19.9	34,400
Copper	8.85	.093	1981.4	5050	.0051	89	99.5	30,000
German silver	8.5	.095	1850±	....	.0055	8	10 – 32	......
Gold	19.258	.032	1945.5	3992	.0044	53.2	76.7	14,000
Iridium	22.38	.032	4280±	....	.0020	34	30	......
Iron	7.9	.113	2786	4442	.0036	11 – 18	9.9 – 17	39,500±
Iron, cast	7.22	.1298	1900 – 2200	....	.0033	11.9	2.8 – 1.4	......
Iron, wrought	7.70	.1138	2700 – 2900	....	.0035	....	17	50,000±
Lead	11.38	.031	621.3	2900 – 3600	.0088	8.2	7.6	1,600 – 2,400
Magnesium	1.75	.025	1204	2048	.0083	37.6	35.8	20,000
Manganese	8.0	....	2246	3452	.....	....	....	......
Mercury	13.58	.033	-37.97	680	.0182	1.8	1.7	......
Nickel	8.8	.109	2642	....	.0038	14	14.5 – 9.9	50,000 – 100,000
Osmium	22.5	.031	4890±	....	.0020	....	16	......
Palladium	12.0	.058	2822	....	.0036	17	15	50,000
Platinum	21.5	.032	3191	....	.0027	17	20 – 10	30,000 – 50,000
Rhodium	12.4	.058	3542	....	.0026	30	23	......
Silver	10.51	.057	1760.9	3550	.0058	100	100	36,000
Steel	7.9	.117	2550±	....	.0041 – .0030	6 – 14	16 – 3	50,000 – 20,000
Tantalum	14.1 – 16.1	.036	5250	....	.0024	....	9.9	......
Tin	7.35	.056	449.4	3800	.0069	15.2	11.3	5,00
Titanium	3.54	....	3260±	....	.....	....	13.7	......
Tungsten	18.8	.033	6152	....	.....	....	23	500,000
Zinc	7.14	.096	786.9	1724	.0088	28.1	26	9,000 – 24,000

APPENDIX F (Contd.)

TABLE XXVI.

HEAT CONDUCTION DATA FOR VARIOUS MATERIALS AT APPROXIMATELY 65°C.

Material	ρ Density lb./cu.in.	Thermal Conductivity - k $\frac{\text{watts - in.}}{\text{in.}^2 - {}^\circ\text{C}}$	k $\frac{\text{Btu-ft.}}{\text{m.-ft.}^2 - {}^\circ\text{F}}$	Thermal Resistivity $\frac{{}^\circ\text{C - in.}^2}{\text{watt-in.}}$
Silver	0.380	10.6	241	0.094
Copper	0.322	9.7	220	0.103
Gold	0.696	7.5	171	0.133
Aluminum, Pure	0.098	5.5	125	0.182
Aluminum, 63S	0.100	5.1	116	0.196
Magnesium	0.063	4.0	91	0.250
High-beryllia ceramics	0.109 to 0.136	1.7 to 3.9	38.7 to 88.7	0.590 to 0.256
Red Brass	0.316	2.8	63.7	0.356
Yellow Brass	0.310	2.4	54.6	0.416
Chrome Copper*		8.25	187.0	0.121
"Berylco" #10 Alloy		6.25	142.0	0.160
Beryllium Copper	0.297	2.1	47.8	0.475
Pure Iron	0.284	1.9	43.2	0.526
Phosphor Bronze	0.318	1.3	29.6	0.770
Soft Steel	0.284	1.18	26.8	0.847
Monel	0.318	0.9	20.5	1.110
Lead	0.409	0.83	18.9	1.120
Hard Steel	0.284	0.65	14.8	1.540
Steatite	0.094	0.06	1.36	15.0
Pyrex	0.094	0.032	0.728	31.2
Grade A Lava	0.085	0.03	0.683	33.3
Soft Glass	0.094	0.025	0.569	40.0
Water	0.0361	0.0167	0.380	60.0
Mica	0.101	0.015	0.341	66.6
Paper-base Phenolic	0.0497	0.007	0.159	143.0
Plexiglas	0.043	0.0047	0.107	213.0
P-43 Casting Resin	0.045	0.0046	0.105	217.0
Maple	0.025	0.0042	0.096	238.0
Pine	0.018	0.003	0.067	333.0
Polystyrene	0.038	0.0027	0.061	370.0
Glass Wool	0.001	0.001	0.023	1000.0
Air	0.000043	0.0007	0.016	1430.0
Firebrick	0.011	0.0022	0.050	454.0

(Calculated from Ref. 21)

* From American Brass Co.

TABLE XXVII

TEMPERATURE CONVERSIONS

The middle column of figures contains the reading (F or C) to be converted. If converting from degrees Fahrenheit to degrees Centigrade, read the Centigrade equivalent in the column headed "C". If converting from degrees Centigrade to degrees Fahrenheit, read the Fahrenheit equivalent in the column headed "F".

C		F	C		F	C		F
-57.0	-70	-94.0	-1.1	30	86.0	46.1	115	239.0
-51.0	-60	-76.0	-0.6	31	87.8	48.9	120	248.0
-46.0	-50	-58.0	0.0	32	89.6	51.7	125	257.0
-40.0	-40	-40.0	0.6	33	91.4	54.4	130	266.0
-39.4	-39	-38.2	1.1	34	93.2	57.2	135	275.0
-38.9	-38	-36.4	1.7	35	95.0	60.0	140	284.0
-38.3	-37	-34.6	2.2	36	96.8	62.8	145	293.0
-37.8	-36	-32.8	2.8	37	98.6	65.6	150	302.0
-37.2	-35	-31.0	3.3	38	100.4	68.3	155	311.0
-36.7	-34	-29.2	3.9	39	102.2	71.1	160	320.0
-36.1	-33	-27.4	4.4	40	104.0	73.9	165	329.0
-35.6	-32	-25.6	5.0	41	105.8	76.7	170	338.0
-35.0	-31	-23.8	5.6	42	107.6	79.4	175	347.0
-34.4	-30	-22.0	6.1	43	109.4	82.2	180	356.0
-33.8	-29	-20.2	6.7	44	111.2	85.0	185	365.0
-33.3	-28	-18.4	7.2	45	113.0	87.8	190	374.0
-32.8	-27	-16.6	7.8	46	114.8	90.6	195	383.0
-32.2	-26	-14.8	8.3	47	116.6	93.3	200	392.0
-31.7	-25	-13.0	8.9	48	118.4	96.1	205	401.0
-31.1	-24	-11.2	9.4	49	120.2	98.9	210	410.0
-30.6	-23	-9.4	10.0	50	122.0	101.7	215	419.0
-30.0	-22	-7.6	10.6	51	123.8	104.4	220	428.0
-29.4	-21	-5.8	11.1	52	125.6	107.2	225	437.0
-28.9	-20	-4.0	11.7	53	127.4	110.0	230	446.0
-28.3	-19	-2.2	12.2	54	129.2	112.8	235	455.0
-27.8	-18	-0.4	12.8	55	131.0	115.6	240	464.0
-27.2	-17	+1.4	13.3	56	132.8	118.3	245	473.0
-26.7	-16	3.2	13.9	57	134.6	121.1	250	482.0
-26.1	-15	5.0	14.4	58	136.4	123.9	255	491.0
-25.6	-14	6.8	15.0	59	138.2	126.7	260	500.0
-25.0	-13	8.6	15.6	60	140.0	129.4	265	509.0
-24.4	-12	10.4	16.1	61	141.8	132.2	270	518.0
-23.9	-11	12.2	16.7	62	143.6	135.0	275	527.0
-23.3	-10	14.0	17.2	63	145.4	137.8	280	536.0
-22.8	-9	15.8	17.8	64	147.2	140.6	285	545.0
-22.2	-8	17.6	18.3	65	149.0	143.3	290	554.0
-21.7	-7	19.4	18.9	66	150.8	146.1	295	563.0
-21.1	-6	21.2	19.4	67	152.6	148.9	300	572.0
-20.6	-5	23.0	20.0	68	154.4	154.4	310	590.0
-20.0	-4	24.8	20.6	69	156.2	160.0	320	608.0
-19.4	-3	26.6	21.1	70	158.0	165.6	330	626.0
-18.9	-2	28.4	21.7	71	159.8	171.1	340	644.0
-18.3	-1	30.2	22.2	72	161.6	176.7	350	662.0
-17.8	0	32.0	22.8	73	163.4	182.2	360	680.0
-17.2	1	33.8	23.3	74	165.2	188.2	370	698.0
-16.7	2	35.6	23.9	75	167.0	193.3	380	716.0
-16.1	3	37.4	24.4	76	168.8	198.9	390	734.0
-15.6	4	39.2	25.0	77	170.6	204.4	400	752.0
-15.0	5	41.0	25.6	78	172.4	210.0	410	770.0
-14.4	6	42.8	26.1	79	174.2	215.6	420	788.0
-13.9	7	44.6	26.7	80	176.0	221.1	430	806.0
-13.3	8	46.4	27.2	81	177.8	226.7	440	824.0
-12.8	9	48.2	27.8	82	179.6	232.2	450	842.0
-12.2	10	50.0	28.3	83	181.4	237.8	460	860.0
-11.7	11	51.8	28.9	84	183.2	243.3	470	878.0
-11.1	12	53.6	29.4	85	185.0	248.9	480	896.0
-10.6	13	55.4	30.0	86	186.8	254.4	490	914.0
-10.0	14	57.2	30.6	87	188.6	260.0	500	932.0
-9.4	15	59.0	31.1	88	190.4	265.0	510	950.0
-8.9	16	60.8	31.7	89	192.2	271.0	520	968.0
-8.3	17	62.6	32.2	90	194.0	276.0	530	986.0
-7.8	18	64.4	32.8	91	195.8	282.0	540	1004.0
-7.2	19	66.2	33.3	92	197.6	288.0	550	1022.0
-6.7	20	68.0	33.9	93	199.4	293.0	560	1040.0
-6.1	21	69.8	34.4	94	201.2	299.0	570	1058.0
-5.6	22	71.6	35.0	95	203.0	304.0	580	1076.0
-5.0	23	73.4	35.6	96	204.8	310.0	590	1094.0
-4.4	24	75.2	36.1	97	206.6	315.0	600	1112.0
-3.9	25	77.0	36.7	98	208.4	321.0	610	1130.0
-3.3	26	78.8	37.2	99	210.2	326.0	620	1148.0
-2.8	27	80.6	37.8	100	212.0	332.0	630	1166.0
-2.2	28	82.4	40.6	105	221.0	338.0	640	1184.0
-1.7	29	84.2	43.3	110	230.0	343.0	650	1202.0

APPENDIX F (Contd.)

TABLE XXVIII.

PROPERTIES OF AIR, HELIUM, HYDROGEN, AND NITROGEN

AT 100°C.

	c_p Btu / (lb.)(°F)	ρ lb. / cu.ft.	μ lb. / (ft.)(hr.)	k Btu / (hr.)(ft.)(°F)	$\frac{c_p \mu}{k}$	a 1 / (cu.ft.)(°F)
Air	0.241	0.0591	0.0527	0.0184	0.689	539,000
Helium	1.25	0.00816	0.0544	0.097	0.700	9,810
Hydrogen	3.43	0.00411	0.0254	0.129	0.676	11,000
Nitrogen	0.25	0.0571	0.0507	0.0180	0.704	566,000

From: Several sources.

APPENDIX F (Contd.)

TABLE XXIX.

TOTAL EMISSIVITY VALUES FOR VARIOUS METALS & GLASSES*

Material	Condition	At		
		100°C.	320°C.	500°C.
Alleghany Metal	No. 4 Polish	.13		
Alleghany Alloy No. 66	Polished	.11		
Aluminum	Commercial Sheet	.09		
Aluminum	Polished	.095		
Aluminum	Rough Polish	.18		
Brass	Polished	.059		
Carbon	Rough Plate	.77	.77	.72
Carbon, Graphitized	Rough Plate	.76	.75	.71
Chromium	Polished	.075		
Copper	Polished	.052 to .04		
Copper - Nickel	Polished	.059		
Iron	Dark Gray Surface	.31		
Iron	Roughly Polished	.27		
Lampblack	Rough Deposit	.84		.78
Molybdenum	Polished	.071		
Nickel	Polished	.072		
Nickel-Silver	Polished	.135		
Radiator Paint, White	Clean	.79		
Radiator Paint, Cream	Clean	.77		
Radiator Paint, Black	Clean	.84		
Radiator Paint, Bronze	Clean	.51		
Silver	Polished	.052 to .03		
Stainless Steel	Polished	.074		
Steel	Polished	.066		
Tin	Polished	.069		
Tin	Commercial Coat	.084		
Tungsten	Polished Coat	.066		
Zinc	Commercial Coat	.21		
Fuzed Quartz	1.96 mm. Thick	.775	.76	.67
Covex D (Glass)	3.40 mm. Thick	.83	.90	.91
Nonex (Glass)	1.57 mm. Thick	.835	.87	.82
Aluminum Paint		.29		

*From several sources.

APPENDIX F (Contd.)

TABLE XXX.

EMISSIVITY VALUES OF COMMON MATERIALS

Materials	*Emissivity at 100°C.
Lampblack	0.95
Dull-oxide-type Paints	0.94
Asbestos and Most Non-metallic Insulating Materials	0.93
Most Glass-type Paint and Enamels	0.88
Oxidized Steel	0.75
Oxidized Copper	0.70
Oxidized Brass	0.60
Aluminum Paint	0.27 to 0.67
Oxidized Nickel or Monel	0.42
Anodized Aluminum)) Oxidized Aluminum)	0.22 to 0.40 normally, but may vary from 0.05 to 0.75, depending on thickness of film.
Surface	
Oil Paint (Any Color)	.92 to .96
Enamel (Any Color)	.88 to .91
Varnish	.88 to .91
Black Lacquer	.80 to .95
Aluminum Paint	.27 to .67
Dull Sheet Steel	.80

From: Several sources.

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