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Cooling of Electronic Equipment

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Cooling of Electronic Equipment

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Preface

This book is written for electronic engineers, electronic technicians, and mechanical engineers who are concerned with the cooling of electronic equipment.

All electronic equipment needs cooling, whether it uses only a few low power transistors or many high power tubes. In most equipment, the cooling design is as important as the electronic design itself.

Unfortunately, no good textbooks are available on the cooling of electronic equipment. The many textbooks on heat transfer and cooling that are available are written for furnace, boiler, and air conditioning designers. These standard textbooks concentrate on the derivation of the basic hydrodynamic equations and on such practical problems as heat exchanger design or the effect of scale in boilers.

The formulas in the standard heat transfer and cooling textbooks are difficult for the electronic engineer to use, because they are expressed in the units of the heat exchanger or furnace designer. It is a formidable task for the electronic engineer to convert from the BTU per hour, °F, and square feet of these formulas to the watts, °C, and inches that are commonly used in designing electronic equipment. Even the tables in the standard textbooks on cooling are useless to the electronic engineer, who has no need to know the thermal conductivity of fire bricks or the thermal properties of carbon tetrachloride.

This book is written to overcome these problems. It is intended for electronic engineers and is written with a terminology and approach they can readily understand.

I am an electronic engineer and my main interest is in electronic design. At first, I tried to ignore the cooling problem, but when some of my finest designs "went up in smoke," I realized I had to face the cooling problem head-on.

My first step at trying to design cooling for electronic equipment was to get out my old college textbooks on hydrodynamics and heat transfer. It was easy enough to understand the theory of thermal radiation, forced convection of air and liquids, or evaporation cooling. Certainly any electronic engineer

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can understand the concept of Reynolds number, laminar flow, turbulent flow, etc. The problem arises in converting the basic hydrodynamic formulas into a useful form and into an "electronic engineering" system of units.

I soon found that I was not alone among electronic engineers who suffered from a lack of literature on how to cool electronic equipment. This book has grown out of my need to have simple formulas for designing the cooling of my own equipment. Soon I was distributing my notes to fellow engineers, so that they could design the cooling for their equipment. Then I was distributing these notes to our customers, so they could use our equipment without burning it up. I have presented the material of this book in university extension courses and in special industrial courses to electronic engineers who were, out of necessity, also fighting the cooling problem.

This book presents the cooling formulas which can be found in any standard textbook on hydrodynamics or heat transfer in simple form. The formulas are all expressed in terms of power, temperature, dimensions, and the thermal properties of materials. The traditional normalization in terms of Reynolds number, Prandtl number, or the other dimensionless constants of heat transfer theory has been eliminated. An electronic engineer need not learn heat transfer theory, although he probably already knows it, to be able to apply the simple formulas of this book.

Perhaps most importantly, all the formulas in this book are expressed in units of the electronic engineer. Power is in watts, all dimensions are in inches, and temperature is in °C. Thermal properties of all materials commonly used in electronic equipment are tabulated in the same consistent set of units. Air flow and liquid flow are in units which are easily measured in electronic equipment. Finally, a complete appendix is provided with conversion tables to convert from the other systems of units used for heat transfer calculations to the "electronic engineering" systems of units of this book.

A consistent set of symbols has been used throughout the book. All symbols are defined as they are introduced in each chapter, and a summary of symbols is presented in the appendix. Each symbol always represents the same quantity throughout the book, and is always in the same unit.

All the important methods of cooling used in electronic equipment are covered in this book, including:

- Conduction.
- Radiation and natural convection.
- Forced air cooling.
- Forced liquid cooling.
- Liquid evaporation.
- Heat pipes.
- Refrigeration and cryogenic cooling.

Chapter 1 discusses the problem of cooling electronic equipment. The

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various cooling methods listed above are compared and the design problems that each cooling method presents are described.

Chapters 2 through 6 discuss the basic methods of cooling: conduction, radiation and natural convection, forced air cooling, forced liquid cooling, and liquid evaporation, respectively. Each chapter begins with a discussion of a typical electronic equipment cooled by the method of that chapter. Then each chapter presents the necessary design formulas and the properties of materials commonly used in electronic equipment that are necessary for making calculations with this type of cooling. Where auxiliary equipment is required, such as fans or heat exchangers, available equipment and its capabilities are discussed. At the end of each chapter, sample calculations are given. The first of these sample calculations is usually for the actual electronic equipment used as the illustrative example at the beginning of the chapter. Finally, a list of references is presented in each chapter for further study.

Chapter 7 discusses heat pipes. Chapter 8 describes refrigerated electronic equipment, including cryogenic coolers and thermoelectric cooling modules. Chapter 9 discusses transient cooling effects, such as occur in equipment warmup.

The design information presented in this book gives only approximate answers. Therefore, after the cooling has been designed and the electronic equipment built, the thermal performance of the equipment must be measured. Techniques for making the necessary thermal measurements on electronic equipment are presented in Chapter 10.

It would be nice if we electronic engineers could simply avoid the cooling problem and get on with our electronic design. Unfortunately, the cooling problem won't go away. This book permits the cooling problem to be faced and solved simply and quickly.

I hope it will be of help to you.

Allan W. Scott

Palo Alto, California

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1

Understanding the Cooling Problem

All electronic equipment needs cooling, whether it uses only a few low power transistors or many high power tubes. In most equipment, the cooling design is as important as the electronic design itself. A typical electronic cooling design is shown in Figure 1.1. A 10 watt transistor is mounted on a finned aluminum heat sink. The transistor is 50% efficient, so 20 W of electrical

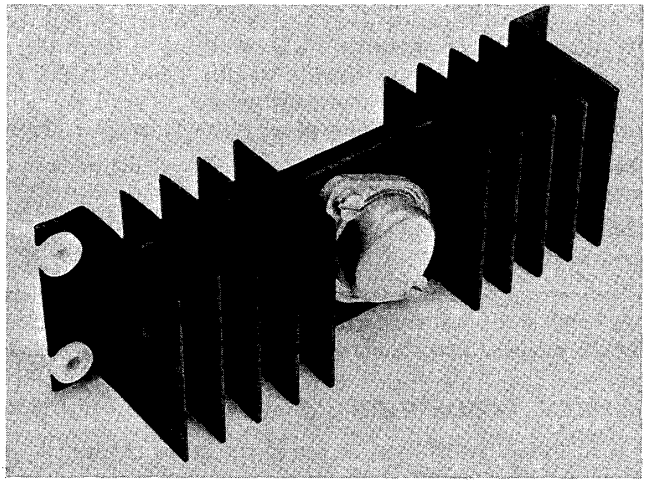

FIGURE 1.1
Transistor mounted on an aluminum heat sink (Photo courtesy of Wakefield Engineering, Inc.)

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input power are converted to 10 W of useful electrical output power and 10 W of heat. The 10 W of heat must be conducted from the transistor to the cooling fins, where it is then transferred to the surrounding environment.

Heat always flows from hotter objects to cooler ones. Therefore, the transistor must be hotter than the cooling fins, and the cooling fins must be hotter than the surroundings. In any electronic equipment, the temperature of each component will rise until it is hot enough to transfer its internally generated heat to the surroundings. If the cooling design is not adequate, the component will get so hot, in an effort to transfer this heat, that it will destroy itself.

1.1 THE BASIC COOLING PROBLEM

The purpose of cooling electronic equipment is to keep the temperature of each component below its safe operating value.

The electric component temperature is determined from:

$$T\,(\text{component}) = \Delta T\,(\text{conduction}) + \Delta T\,(\text{transfer}) + T\,(\text{surroundings}) \quad (1.1)$$

where:

T (component)	is the temperature of the electronic component.
ΔT (conduction)	is the temperature difference required to conduct the heat from the electronic component to the cooling fins.
ΔT (transfer)	is the temperature rise of the fins above the surroundings required to transfer the heat from the fins to the surrounding environment.
T (surroundings)	is the temperature of the surrounding environment.

As Equation 1.1 shows, the cooling of electronic equipment consists of two parts:

1. Conduction of heat from the electronic component to the cooling fins.
2. Transfer of heat from the cooling fins to the environment.

This book provides the necessary design information so that the temperature rises required for heat conduction and heat transfer can be calculated, so that the component temperature can be kept within acceptable limits.

1.2 HEAT CONDUCTION

The following factors must be considered in the conduction of heat from the electronic component to the cooling fins:

1. The materials used for conducting heat.
2. The geometry of the heat flow paths.

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3. Thermal interfaces where parts are joined together.
4. Conduction of heat through the cooling fins themselves.

Complete design information which considers all the above factors, and which permits a calculation of the conduction temperature rise, ΔT (conduction) of Equation 1.1, is presented in Chapter 2.

1.3 CHOICE OF HEAT TRANSFER METHOD

Once the heat has been conducted from the electronic component to the cooling fins, it must then be transferred to the surrounding environment by one of the following means:

1. Radiation and natural convection.
2. Forced air cooling.
3. Forced liquid cooling.
4. Liquid evaporation.

The above list of heat transfer methods is arranged in order of increasing heat transfer effectiveness. For a given fin area, the least heat can be transferred by radiation and natural convection, more can be transferred by forced air cooling, even more can be transferred by forced liquid cooling, and the most can be transferred by liquid evaporation.

The list is also arranged in order of increasing cooling system complexity. Heat transfer by radiation and natural convection requires no auxiliary equipment—just the cooling fins themselves—and is the simplest design. Forced air cooling requires a fan and fan controls and is more complicated. Forced liquid cooling requires a pump, coolant reservoir, cooling fluid, etc., and is even more complicated.

The effectiveness of each of the above methods of heat transfer is compared in Figure 1.2, which shows the heat that can be transferred per square inch of surface, when the fin surface is at 100°C and the surroundings are at 20°C. For any one heat transfer method the amount of power that can be transferred varies over a wide range, depending on the details of the cooling design.

Figure 1.2 also illustrates that for most applications more than one heat transfer method could be used. The best choice depends on a tradeoff between system simplicity, heat sink complexity, and fin area.

The problems of the design of heat transfer by each of the four methods—radiation and natural convection, forced air cooling, forced liquid cooling, and liquid evaporation—are summarized in the next four sections of this chapter. Complete design information on each heat transfer method is presented in Chapters 3 through 6 respectively. Each of these chapters begins

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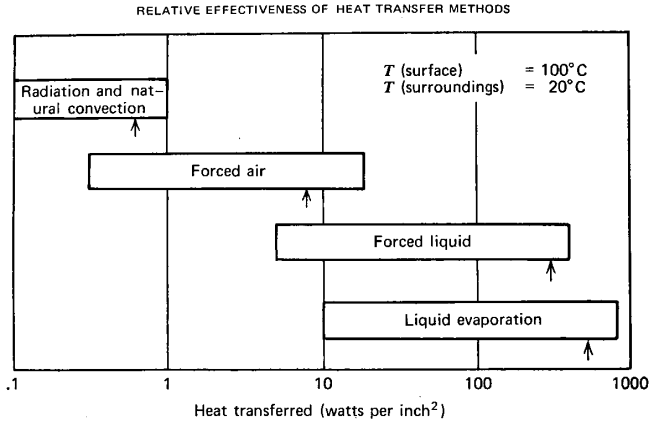

FIGURE 1.2
Relative effectiveness of heat transfer methods

with a typical example of heat transfer in electronic equipment, and the amount of heat transferred per square inch of surface in these examples is shown by the arrows in Figure 1.2.

1.4 RADIATION AND NATURAL CONVECTION

Radiation and natural convection are the simplest to use of all the heat transfer methods. No auxiliary equipment is required, just the cooling fins themselves. The hot fins radiate heat directly to the cooler surroundings. At the same time, the air near the hot fins is heated and rises and is replaced by cooler air. This convective air current provides additional heat transfer.

A typical heat sink cooled by radiation and natural convection was shown in Figure 1.1.

In most electronic equipment, heat transfer occurs by radiation and natural convection simultaneously. However, the amount of heat transferred by each method depends on heat sink temperature, geometry, and orientation in different ways. For a particular electronic cooling design, the heat transferred by each method must be calculated separately and then combined.

The amount of heat that can be transferred by radiation depends on:

1. The temperature of the radiating surface.
2. The temperature of the surroundings.
3. Surface conditions of the fins.
4. Shielding effects of adjacent fins.

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The amount of heat that can be transferred by natural convection depends on the following factors:

1. Temperature difference between the surface and the surrounding air.
2. Dimensions of the surface.
3. Orientation of the surface.
4. Spacing between adjacent surfaces.
5. Altitude (which determines air density).

Complete design information on heat transfer by radiation and natural convection, which considers all the above factors, and which permits a calculation of the transfer temperature rise, ΔT (transfer) of Equation 1.1, is presented in Chapter 3.

1.5 FORCED AIR COOLING

As Figure 1.2 shows, the amount of heat that can be transferred from a given cooling fin area is increased by more than an order of magnitude by blowing air over the cooling fins, rather than relying on radiation and natural convection. Forced air cooling is more complicated to implement than cooling by radiation and natural convection, because a fan and the associated fan controls are required. Forced air cooling is, however, much simpler than forced liquid cooling, because a supply of cooling air is readily available and air does not have the freezing, boiling, or dripping problems of cooling liquids.

A typical electronic cooling design using forced air cooling is shown in Figure 1.3. The design of air cooling poses two problems:

1. Choice of the fan or blower.
2. Design of the cooling fin geometry.

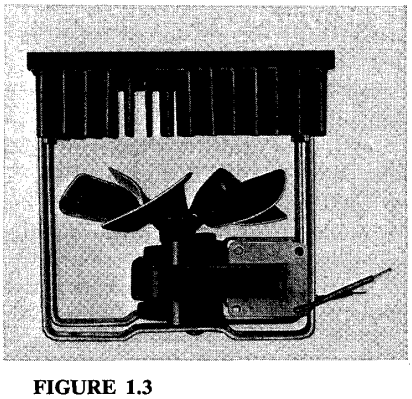

FIGURE 1.3
Forced air cooled heat sink (Photo courtesy of Wakefield Engineering)

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These two problems must be solved jointly. The amount of air flow that a particular fan can provide is determined by the pressure into which the fan must work. Both the amount of heat transfer that can be obtained from forced air cooling and the pressure required to force air through the cooling fins depends on air flow and fin geometry. Consequently, the choice of fan must be made in conjunction with the fin design.

Complete design information which permits the most suitable fan and the most optimum cooling fin design to be chosen is presented in Chapter 4. Formulas are presented which permit a calculation of the heat transfer temperature rise, ΔT (transfer) of Equation 1.1, for the case of forced air cooling. The effects of high altitude operation on forced air cooling are included.

1.6 FORCED LIQUID COOLING

Forced air cooling, although simpler to implement than forced liquid cooling, does have several disadvantages. Forced air cooling may not be suitable for electronic equipment which must be operated at high altitudes where air density is low. The acoustic noise of the fan may be objectionable and the vibration of the fan may adversely affect the performance of the electronic equipment. The hot air ejected from the electronic package may also be objectionable.

All of these disadvantages of forced air cooling are eliminated by the use of forced liquid cooling. The cooling pumps and coolant reservoir can be removed from the electronic package, so that quiet, vibration-free operation can be maintained. As shown in Figure 1.2, the use of forced liquid cooling provides an order of magnitude greater heat transfer per unit area than forced air cooling. For very high power electronic equipment, this greater heat transfer capability is a necessity, because the cooling fin area cannot be made great enough to transfer heat by forced air cooling. The use of liquid cooling also permits high density mounting of lower power electronic components. This high density mounting may not only be desirable from a packaging standpoint; it may be a necessity to achieve required electronic performance.

A typical forced liquid cooling design of an electronic component is shown in Figure 1.4. The particular component is the liquid cooled collector of a high power microwave transmitter tube. The coolant inlet and outlet fittings and the multiple liquid cooling ducts can be seen in the cutaway view.

The following factors must be considered in designing forced liquid cooling of electronic equipment:

1. The design of the cooling ducts.

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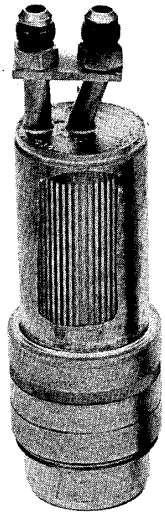

FIGURE 1.4
Forced liquid cooled collector of a high power microwave transmitter tube (Photo courtesy of Varian Associates)

2. The type of cooling liquid.
3. The effect of coolant inlet temperature.
4. The coolant pump, heat exchanger, and other auxiliary equipment.

Complete design information for forced liquid cooling, which considers all of the above factors, is given in Chapter 5. Formulas are presented which permit a calculation of the transfer temperature rise, ΔT (transfer) of Equation 1.1, for forced liquid cooling.

1.7 COOLING BY LIQUID EVAPORATION

Evaporation cooling can be effectively used in the following different ways in electronic equipment:

1. Cooling of high power components at high power densities.
2. Cooling of all components in an electronic equipment by immersing the entire assembly in a package filled with dielectric oil.
3. Maintaining a constant temperature bath for electronic components.
4. Simple expendable cooling systems.

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Figure 1.5 shows an electron tube cooled by evaporation. The anode of the tube, where heat is generated, is immersed in liquid. Heat is transferred from the tube by boiling the liquid. The vapor from the boiling process rises from

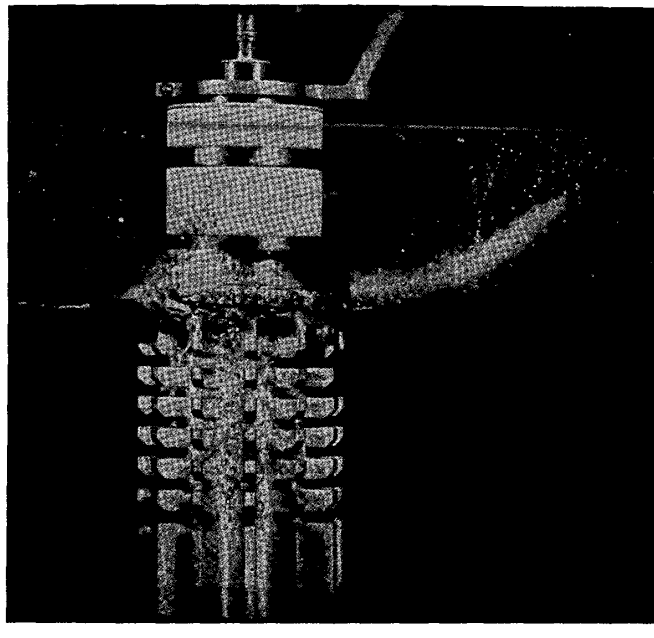

FIGURE 1.5
Cooling of a transmitter tube by liquid evaporation (Photo courtesy of ITT)

the liquid bath, is condensed in a condenser, and is then returned to the cooling bath. The condenser heat transfer area can be made large enough, because it is separated from the electronic equipment, so that heat can be transferred from it to the surroundings by radiation and convection or by forced air cooling.

No coolant pumps are needed for evaporation cooling. Consequently, it is an extremely simple cooling method. As shown in Figure 1.2, evaporation cooling provides greater heat transfer per unit surface area than any of the other methods.

The disadvantages of cooling by evaporation are:

1. The system can operate in one orientation only, with the liquid bath at the lowest point in the system.

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2. Immediate destruction of the electronic component occurs if the maximum heat transfer rate is exceeded, because the temperature of the component increases very rapidly above the critical heat transfer rate.

In designing cooling by evaporation, the following factors must be considered:

1. Choice of the cooling fluid.
2. Design of the component surface where evaporation occurs.
3. Design of the condenser.
4. Pressure and pressure equalization.
5. Equipment orientation.

Complete design information for heat transfer by liquid evaporation, including consideration of all the above factors, is presented in Chapter 6.

1.8 HEAT PIPES

The effective cooling design of electronic equipment requires minimizing both the temperature rise required for conducting heat from the electronic component to the cooling fins and the temperature rise required for transferring heat from the cooling fins to the surroundings. As shown in Equation 1.1, both factors contribute to the operating temperature of the electronic component.

In many designs sufficient space may be available for a large number of cooling fins, but the major problem is conducting the heat from the electronic component to the cooling fins. Even if the heat sink is made of copper, the temperature rise due to conduction may be excessive.

Heat pipes offer a solution to this problem. A schematic drawing of a heat pipe is shown in Figure 1.6. The heat pipe consists of a hollow tube which

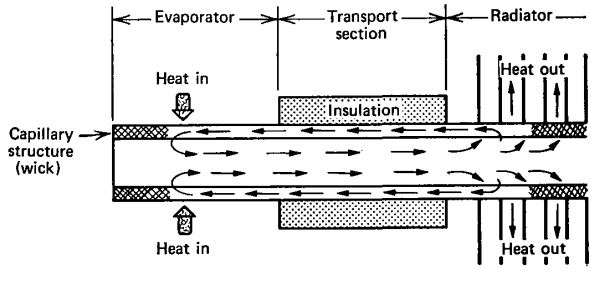

FIGURE 1.6
Schematic drawing of a heat pipe

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has been evacuated and then filled with a coolant liquid. The incident heat evaporates the liquid at one end of the pipe and the vapor transports the heat to the other cooler end of the pipe. At the cool end of the pipe the liquid condenses and transfers the heat. So far, this means of heat transfer is the same as evaporation cooling, discussed in the previous section. However, the heat pipe has an additional feature that permits it to be operated in any orientation. The inner surface of the heat pipe contains a capillary structure or "wick" which returns the condensed liquid to the hot evaporator end of the pipe by capillary action. The heat pipe can therefore even operate against gravity, that is, with its evaporator end upward.

For a given temperature rise, a heat pipe can conduct several orders of magnitude more heat than a solid copper rod of the same diameter. Consequently, heat pipes are finding increasing use for the conduction of heat in electronic equipment.

Complete information on the use of heat pipes in electronic equipment is presented in Chapter 7.

1.9 REFRIGERATED EQUIPMENT

The purpose of cooling electronic equipment is to keep the temperature of the electronic components at some desired temperature. With the methods of cooling discussed in all the previous sections, the electronic component is always hotter than the temperature of the surroundings. Heat flows from hot to cold bodies, so the electronic component had to be the hottest element in order to transfer its heat to the surroundings.

If necessary for achieving the desired electronic performance, the electronic component can be cooled to a temperature *below* the temperature of the surroundings by the use of refrigeration. In refrigerated equipment, heat does not flow from the electronic component, but is "pumped" by the refrigeration system from the cold component to the hot surroundings.

Examples of electronic equipment where refrigeration is required are:

- Infra-red detectors.
- Masers.
- Parametric amplifiers.
- Equipment which must work in surroundings which are at a higher temperature than the safe operating temperature of the electronic components.

Refrigeration is not a substitute for good conventional cooling design. If the electronic component can be hotter than the surroundings, then by careful design, the heat can always be transferred from the component by

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conduction, radiation and natural convection, forced air cooling, forced liquid cooling, or evaporation cooling, and refrigeration is not necessary.

Refrigeration systems for electronic equipment may be classified into the following types:

- Refrigerated cooling air or cooling liquid.
- Refrigerated heat sinks.
- Liquid nitrogen baths.
- Thermoelectric coolers.

These refrigeration systems are discussed in detail in Chapter 8.

1.10 TRANSIENT EFFECTS

Chapters 2 through 8 all consider steady state conditions. Under steady state conditions, heat is transferred to the surroundings as fast as it is generated by the electronic component, and the temperatures of all elements—the component itself, the heat sink, the cooling fins, and the cooling air or liquid —remain constant with time.

However, when electronic equipment is first turned on, or when the power input or the coolant flow rates are changed, the temperature of the electronic component and all other elements in the cooling system vary with time, until the equipment reaches its steady state temperature. Because many electronic components change their electrical characteristics when their temperature changes, the transient thermal characteristics of the equipment must be considered.

Design information on transient thermal conditions in electronic equipment is presented in Chapter 9.

1.11 THERMAL MEASUREMENT TECHNIQUES FOR ELECTRONIC EQUIPMENT

After the cooling of a particular electronic equipment has been designed, and the equipment has been built, the thermal performance of the equipment must be measured. All cooling design formulas give only approximate results, so experimental measurements are essential.

The thermal measurements that must be made on electronic equipment include:

1. Temperature of the electronic component, the heat sink, and the cooling fins.
2. Temperature distribution through the various parts.
3. Air flow rate.

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4. Air pressure.
5. Liquid flow rate.
6. Liquid pressure.

Techniques for making the necessary thermal measurements on electronic equipment are presented in Chapter 10.

1.12 SYMBOLS, UNITS, AND CONVERSION FACTORS

A consistent set of units has been used throughout this book. Each symbol is defined as it is introduced in the text, and each symbol stands for the same quantity throughout the book. The units in which each symbol is measured are the same throughout the book. A listing of each symbol and the units in which it is measured is also presented in Appendix A.

This book is intended for use by practicing engineers for the design of cooling of electronic equipment. For this reason the units of measurement used in all formulas have been chosen to be the easiest for electronic equipment designers to use. For example, power or heat is expressed in *watts*, rather than in British thermal units per hour (BTU/hr) or calories per second. This choice of watts as the unit of measurement is obvious, because this is the common measurement unit of power for electronic equipment. All dimensions are expressed in *inches*, rather than in feet or centimeters. Temperature is expressed in degrees Centigrade.

A variety of different systems of units are used in the listed references and other published data. To permit maximum use of other material, a conversion table from all other systems of units to the system of units of this book is presented in Appendix B.

1.13 REFERENCES

Appropriate references are given at the end of each chapter, and the useful information presented in each reference is briefly discussed.

General references on cooling of electronic equipment are as follows:

1. Moore, A. D., *Heat Transfer Notes for Electrical Engineering*, George Wahr Publishing Co., Ann Arbor, Michigan, 1963.

2. *Guide Manual of Cooling Methods for Electronic Equipment*, NAVSHIPS 900, 190, U.S. Government Printing Office, Washington, D.C., 1955.

3. Krauss, A. D., *Cooling Electronic Equipment*, Prentice Hall, Inc., Englewood Cliffs, New Jersey, 1965.

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4. Holman, J. P., *Heat Transfer*, McGraw-Hill, New York, 1968.

5. McAdams, W. H., *Heat Transmission*, McGraw-Hill, New York, 1954.

References 1 through 3 describe the cooling of electronic equipment. All contain the basic thermal and hydrodynamic equations from which the formulas, graphs, and tables of this book are derived.

References 4 and 5 are general reference books on heat transmission and cooling.

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2

Heat Conduction

The problem of cooling electronic equipment has two parts:

1. Conduction of heat from the electronic component to the cooling fin surface.
2. Transfer of heat from the cooling fins to the surrounding environment.

This chapter discusses the first part of the problem: the conduction of heat from the electronic component, through the heat sink, to the cooling fins.

Several factors must be considered:

1. The materials used in the heat sink.
2. The geometry of the heat sink.
3. The mounting interface between the component and the heat sink.
4. Conduction of heat through the cooling fins themselves.

A typical heat conduction design for electronic equipment is described in Section 2.1. The basic heat conduction formula is presented in Section 2.2. The materials of the heat sink and the geometry of the heat sink are discussed in Sections 2.3 and 2.4, respectively. Mounting interfaces are discussed in Section 2.5, and conduction of heat through the cooling fins is described in Section 2.6. Examples of conduction of heat in electronic equipment are given in Section 2.7 to illustrate the use of the equations, figures, and tables of this chapter.

2.1 A TYPICAL HEAT CONDUCTION DESIGN

A typical heat conduction design for electronic equipment—a transistor mounted on a heat sink—is shown in Figure 2.1. Figure 2.2 shows a cross section drawing of the various parts which make up the heat sink.

Heat is generated in the transistor because, like any electronic component,

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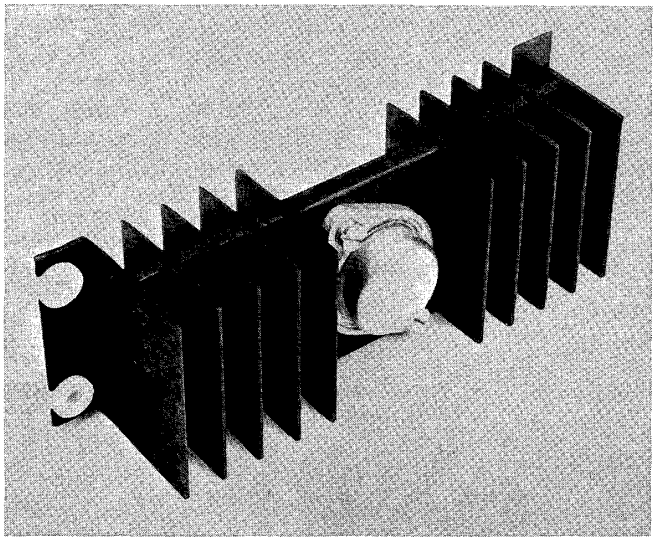

FIGURE 2.1
Transistor mounted on an aluminum heat sink (Photo courtesy of Wakefield Engineering, Inc.)

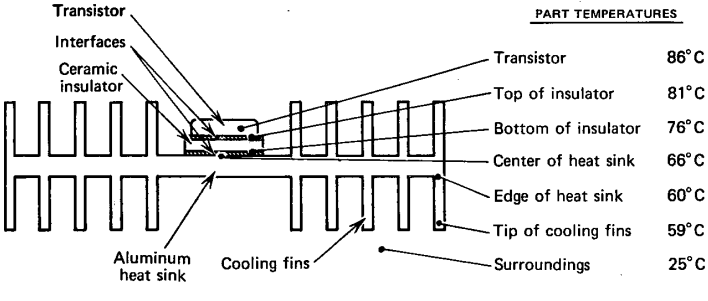

FIGURE 2.2
Cross section drawing of the transistor and heat sink of Figure 2.1 showing part temperatures

it is less than 100% efficient. This heat flows across the interface between the transistor and the ceramic insulator, through the ceramic insulator, across the interface between the ceramic insulator and the aluminum heat sink, through the heat sink to the cooling fins, and through the cooling fins themselves where the heat is transferred to the surrounding environment.

Heat always flows from high temperature parts to low temperature ones. Therefore, the electronic component is the hottest part. The total temperature difference required to conduct heat from the electronic component to the cooling fins is given by the following formula:

$$\Delta T\,(\text{conduction}) = \Delta T\,(\text{interface}) + \Delta T\,(\text{insulator}) + \Delta T\,(\text{metal}) + \Delta T\,(\text{fin}) \tag{2.1}$$

where:

ΔT (interface)	is the temperature difference required to conduct heat across the interfaces where the parts are mounted together.
ΔT (insulator)	is the temperature difference required to conduct heat through the insulating elements which make up the heat sink.
ΔT (metal)	is the temperature difference required to conduct heat through the metal elements which make up the heat sink.
ΔT (fin)	is the temperature difference required to conduct heat through the fins from their base to their tip.

Formulas for calculating all the temperature differences in Equation 2.1 are presented in this chapter.

The fins must be hotter than the surroundings to transfer heat. The temperature difference between the fins and the surroundings depends on the method of heat transfer, whether it is by radiation and natural convection, forced air cooling, forced liquid cooling, or liquid evaporation. This transfer temperature difference is discussed in detail in Chapters 3 through 6.

Figure 2.2 also shows the temperatures of the individual parts of the heat sink. When 10 W of heat are generated in the transistor, it must be 5°C hotter than the top side of the ceramic insulator to conduct the heat through the mounting interface between it and the insulator. The top surface of the ceramic insulator must be 5°C hotter than its bottom surface to conduct the heat through it. The bottom surface of the ceramic insulator must be 10°C hotter than the region of the heat sink directly under it to conduct the heat across this mounting interface. The region of the heat sink directly under the transistor must be 6°C hotter than the edges of the heat sink to conduct heat from the center of the heat sink out to its edges. Finally, the base of the

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fins must be 1°C hotter than their tip to conduct the heat along the fins. The calculation of these temperature differences, using the design information of this chapter, is illustrated in the first example of Section 2.7.

The total temperature difference from the transistor to the base of the fins must be 26°C to conduct 10 W of heat. If more power must be conducted, the temperature difference will be greater; if less power is conducted, the temperature difference will be less. The temperature difference is approximately directly proportional to the power that must be conducted. Therefore, it is possible to define a thermal resistance for the heat sink, which is as follows:

$$\text{Thermal resistance} \equiv \theta = \frac{\Delta T}{Q} \tag{2.2}$$

where:

ΔT is the temperature difference between the component and the cooling fins (°C).
Q is the power that must be conducted from the component to the fins (watts).

The thermal resistance of the design shown in Figure 2.1 is 2.6°C/W. This means that if 1 W of power must be conducted, the component will be 2.6°C hotter than the cooling fins; for 10 W of power, the component will be 26°C hotter; for 20 W of power, the component will be 52°C hotter, etc.

Because most electronic components, like the transistor, can operate over a range of power, the concept of thermal resistance is extremely useful. It permits the component temperature to be determined at whatever power the component is operating. The thermal resistance of most semiconductor diodes, SCRs, and transistors, is given on the component's data sheet. An example showing the calculation of the junction to case thermal resistance of a high frequency transistor is presented in the second example of Section 2.7 to illustrate the use of the design information of this chapter.

2.2 THE BASIC HEAT CONDUCTION FORMULA

The basic formula describing the conduction of heat through metals and through insulators is:

$$\Delta T = \frac{Q\lambda}{k\alpha} \tag{2.3}$$

where:

ΔT is the temperature difference required to conduct the heat (°C).
Q is the heat that is being conducted (watts).
λ is the length through which heat is conducted (inch).

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α is the cross section of the metal or insulator through which the heat is being conducted (inch^2).
k is the thermal conductivity of the metal or insulator (watts/inch °C).

The relationship between the dimensions used in Equation 2.3 is shown in Figure 2.3. An examination of Equation 2.3 reveals that the temperature difference is directly dependent on the power that must be conducted and on

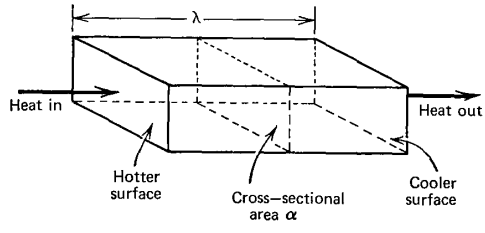

FIGURE 2.3
Geometry for basic heat conduction formula

the length through which it is conducted. The longer the length, the greater the temperature difference. The temperature difference depends inversely on the cross sectional area and on the thermal conductivity of the material. The greater the cross sectional area, the less the temperature difference.

2.3 THE MATERIALS OF THE HEAT SINK

The thermal conductivity of the materials commonly used in electronic equipment is tabulated in Table 2.1. The materials are listed alphabetically in the categories of metals, semiconductors, and insulators.

TABLE 2.1
Thermal Conductivity of Materials Commonly Used in Electronic Equipment

Material	Thermal Conductivity (Watts/Inch °C) at 100°C
Metals	
Aluminum	5.5
Beryllium	4.5
Beryllium copper	2.7
Brass (70% copper—30% zinc)	3.1
Copper	10.0

Continued

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TABLE 2.1—*Continued*

Material	Thermal Conductivity (Watts/Inch²°C) at 100°C
Gold	7.4
Iron	1.7
Kovar	0.42
Lead	0.87
Magnesium	4.0
Molybdenum	3.3
Monel	0.50
Nickel	2.3
Silver	10.6
Stainless steel 321	0.37
Stainless steel 410	0.61
Steel, low carbon	1.7
Tin	1.6
Titanium	0.40
Tungsten	5.0
Zinc	2.6
Semiconductors	
Gallium arsenide	1.5
Silicon (pure)	3.7
Silicon (doped to resistivity of .0025 ohm-cm)	2.5
Insulators	
Still air	0.0007
Alumina (99.5%)	0.70
Alumina (85%)	0.30
Beryllia (99.5%)	5.0
Beryllia (97%)	4.0
Beryllia (95%)	3.0
Boron nitride (hot pressed)	1.0
Diamond	16.0
Epoxy	0.005
"Thermally conducting" epoxy	0.02
Glass	0.02
"Heat sink compound" (metal oxide loaded epoxy)	0.01
Mica	0.018
Mylar	0.005
Phenolic	0.005
Silicone Grease	0.005
Silicone Rubber	0.005
Teflon	0.005

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The thermal conductivity of most of these materials varies somewhat with temperature. The values of thermal conductivity given in Table 2.1 are all for materials at 100°C.

The thermal conductivity of some of the metals listed in Table 2.1 is shown as a function of temperature in Figure 2.4. A similar graph is shown for ceramic insulators in Figure 2.5. Note that the thermal conductivity of the metals changes slowly with temperature, but that the thermal conductivity of the ceramic insulators changes rapidly. For example, when the

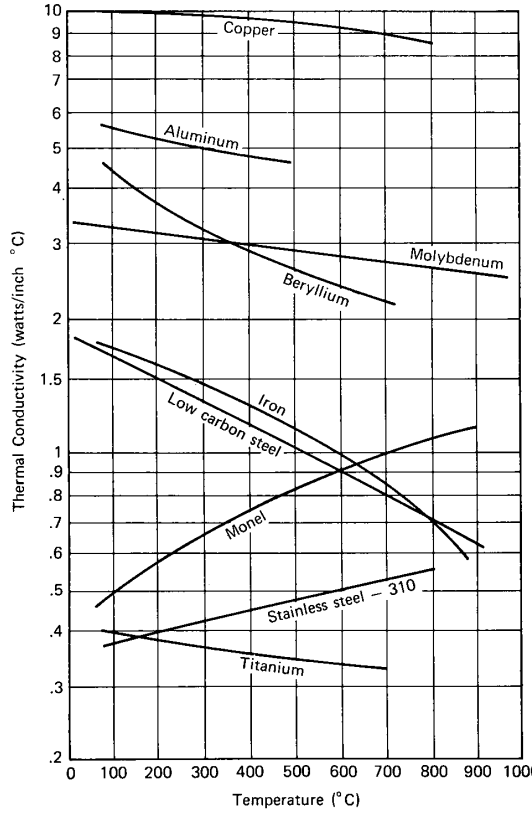

FIGURE 2.4
Thermal conductivity of metals as a function of temperature

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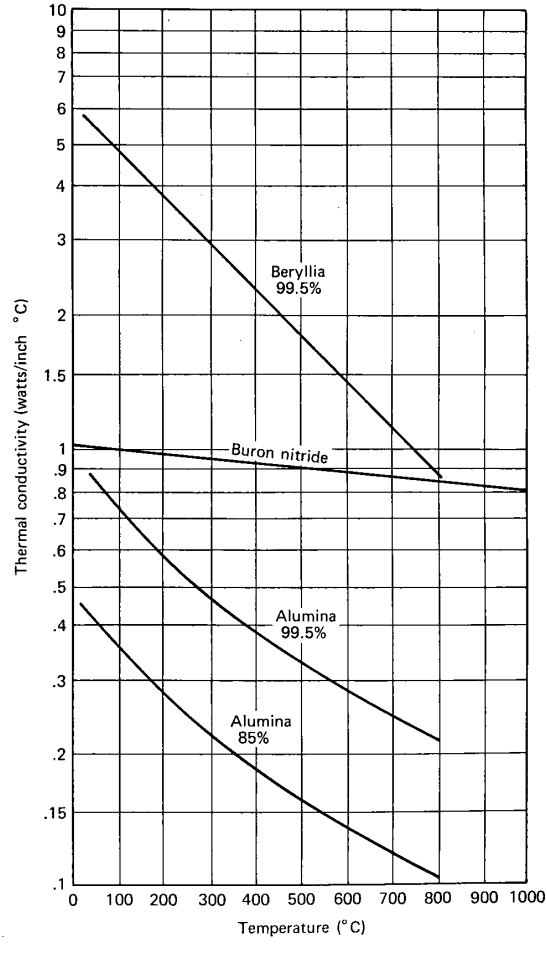

FIGURE 2.5
Thermal conductivity of ceramic insulators as a function of temperature

temperature changes from room temperature to 600°C, the thermal conductivity of the beryllia and alumina ceramics is reduced to one-third of the room temperature value.

The exact value of thermal conductivity of epoxy, phenolic, silicone rubber, and teflon depend on the particular type of material and so may vary around the values given in Table 2.1. The important fact, however, is that all of these materials have extremely poor conductivity, only 1/1000 that of aluminum.

It is instructive to use the basic heat conduction Equation 2.3 and the values of thermal conductivity of various materials from Table 2.1 to develop a feeling for the temperature differences that are to be expected. Assume that a power of 10 W must be conducted through a length of 1 in. in a part with a 1 in^2 cross section. The temperature difference required to conduct this heat through various materials is shown in Table 2.2.

TABLE 2.2
Temperature Difference Required to Conduct 10 W of Heat Through a 1 in. Length of Material with a Cross Section Area of 1 in^2.

Material	Temperature Difference (°C)
Copper	1
Aluminum	2
Low carbon steel	6
Stainless steel #321	25
Beryllia ceramic	2
Alumina ceramic	15
"Thermally conducting" epoxy	500

Of all the commonly used materials, copper is the best thermal conductor. Aluminum is very good also. Stainless steel is poor, with a temperature rise 25 times that of copper. Of the insulating materials, beryllia is extremely good—as good as aluminum metal. Note, however, that the high thermal conductivity of beryllia is critically affected by the material's purity and operating temperature. Alumina ceramic is much poorer than beryllia, but is better than stainless steel.

The poor thermal conductivity of epoxy is clearly evident from Table 2.2. For this comparison, the best thermally conductive epoxy (which has metal particles added to it to improve its conductivity) was used. This material has about four times better conductivity than typical epoxy potting compounds,

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but even so, the calculated temperature rise is 500°C. In a practical situation the epoxy could not stand this high temperature and would decompose.

2.4 THE GEOMETRY OF THE HEAT SINK

The basic heat conduction formula given in Equation 2.3 applies to designs where the cross section of the conducting metal or insulator is constant over the entire length through which heat is being conducted. Many complicated geometries can be approximately solved by this simple formula by:

1. Assuming an average cross section over the length.
2. Dividing the total length into several segments, each of which has a different cross section, and then applying Equation 2.3 consecutively to each segment.

Two other geometries which are commonly encountered in cooling designs for electronic equipment, and for which simple heat conduction equations are available, are shown in Figures 2.6 and 2.7. Figure 2.6 illustrates radial heat flow in which heat generated at one diameter flows radially outward to another diameter. In this case the cross sectional area of the material is continually increasing. The high power transmitter tubes shown in Figures 1.4 and 1.5 of Chapter 1 have this heat conduction geometry. This type of

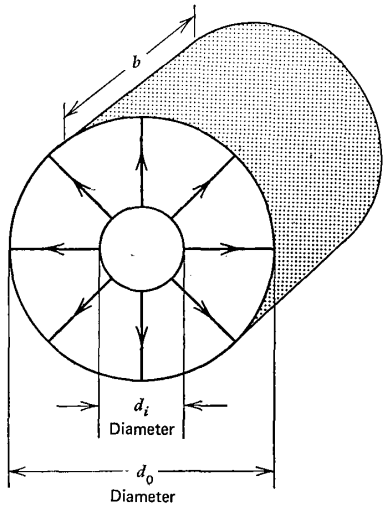

FIGURE 2.6
Radial heat flow geometry

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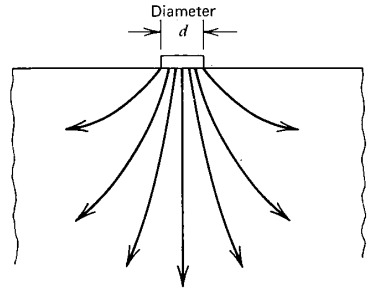

FIGURE 2.7
Spreading heat flow geometry

heat flow geometry also approximates the case of a small electronic component mounted in the center of a large thin heat sink plate.

Figure 2.7 illustrates the case of spreading heat flow where a small electronic component is mounted on an extremely large heat sink. In this case the heat flows radially away from the component as it spreads through the heat sink. The major temperature drop occurs very near the component itself where the heat is confined within a small volume.

The temperature difference required to conduct heat through the radial heat flow geometry shown in Figure 2.6 is given by the following formula:

$$\Delta T = \frac{Q}{2\pi kb} \log \frac{d_o}{d_i} \tag{2.4}$$

where:

ΔT	is the temperature rise required to conduct heat from one diameter to the other (°C).
Q	is the power that is being conducted (watts).
d_i	is the inner diameter (inch).
d_o	is the outer diameter (inch).
b	is the axial length of the cylinder (inch).
k	is the thermal conductivity of the material (watts/inch °C).
$\log d_o/d_i$	is the natural logarithm of the ratio of the diameters.

The temperature difference required to conduct heat in the spreading heat flow geometry shown in Figure 2.7 is given by the following formula:

$$\Delta T = \frac{2}{\pi} \frac{Q}{kd} \tag{2.5}$$

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where:

ΔT is the temperature difference between the electronic component and the distant end of the heat sink (°C).
Q is the power that is being conducted (watts).
d is the diameter of the electronic component (inch).
k is the thermal conductivity of the heat sink material (watts/inch °C).

Involved mathematical techniques are described in Reference 1 for the solution of heat conduction in complicated geometries. For most electronic cooling designs, the most practical approach is to synthesize the heat flow geometry into one of the three cases for which simple formulas are available and then make an approximate calculation. When the electronic equipment has been built, the temperature distribution can be measured. Techniques for measuring temperature distribution are presented in detail in Chapter 10.

2.5 MOUNTING INTERFACES

The mounting of the electronic component to the heat sink and the mounting of the various parts of the heat sink to each other present critical thermal problems.

These mounting problems are illustrated in Figure 2.8, where magnified views of the interface between two parts are shown. The surfaces of the parts are not perfectly smooth and flat on a microscopic basis as illustrated in

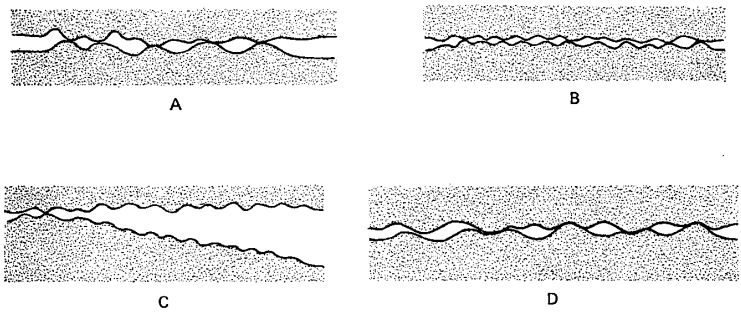

FIGURE 2.8
Magnified views of interfaces between two parts. A. Interface when parts have rough surfaces. B. Interface when parts have smooth surfaces. C. Interface when surfaces are not flat. D. Interface when parts are tightly clamped

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Figure 2.8A. Consequently, the two surfaces touch only at their high points. Heat is conducted across the interface only where the surfaces touch. Because they touch at only a few points, the cross sectional area through which heat can be conducted is greatly reduced, often to less than 1% of the total surface area. Unless great care is taken to maximize the contact area, the temperature difference required just to conduct heat across the interface can be as high as that required to conduct heat through several inches of the metal heat sink.

To minimize the temperature difference across the mounting interfaces the following steps can be taken:

1. Improve the surface finish of the heat sink and the electronic component.
2. Apply large clamping pressures to force the component against the heat sink.
3. Solder the electronic component to the heat sink.
4. Use thermal paste ("heat sink compound") between the electronic component and the heat sink.

The Effect of Surface Finish and Contact Pressure

The effect of improving surface finish is illustrated in Figure 2.8B. With a rough surface, such as shown in Figure 2.8A, the parts make contact at only a few points. When the surface finish is improved as in Figure 2.8B, the peaks and valleys along the surfaces are reduced, and the parts make contact at more points. Consequently, the effective cross sectional area for heat conduction is increased.

The overall flatness of the surfaces is even more critical than their surface finish, as illustrated in Figure 2.8C. If the surfaces do not touch at all, no amount of improvement in surface finish will help.

The effect of clamping the surfaces tightly together is illustrated in Figure 2.8D. The microscopic detail of the surfaces is distorted by the clamping pressure to push down the high points and bring the surfaces in better contact. The clamping pressure must be applied uniformly so that the overall flatness of the surfaces is not impaired; otherwise the clamping will bring about the condition illustrated in Figure 2.8C where the surfaces do not touch at all.

Quantitative values for the effect of surface finish and clamping pressure on the temperature difference required to conduct heat across an interface are shown in Figure 2.9. This figure shows the thermal resistance per unit area as a function of contact pressure with surface finish as a parameter. Curves are shown for aluminum and steel heat sinks. Thermal resistance is

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simply the temperature difference required to conduct heat across the interface per watt of power. As Figure 2.9 shows, by increasing the clamping pressure the temperature difference is decreased. Also, the better the surface finish, the lower the temperature difference at a given clamping pressure.

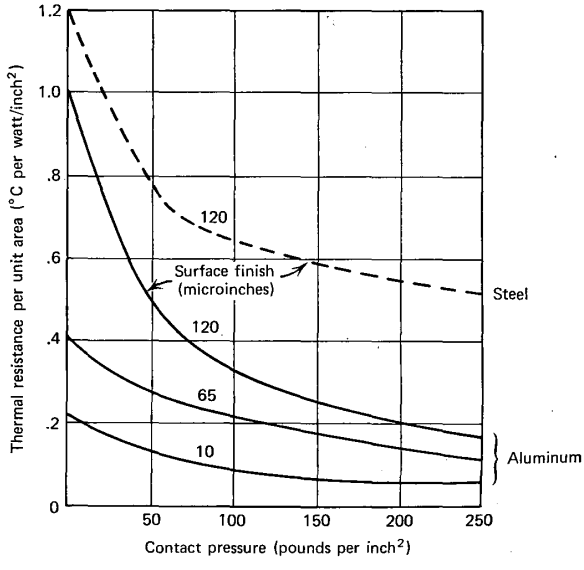

FIGURE 2.9
Thermal resistance of interfaces as a function of contact pressure

Most of the data presented in Figure 2.8 are for two aluminum surfaces. If a different material with a different hardness were used the results would be different, as shown by the one curve for two steel surfaces.

The better the surface finish and the greater the clamping pressure, the greater the amount of heat that can be transferred across the interface for a given temperature rise.

A simple calculation using Figure 2.9 will show the critical nature of the mounting interfaces. Assume that the surfaces of both the electronic component and the heat sink are aluminum and are machined to a 120 microinch finish, that the contact pressure across the surface is 50 psi (pounds per inch^2), that the overall contact area is 1 in^2, and that 10 W of power must be conducted across the interface. From Figure 2.9, the thermal resistance under these conditions is .5°C/W. To conduct 10 W of power across this

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interface would require a temperature difference of 5°C. Referring back to Table 2.2, this temperature difference is the same as is required to conduct the same 10 W of heat through a 2 1/2 in. length of aluminum bar! By increasing the surface finish to 10 microinches and increasing the contact pressure to 100 psi, the temperature difference across the interface can be reduced to only 1°C for the conduction of 10 W of heat, but this is still as much temperature difference as is necessary to conduct the same amount of heat through a 1/2 in. long aluminum bar.

Soldering the Component to the Heat Sink

The best means of eliminating the mounting interface problem is to solder the electronic component to the heat sink. In this way, the voids between the component and the heat sink surface are filled with solder metal. However, soldering the component to the heat sink does have the following disadvantages:

1. Replacement of the electronic component is difficult.
2. The heat sink and the component must be plated so that the solder will wet the surfaces.
3. Special care must be taken so that the component is not damaged by the heat applied during soldering.

Heat Sink Compounds

An alternative to soldering the component to the heat sink is to clamp the component to the heat sink and use a thermally conductive paste or heat sink compound at the interface. Suitable commercially available heat sink compounds are the following.

Manufacturer	Type
Dow Corning	Type 340 Silicone Heat Sink Compound
Emerson & Cumming	Ecotherm TC-4
Wakefield Engineering	Type 120 Thermal Joint Compound

These heat sink compounds are made of silicones loaded with thermally conductive metal oxides. They are in the form of thick, white pastes which fill the voids between the mating surfaces of the electronic component and the heat sink. They will not leak out of the interface, dry out, harden, or melt even after long exposure at temperatures up to 200°C.

Figure 2.10 shows the correct and incorrect use of heat sink compound. Only a small amount of compound should be applied to just fill the voids

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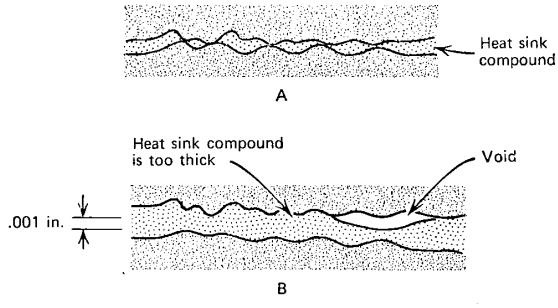

FIGURE 2.10
Magnified views of interfaces when heat sink compound is used.
A. Correct application. B. Incorrect application

and still allow the high points of the metal surfaces to touch, as shown in Figure 2.10A. The surfaces must be perfectly flat so that the thickness of heat sink compound is minimized across the entire surface. Care should be taken so that no brush hairs or foreign particles get into the heat sink compound to separate the surfaces.

Incorrect techniques in using heat sink compound are illustrated in Figure 2.10B.

The heat sink compounds have a thermal conductivity of .01 to .04 W/in. °C. Note that a thermal conductivity of .01 W/in. °C is 1/1000 that of copper. Consequently, the same temperature difference occurs across a .001 in. layer of heat sink compound as occurs across a 1 in. length of copper. For this reason the layer of heat sink compound must be kept very thin.

In spite of their poor thermal conductivity, the use of heat sink compounds can significantly reduce the temperature difference across a mounting interface. A simple calculation will illustrate this. As discussed in a previous paragraph, the thermal resistance across a mounting interface is .5°C/W/in^2 if the parts have a 120 microinch surface finish and the clamping pressure is 50 psi. If, instead, the interface is filled with heat sink compound with an average thickness of .001 in., application of Equation 2.3 shows that the thermal resistance is reduced by a factor of 5 to .1°C/W/in^2. To obtain this much improvement without the use of heat sink compound would require an improvement in surface finish to 10 microinches and an increase in contact pressure to 100 psi.

Some experimental data which support the above calculation are shown in Figure 2.11. The figure shows the measured thermal resistance between a

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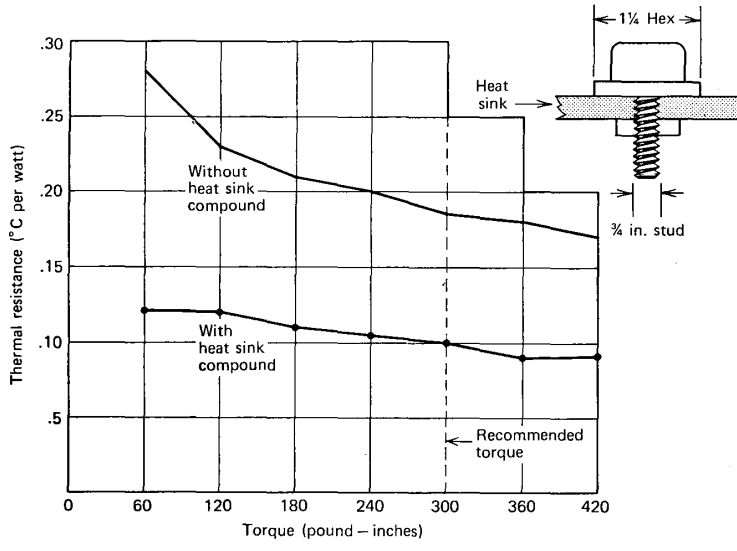

FIGURE 2.11
Measured thermal resistance of the interface between a high power SCR and a heat sink as a function of clamping torque

high power SCR (which has a mounting area of about 1 in^2) and its heat sink as a function of the torque applied to clamp the SCR to the heat sink.

The upper curve shows the results when heat sink compound is not used, and as expected the thermal resistance decreases with increasing clamping pressure. The lower curve shows the results when heat sink compound is used. Thermal resistance is reduced by a factor of 2 to .1°C/W, and is relatively independent of clamping pressure.

2.6 CONDUCTION OF HEAT THROUGH THE COOLING FINS

Heat conduction through cooling fins is different than heat conduction through a heat sink because the power that is conducted through the fins is not constant. This effect is illustrated in Figure 2.12A. At any point along the fins, part of the heat is transferred to the surrounding environment and the remainder of the heat is conducted to the adjacent section of the fin. The resulting power at any point along the length of the fin is shown in Figure

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2.12B. The power decreases from the total power at the base of the fin to zero at the tip of the fin, since when the tip is reached, all the power has been transferred to the surroundings. The corresponding fin temperature as a function of distance along the fin is shown in Figure 2.12C.

The surface of the cooling fin must be hotter than the surrounding environment to transfer heat. The hotter the fin surface, the greater the amount

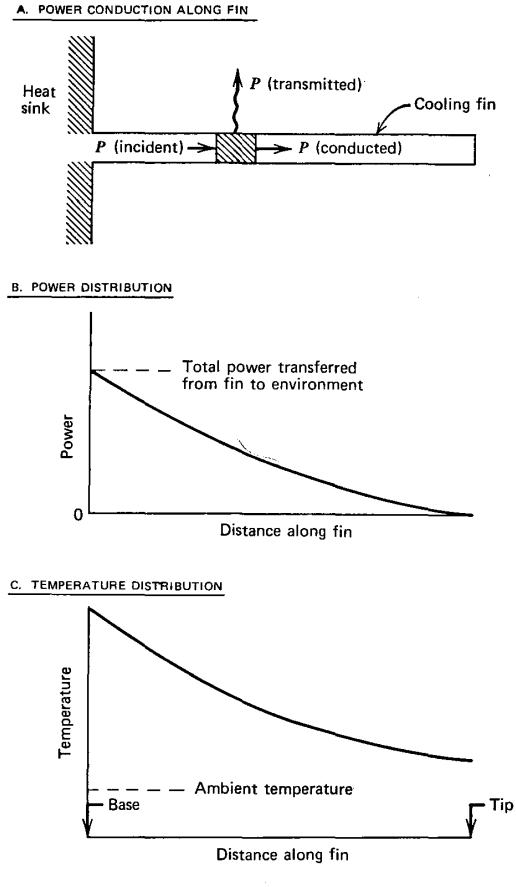

FIGURE 2.12
Conduction of heat through a cooling fin

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of heat that can be transferred. Therefore, the temperature difference between the base and the tip of the fin should be minimized, so that the tip of the fin will be as hot as possible to transfer the maximum amount of heat.

"Fin efficiency" is defined as the ratio of the heat actually transferred from the fin to the heat that would have been transferred if the fin surface were at a constant temperature equal to its maximum temperature at the base. Fin efficiency, therefore, takes into account the effect of the temperature difference from the base to the tip of the fin on its heat transfer capability.

Fin efficiency is determined both by the thermal conduction characteristics of the fins and by their heat transfer characteristics. If the heat transferred from the fins is small (for example, if the heat is transferred by radiation and natural convection) then the fins can be long and thin and made of a material whose thermal conductivity is low. If the heat transferred is large (as in the case of forced liquid cooling) then the fins must be thick and short and made of a material with a high thermal conductivity.

Fin efficiency and the base to tip temperature difference are shown in Figure 2.13 as a function of the dimensionless factor:

$$\frac{\left(\dfrac{\Delta T}{Q}\right)_{\text{conduction}}}{\left(\dfrac{\Delta T}{Q}\right)_{\text{transfer}}} \tag{2.6}$$

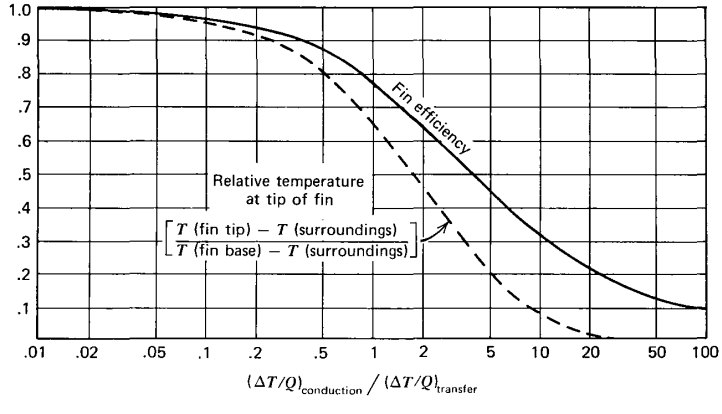

FIGURE 2.13
Fin efficiency and the relative temperature at the tip of the fin as a function of $(\Delta T/Q)_{\text{conduction}}/(\Delta T/Q)_{\text{transfer}}$

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where:

$\left(\frac{\Delta T}{Q}\right)_{\text{conduction}}$ is the temperature difference required to conduct a constant amount of heat from the base to the tip of the fin, neglecting the effect of heat transfer from the fins.

$\left(\frac{\Delta T}{Q}\right)_{\text{transfer}}$ is the uniform temperature difference between the fins and the surroundings required to transfer the constant amount of power from the fins to the surroundings, neglecting the effect of heat conduction through the fins.

The temperature difference required to conduct heat from the base to the tip of the fin, neglecting heat transfer, which is the $(\Delta T/Q)_{\text{conduction}}$ term of Equation 2.6, can be directly calculated from Equation 2.3.

The temperature difference required to transfer this heat from the fins to the surroundings, neglecting the effect of heat conduction, which is the $(\Delta T/Q)_{\text{transfer}}$ term of Equation 2.6, depends on the method of heat transfer used—radiation and natural convection, forced air cooling, or forced liquid cooling. For any given type of heat transfer, the value depends on fin geometry and coolant flow rate, and can be calculated from the design information of Chapters 3 through 6.

Figure 2.13 shows that if the ratio of the temperature difference for heat conduction to the temperature difference for heat transfer is small, which would be the case if the fin were a very good conductor or a very poor radiator, then fin efficiency approaches 100%. If the ratio is one, that is, if the temperature difference required to conduct heat from the base to the tip of the fin is the same as the temperature difference required to transfer the heat from the fin to the surroundings, then fin efficiency is only 76%, and only 76% as much power is transferred from the fin as would be transferred if the entire fin surface were at the temperature of its base. Expressed another way, to transfer the same amount of heat as could be transferred if the fin were a perfect conductor, the base of the fin, where the electronic component is mounted, would have to be $1/.76 = 1.3$ times hotter.

Note also from Figure 2.13, that in this case the temperature difference between the tip of the fins and the surroundings is only 65% of the temperature difference between the base of the fins and the surroundings.

If the ratio of the temperature differences for heat conduction and heat transfer is 20, which would occur if the fin were a very poor conductor or an excellent radiator, then, as shown in Figure 2.13, fin efficiency is only 22% and the temperature of the tip is practically at the temperature of the surroundings. In this case very little power is transferred from the tip end of the fin.

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The proper choice of fin material to achieve a high fin efficiency is best illustrated by a simple example. Assume a 1 in. × 1 in. fin whose thickness is 1/16 in. The temperature difference required to conduct a unit of heat from the base to the tip, neglecting heat transfer from the fin, for various fin materials, can be calculated from Equation 2.3, and is as follows.

Fin Material	$\left(\frac{\Delta T}{Q}\right)_{\text{conduction}}$
Copper	1.6°C/W
Aluminum	2.9°C/W
Steel	9.4°C/W

The temperature difference between the fin surface and the surroundings required to transfer a unit of heat is as follows (from the design information of Chapters 3 through 5).

Heat Transfer Method	$\left(\frac{\Delta T}{Q}\right)_{\text{transfer}}$
Radiation and natural convection	50°C/W
Forced air cooling	5°C/W
Forced liquid cooling	.5°C/W

Fin efficiency can be determined from the above data for each type of fin material and each cooling method by the use of Figure 2.13. The results are shown in Table 2.3.

TABLE 2.3
Fin Efficiency of a 1 in. × 1 in. × 1/16 in. Fin for Various Methods of Heat Transfer and Various Fin Materials

Heat Transfer Method	Fin Material Copper	Aluminum	Steel
Radiation and natural convection	99%	98%	94%
Forced air cooling	92%	85%	64%
Forced liquid cooling	54%	41%	22%

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For a fin with these dimensions, any of the three materials are suitable if the fin is cooled by radiation and natural convection. If forced air cooling is used, copper is the best choice of material, aluminum is satisfactory, and steel should not be used. For the forced liquid cooled case, only copper should be used, and even then the copper fins should be increased in thickness. / why — what's the criteria?

2.7 EXAMPLES

The temperature rise of any electronic component above its cooling fins can be calculated from the equations, tables, and figures of this chapter. The correct use of this design information is illustrated in this section by the following examples:

- The transistor heat sink described in Section 2.1.
- A transistor chip inside a stud mounted transistor package.
- A lead mounted silicon diode, potted in epoxy.

EXAMPLE 1
A Transistor Mounted on an Aluminum Heat Sink

Figures 2.1 and 2.2 showed a transistor mounted on an aluminum heat sink Also shown were the temperature differences required to conduct heat across the interface from the transistor to the ceramic insulator, through the ceramic insulator, across the interface between the ceramic insulator and the heat sink, through the heat sink, and finally through the cooling fins themselves from their base to their tip. These temperature differences will be calculated in this first example.

The important dimensions of the transistor and the heat sink are shown in Figure 2.14. The heat that must be conducted from the transistor is 10 W.

The first step is to calculate the temperature difference across the mounting interface between the transistor and the ceramic insulating washer. Heat sink compound .001 in. thick is used at this interface. The temperature difference is calculated using Equation 2.3:

$$\Delta T = \frac{Q\lambda}{k\alpha} = \frac{10\ \text{W} \times .001\ \text{in.}}{.01\ \text{W/in. °C} \times .20\ \text{in}^2} = 5°\text{C}$$

where:

λ (the length of heat sink compound through which heat is conducted) is .001 in.
α (the area of the mounting surface of the transistor) is .20 in^2.
k (the thermal conductivity of the heat sink compound) is .01 W/in. °C.

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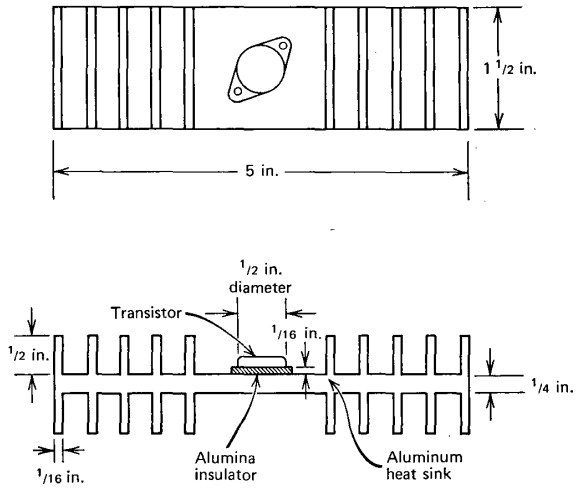

FIGURE 2.14
Dimensions of the transistor heat sink shown in Figure 2.1

The temperature difference required to conduct heat through the heat sink compound at the interface between the transistor and the insulating washer is therefore 5°C.

The second step is to calculate the temperature difference required to conduct the 10 W of heat through the alumina insulating washer. Again Equation 2.3 is used:

$$\Delta T = \frac{Q\lambda}{k\alpha} = \frac{10 \text{ W} \times .063 \text{ in.}}{.7 \text{ W/in. °C} \times .20 \text{ in}^2} = 5°\text{C}$$

where:

λ (the length of the alumina washer through which heat is conducted) is .063 in.
α (the cross sectional area of the alumina washer) is .20 in^2.
k (the thermal conductivity of the alumina washer) is .7 W/in. °C.

The value of thermal conductivity of .7 W/in. °C for the alumina ceramic washer was obtained from Table 2.1. The temperature difference required to conduct heat through the alumina washer is 5°C.

The third step is to calculate the temperature difference required to conduct heat across the interface between the alumina ceramic insulator and the heat

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sink. At this interface heat sink compound is not used. The alumina washer and the heat sink both have a 65 microinch finish and are clamped together with a pressure of 100 psi. The temperature difference required to conduct heat across this interface is determined from Figure 2.9. As shown, for a surface finish of 65 microinches and a clamping pressure of 100 psi, the thermal resistance across the interface is .2°C/W/in^2. The power density is 10 W/.2 in^2 or 50 W/in^2, so the temperature rise across the interface is 10°C. Better design practice would have been to metallize and solder the alumina washer to the heat sink and thereby completely eliminate this element of the total temperature rise.

The fourth step is to calculate the temperature difference required to conduct the heat from the region directly under the ceramic insulator through the heat sink to the base of the fins. The complicated heat flow geometry in this part of the example can be approximated by assuming that half of the power flows in each direction from the centerline of the heat sink to the ends, and using Equation 2.3:

$$\Delta T = \frac{Q\lambda}{k\alpha} = \frac{5\ \text{W} \times 2.5\ \text{in.}}{5.5\ \text{W/in. °C} \times .25\ \text{in.} \times 1.5\ \text{in.}}$$
$$= 6°\text{C}$$

where:

λ (the length of the heat sink from the center-line to the edge) is 2.5 in.
α (the cross sectional area of the heat sink) is .25 × 1.5 in^2.
k (the conductivity of aluminum from Table 2.1) is 5.5 W/in. °C.

The temperature rise from the region directly under the SCR to the outer edge of the heat sink is 6°C.

The final step is to calculate the fin efficiency and the temperature difference from the base of the fins to their tip. Anticipating the results of Chapter 3, $(\Delta T/Q)_{\text{transfer}} = 3.5°\text{C/W}$ for this cooling fin assembly of 20 fins, and $(\Delta T/Q)_{\text{transfer}} = 70°\text{C/W}$ for a single fin.

$(\Delta T/Q)_{\text{conduction}}$ can be calculated from Equation 2.3:

$$\left(\frac{\Delta T}{Q}\right)_{\text{conduction}} = \frac{\lambda}{k\alpha} = \frac{.5\ \text{in.}}{5.5\ \text{W/in. °C} \times .063\ \text{in.} \times 1.5\ \text{in.}}$$
$$= 1°\text{C/W}$$

where:

λ (the length of the fin) is .5 in.
α (the cross sectional area of the fin) is .063 × 1.5 in.
k (the conductivity of aluminum) is 5.5 W/in. °C.

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$$\frac{\left(\frac{\Delta T}{Q}\right)_{\text{conduction}}}{\left(\frac{\Delta T}{Q}\right)_{\text{transfer}}} = \frac{1°\text{C/W}}{70°\text{C/W}} = .014.$$

From Figure 2.13, fin efficiency is close to 100% and the temperature difference between the tip of the fins and the surroundings is 99% of the temperature difference between the base of the fins and the surroundings.

Since $(\Delta T/Q)_{\text{transfer}}$ is 3.5°C/W for the complete finned heat sink and 10 W of power is being transferred, the base of the fins is 35°C above the surroundings. If the surroundings are at 25°C, the base of the cooling fins are at 60°C. The tip of the fins are 99% × 35°C = 34.6°C above the surroundings or less than 1°C different in temperature from the fin base.

From the above calculations, the total temperature difference from the transistor to the base of the cooling fins is:

	Temperature Difference
Across the interface between the transistor and the insulating ceramic washer	5°C
Through the insulating ceramic washer	5°C
Across the interface between the ceramic washer and the heat sink	10°C
Through the heat sink	6°C
TOTAL	26°C

Since the base of the cooling fins are at 60°C, the transistor mounting surface is at 86°C when 10 W is being dissipated.

EXAMPLE 2
A Transistor Chip Inside a Stud-Mounted Transistor Package

Figure 2.15 shows a photograph of a microwave transistor, and an enlarged drawing of the inside of the transistor showing the silicon chip, that beryllia ceramic insulator, and part of the copper mounting stud. The important dimensions are also shown.

In this example, the junction-to-stud thermal resistance will be calculated. This thermal resistance is the temperature difference per watt of dissipated

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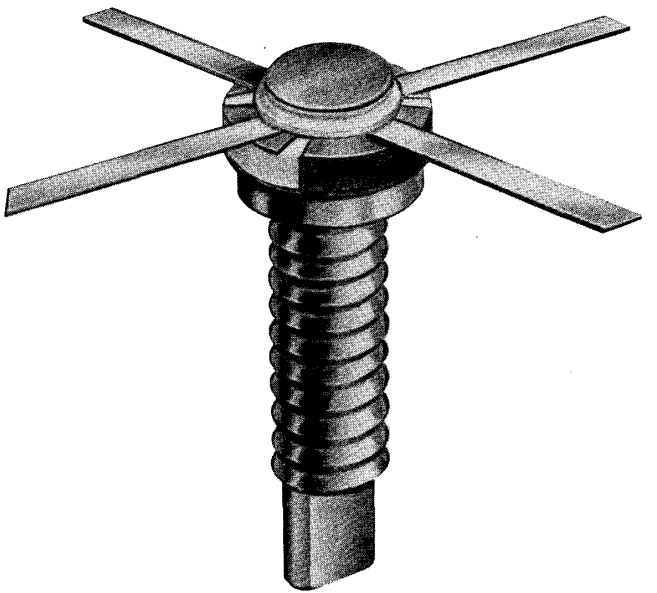

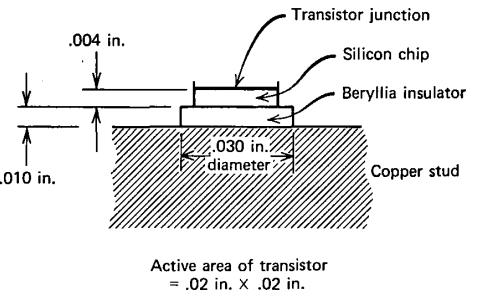

FIGURE 2.15
A high frequency transistor with its important internal dimensions

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power between the base-collector junction inside the silicon chip and the mounting stud.

The first step is to calculate the temperature difference required to conduct 1 W of power from the base-collector junction, where the heat is generated, through the silicon chip to the ceramic insulator. This temperature difference can be calculated from Equation 2.3 as follows:

$$\frac{\Delta T}{Q} = \frac{\lambda}{k\alpha} = \frac{.004 \text{ in.}}{2.5 \text{ W/in. °C} \times .0004 \text{ in}^2} = 4\text{°C/W}$$

where:

λ (the length in the silicon chip through which heat must be conducted) is .004 in.
α (the active area of the transistor) is .0004 in^2.
k (the thermal conductivity of silicon) is 2.5 W/in. °C.

The value for the thermal conductivity of silicon was obtained from Table 2.1. The thermal resistance from the junction through the silicon chip is therefore 4°C/W.

The second step is to calculate the temperature difference required to conduct 1 W of power through the beryllia insulator. The silicon chip is soldered to the beryllia insulator and the insulator in turn is soldered to the copper stud so mounting interfaces can be neglected. Again Equation 2.3 is used:

$$\frac{\Delta T}{Q} = \frac{\lambda}{k\alpha} = \frac{.010 \text{ in.}}{4 \text{ W/in. °C} \times \pi/4 \times (.030)^2 \text{ in}^2} = 3\text{°C/W}$$

where:

λ (the length of the beryllia insulator through which heat is conducted) is .010 in.
α (the cross sectional area of the beryllia insulator) is $(\pi/4)\ (.030)^2$ in^2.
k (the thermal conductivity of the beryllia insulator) is 4 W/in. °C.

The value for the thermal conductivity of beryllia was obtained from Table 2.1. The thermal resistance through the beryllia insulator is therefore 3°C/W.

The third step is to calculate the temperature difference required to conduct 1 W of power through the copper stud from the area directly under the beryllia insulator to the distant end. The "spreading heat flow" formula (Equation 2.5) should be used:

$$\frac{\Delta T}{Q} = \frac{2}{\pi k d} = \frac{2}{\pi \times 10 \text{ W/in. °C} \times .030 \text{ in.}} = 2\text{°C/W}$$

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where:

d (the diameter where the heat enters the copper stud) is .030 in.
k (the thermal conductivity of copper) is 10 W/in. °C.

The thermal resistance to the conduction of heat through the copper stud is 2°C/W.

The final step is to combine the thermal resistance of each part to determine the total junction to stud thermal resistance.

Part	Thermal Resistance (°C/W)
Silicon chip	4
Beryllia insulator	3
Copper stud	2
TOTAL	9°C/W

what's the case?

Therefore, the total junction to case thermal resistance is 9°C/W. If 10 W of power is dissipated in the transistor, the junction must be 90°C hotter than the mounting stud. The mounting stud temperature will be 86°C if the transistor is mounted to the heat sink of Example 1, and in this case the junction temperature would be 176°C.

EXAMPLE 3
A Lead-Mounted Silicon Diode, Potted in Epoxy

All electronic components need cooling, including those which must dissipate only a small amount of power. A typical low power example is a silicon diode rectifier. The total power dissipated in the rectifier is 1 W. Often the electronic equipment will be potted in epoxy, and heat can be transferred from the diode by conduction through its leads and through the epoxy.

A drawing of the silicon diode rectifier with its important dimensions is shown in Figure 2.16.

The first step is to calculate the temperature difference if heat is conducted from the diode through its leads, neglecting conduction through the epoxy.

One-half watt of heat must be conducted through each lead. From Equation 2.3:

$$\Delta T = \frac{Q\lambda}{k\alpha} = \frac{1/2 \text{ W} \times 1 \text{ in.}}{10 \text{ W/in. °C} \times \pi \times (.025)^2 \text{ in}^2} = 26°\text{C}$$

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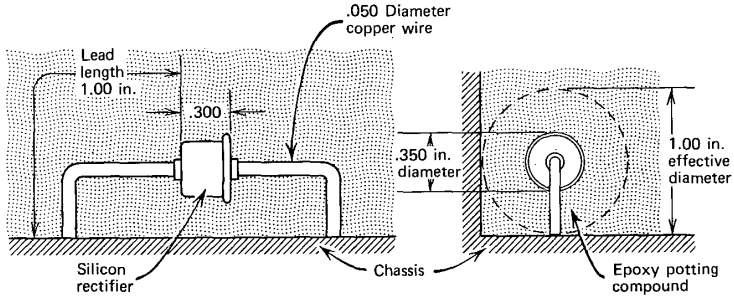

FIGURE 2.16
A lead-mounted silicon diode, potted in epoxy, with its important dimensions

where:

Q (the power that must be conducted through each lead) is 1/2 W.
λ (the length of the leads) is 1 in.
k (the thermal conductivity of copper) is 10 W/in. °C.
α (the cross sectional area of the leads) is $\pi \times (.025)^2$ in^2.

The temperature difference between the case of the silicon diode rectifier and the heat sink is 26°C.

For comparison, the temperature difference if heat is conducted through the epoxy, neglecting conduction through the leads, should be calculated.

The radial heat flow Equation 2.4 provides a reasonable approximation to this geometry:

$$\Delta T = \frac{Q}{2\pi kb} \log \frac{d_o}{d_i} = \frac{1 \text{ W}}{2\pi \times .005 \text{ W/in. °C} \times .300 \text{ in.}} \log \frac{1.000}{.350} = 112°\text{C}$$

where:

d_i (the inner diameter of the epoxy, which is the same as the diameter of the diode) is .350 in.
d_o (the effective outer diameter of the cylindrical heat flow geometry) is 1.00 in.
b (the axial length of the effective cylinder, which is the same as the length of the diode) is .300 in.
k (the thermal conductivity of the epoxy from Table 2.1) is .005 W/in. °C.

The temperature difference required to conduct heat from the diode through the epoxy is approximately four times greater than the temperature

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difference required to conduct heat through the diode leads. In other words, potting the diode in epoxy does very little to improve the conduction of heat from the diode.

2.8 REFERENCES

Useful references on heat conduction in electronic equipment are:

1. Krauss, A. D., *Cooling Electronic Equipment*, Prentice Hall, Inc., Englewood Cliffs, N.J., 1965. Chapter 1, pp. 1–29, Chapter 6, pp. 112–127.
2. *Guide Manual of Cooling Methods for Electronic Equipment*, Navships Publication 900, 190, Navy Bureau of Ships, Washington, D.C., 1955, pp. 17–27.
3. Goldman, W., *A Primer on Heat Sinking*, Wakefield Engineering, Inc., Wakefield, Mass., 1966.
4. Gutzwiller, F. W., ed., *SCR Manual*, General Electric Company, Chapter 17, pp. 341–365, Syracuse, N.Y., 1967.
5. Rosebury, F., *Handbook of Electron Tube and Vacuum Techniques*, Addison Wesley Publishing Co., Reading, Mass., 1965.

Reference 1 gives the basic heat conduction equations in Chapter 1, and presents a thorough discussion of fin efficiency in Chapter 6.

Reference 2 gives the basic heat conduction equations and a good discussion of thermal interfaces.

Reference 3 discusses the practical problems of mounting semiconductors to heat sinks.

Reference 4 provides instructions for proper mounting of high power SCRs to heat sinks to minimize thermal interfaces.

Reference 5 gives the thermal conductivities of most of the metals and insulators used in electronic equipment.

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3

Radiation and Natural Convection

Radiation and natural convection are the simplest to use of all the heat transfer methods. No auxiliary equipment is required, just the cooling fins themselves. The hot fins radiate heat directly to the cooler surroundings. At the same time, the air near the hot fins is heated and rises and is replaced by cooler air. This convective air current provides additional heat transfer.

A typical heat sink cooled by radiation and natural convection is described in Section 3.1. In most electronic equipment, heat transfer occurs by radiation and natural convection simultaneously. However, the amount of heat transferred by each method depends on heat sink temperature, geometry, and orientation in different ways.

Design information for calculating heat transfer by radiation will be discussed in Section 3.2. Natural convection will be discussed in Section 3.3. For a particular electronic cooling design, the heat transferred by each method must be calculated separately and then combined.

Examples of electronic equipment cooled by radiation and natural convection are presented in Section 3.4 to illustrate the use of the equations and figures of the chapter.

3.1 A TYPICAL COOLING DESIGN USING RADIATION AND NATURAL CONVECTION

Figure 3.1 shows a typical heat sink designed for cooling by radiation and natural convection. The electronic component is mounted in the center of the heat sink, and heat is conducted from the component to the cooling

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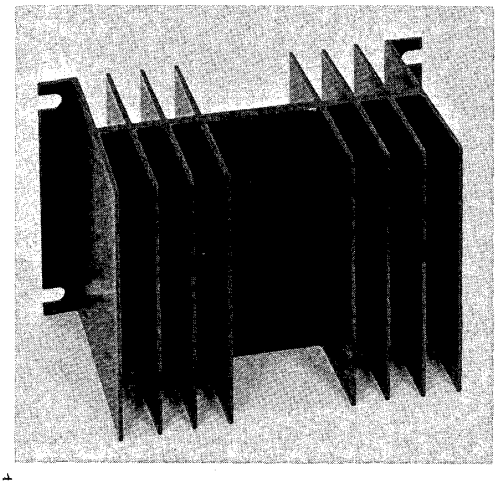

FIGURE 3.1
Heat sink cooled by radiation and natural convection (Photo courtesy of Wakefield Engineering, Inc.)

fins and is then transferred from the cooling fins to the surroundings by radiation and natural convection. The conduction of heat from the component to the cooling fins was discussed in detail in Chapter 2.

The measured temperature of the cooling fins of the particular heat sink shown in Figure 3.1 is shown in Figure 3.2 as a function of the power dissipated by the electronic component. At a power dissipation of 60 W, the temperature of the fins is 100°C, which is 80°C above the temperature of the surroundings. Of the total transferred power of 60 W, approximately 20 W is transferred by radiation and 40 W is transferred by natural convection.

The effectiveness of the various methods of heat transfer were compared in Figure 1.2 of Chapter 1. The value shown for radiation and natural convection was taken from the cooling fin design shown in Figure 3.1. When the fin temperature is 100°C and temperature of the surroundings is 20°C, 60 W of power is transferred. The total surface area of the fins is 107 in^2, so the heat transferred is .6 W/in^2.

The heat transferred from the heat sink shown in Figure 3.1 is calculated in Example 1 of Section 3.4, to illustrate the design information of this chapter.

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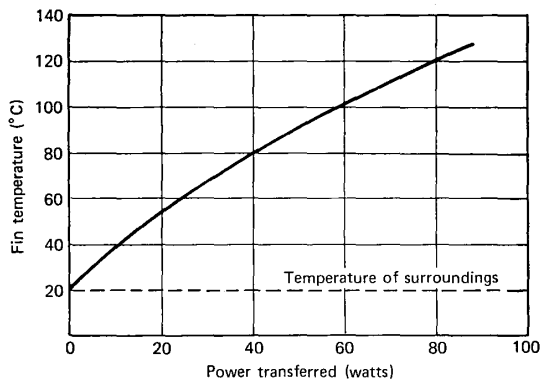

FIGURE 3.2
Measured temperature of the heat sink shown in Figure 3.1 as a function of power transferred

3.2 RADIATION

The amount of heat that can be transferred by radiation depends on:

1. The temperature of the radiating surface.
2. The temperature of the surroundings.
3. Surface conditions of the fins.
4. Shielding effects of adjacent fins.

These factors are related by the following equation:

$$Q = Q_r \varepsilon R S \tag{3.1}$$

where:

Q is the power transferred by radiation (watts).
Q_r is the power per unit area transferred by a perfect, unshielded radiator, and depends only on the temperature of the surface and the temperature of the surroundings (watts per inch^2).
ε is the emissivity of the surface.
S is the area of the surface (inch^2).
R is the reduction in effective surface area caused by the shielding effects of adjacent surfaces.

Design information on each of the factors in Equation 3.1 is presented in the following paragraphs.

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Temperature Dependence

The value of the factor Q_r of Equation 3.1, which is the amount of heat radiated per unit area of surface from a perfect, unshielded radiator, is shown as a function of surface temperature in Figure 3.3. Figure 3.3A shows radiated power over a wide range of surface temperatures up to 700°C. Figure 3.3B shows the same data with an expanded temperature scale over the temperature range up to 350°C. The parameter of the curves is the temperature of the surroundings into which the heat is radiated.

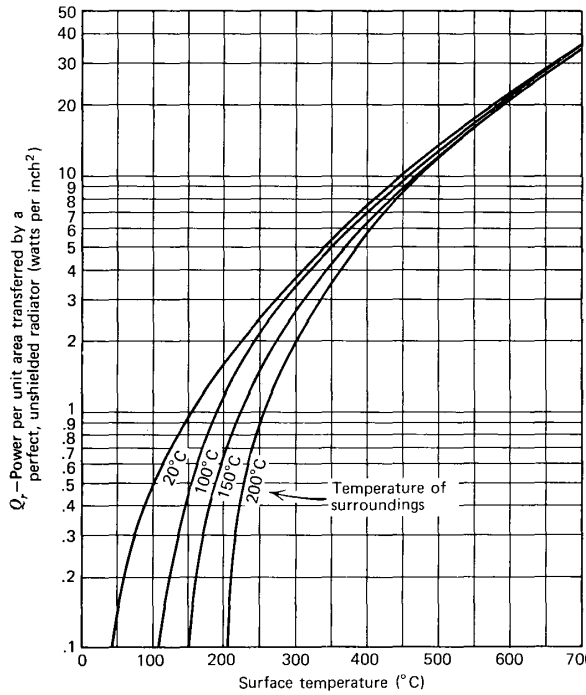

FIGURE 3.3
Power transferred by radiation as a function of surface temperature

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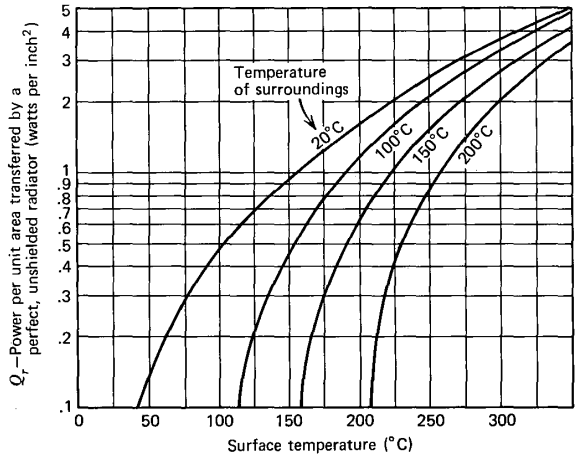

FIGURE 3.3A
Power transferred by radiation as a function of surface temperature

Surface Conditions

The curves of Figure 3.3 show the amount of heat that can be transferred by radiation from a perfect black body radiator. The factor specifying how good a particular surface is relative to a perfect radiator is the "emissivity" of surface, and is the factor ε in Equation 3.1. The emissivity of various materials and various surface treatments commonly used in electronic equipment is shown in Table 3.1.

Table 3.1 clearly shows the importance of surface characteristics on the amount of heat that can be transferred by radiation. If the surface is a polished metal, the heat that can be transferred by radiation is reduced to less than one-tenth of the value shown for an ideal surface in Figure 3.3. Oxidizing the metal surface improves its radiation capabilities to about two-thirds of the values shown for an ideal surface. Note that painting the fin surface with any color paint improves its emissivity to practically the ideal level.

Shielding by Adjacent Fins

The reduction in the effective radiating fin area due to the shielding effect of adjacent fins is shown in Figure 3.4. This is the factor R of Equation 3.1, and it is shown as a function of the ratio of fin height to fin spacing. Separate curves are shown for a square fin whose height is equal to its length, for a rectangular fin which is twice as long as it is high, and for a long rectangular

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TABLE 3.1
Emissivity of Materials Commonly Used in Electronic Equipment

Surface	Emissivity
Commercial aluminum (polished)	.05
Anodized aluminum	.80
Aluminum paint	.27 to .67
Commercial copper (polished)	.07
Oxidized copper	.70
Stainless steel (polished)	.17
Stainless steel (with heavy oxide)	.85
Rolled sheet steel	.66
Air drying enamel (any color)	.85
Oil paints (any color)	.92
Lamp black in shellac	.95
Varnish	.90
Zirconium coating on molybdenum	.65

fin whose length is very much greater than its height. The definitions of the width, the length, and the spacing of the fins are shown in the inset of Figure 3.4.

The shielding effect of adjacent fins significantly reduces the effective radiating area of a finned surface.

A simple calculation using Figure 3.4 clearly illustrates this. Assume heat is to be radiated from a 2 in. × 2 in. surface at 100°C. The surroundings are 20°C, and the surface has been painted black to maximize emissivity. From Figure 3.3, this surface will radiate .5 $W/in^2 \times 4\ in^2 = 2.0$ W. Next, assume that the surface is finned with 1 in. high by 2 in. long fins so that it appears as shown in the inset of Figure 3.4. Table 3.2 shows the amount of heat that will be radiated for different numbers of fins, using the curves of Figure 3.4.

The outer surface of the end fins are unshielded and have the same area as the original surface. The inner surface of the end fins, all other fins, and the original surface are all shielded by each other and their effective radiating area is reduced. Adding two fins at each end of the original surface increases the total area by 3 times. The shielding factor is .86 for this case, and the total radiated power is increased by 2.5 times from the case of the unfinned surface. Adding more fins in addition to the two end fins provides very little improvement. For example, when 9 fins are used, the total surface area is increased more than 3 times from the two fin case, but the shielding factor is .3, so the radiated power is increased only 1.35 times.

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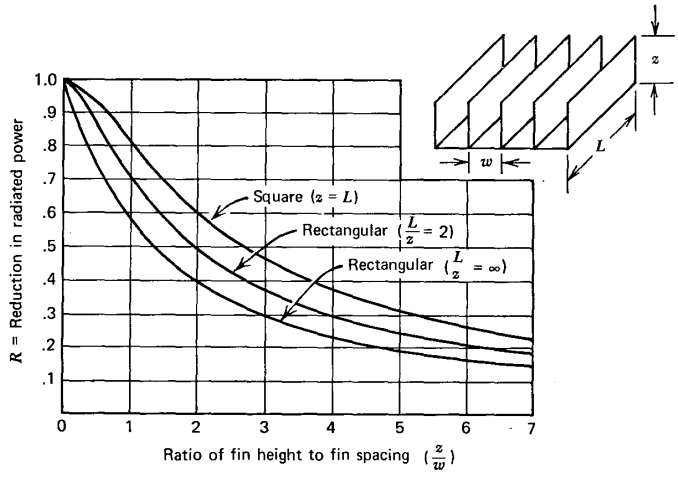

FIGURE 3.4
Radiation shielding effect of adjacent surfaces

particular

TABLE 3.2
Radiation From a Finned Surface – see example.

Number of Fins	Fin Spacing (inch)	Added Area (inch²)	Total Area (inch²)	$\frac{z}{w}$	R	Effective Added Area (inch²)	Radiated Power (watts)
0	—	0	4	—	—	—	2.0
2	2	8	12	1/2	.86	6.9	5.5
3	1	12	16	1	.72	8.6	6.3
5	1/2	20	24	2	.50	10.0	7.0
9	1/4	36	40	4	.30	10.8	7.4

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3.3 NATURAL CONVECTION

Figure 3.5 shows sketches of natural convection air flow over various orientations of heated surfaces. In the vertical orientation shown in Figure 3.5A, which is one of the best orientations for natural convection cooling, the hot air rises from the surface and is replaced by cool air from the bottom. In the case of a horizontal surface facing upward shown in Figure 3.5B, cool air flows along the surface from the sides and as it is heated, rises from the surface. If the horizontal surface faces downward, as shown in Figure 3.5C,

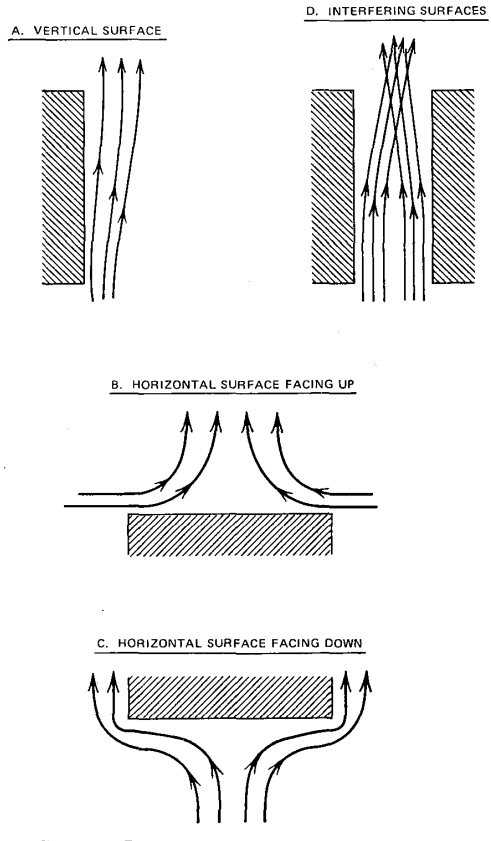

FIGURE 3.5
Natural convection air flow patterns

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the heated air must flow along the surface to the edge before it can then rise, and for this reason, convective cooling of this orientation is only half as effective as if the surface were vertical or facing upward.

Figure 3.5D shows two closely spaced vertical surfaces, where the heated air mixes near the top of the surfaces.

The sketches of Figure 3.5 suggest that the amount of heat that can be transferred by natural convection depends on the following factors:

1. Temperature difference between the surface and the surrounding air.
2. Dimensions of the surface.
3. Orientation of the surface.
4. Spacing between adjacent surfaces.
5. Altitude (which determines air density).

These factors can be related by the following equation:

$$Q = Q_c R_1 R_2 S \tag{3.2}$$

where:

Q is the power transferred by natural convection (watts).
Q_c is the power per unit area that is transferred from a 1 in. long vertical surface, and depends only on the temperature difference between the surface and the surroundings (watts per inch^2).
R_1 is a factor which depends on surface dimensions and orientation.
R_2 is a factor that depends on altitude.
S is the area of the surface (inch^2).

Design information on each of the factors in Equation 3.2 is presented in the following paragraphs.

Temperature Dependence

Figure 3.6 shows the power that is transferred by natural convection from a 1 in. high vertical surface as a function of the temperature difference between the surface and the surroundings. This is the factor Q_c of Equation 3.2.

If the surrounding air is at 20°C, and the surface is at 100°C, so that the temperature difference is 80°C, then as Figure 3.6 shows, 1/2 W of power is transferred by natural convection per inch^2 of surface area. This is the same amount of power as is transferred by radiation from a perfect, unshielded radiator at these same surface and surrounding temperatures.

Surface Dimensions and Orientation

The airflow sketches of Figure 3.5 suggest that the effectiveness of natural convection cooling depends on the length of the surface over which air moves and on the orientation of the surface. Figure 3.7 shows the factor R_1 of

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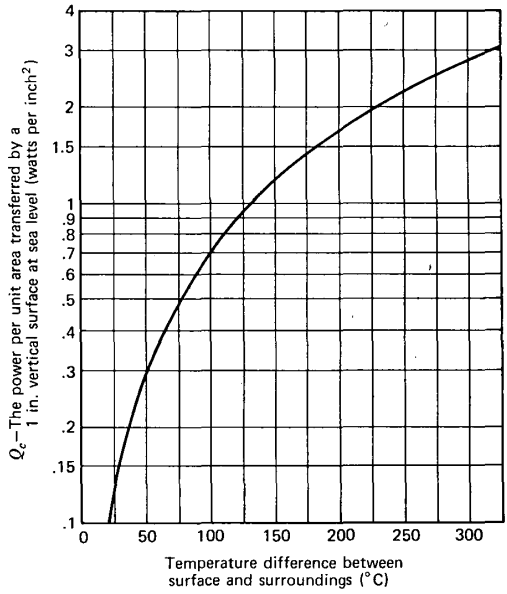

FIGURE 3.6
Power transferred by natural convection as a function of the temperature difference between the surface and the surroundings

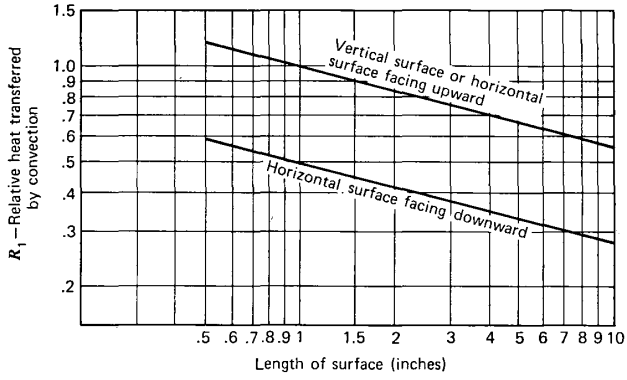

FIGURE 3.7
Relative effectiveness of natural convection cooling as a function of surface dimensions and orientation

Equation 3.2, which includes both of these effects. The relative effectiveness of natural convection cooling (R_1) is shown as a function of the "effective length" over which the air moves. Two curves are shown. One is for horizontal surfaces facing upward and vertical surfaces; the other is for horizontal surfaces facing downward.

The effective length of the surface along which the air moves is equal to the vertical height of a vertical surface. For a horizontal surface, the "effective length for use in Figure 3.7 is the geometric mean of the actual length and width of the surface, since the air moves in from all sides. The geometric mean is given by:

$$\frac{\text{width} \times \text{length}}{\text{width} + \text{length}}$$

For a 1 in. high vertical surface, the relative effectiveness factor (R_1) is 1, so the heat transferred per unit area by natural convection is simply the Q_c term of Equation 3.2, which was shown as a function of temperature in Figure 3.6. For other surface lengths or orientations, the relative effectiveness factor R_1 shown in Figure 3.7 must be included. For example, if a vertical surface is 3 in. high, the heat transferred per unit area by natural convection is reduced to 76% of the value for a 1 in. high surface shown in Figure 3.6.

As expected, cooling by natural convection is much better for a vertical surface or for a horizontal surface facing upward than for a horizontal surface facing downward.

For this reason, a heat sink such as shown in Figure 3.1 should always be operated with its fins vertical rather than horizontal. In the horizontal mounting, as shown by Figure 3.7, the transfer of heat by natural convection from the lower surface of the fins would be reduced by one half.

Spacing Between Adjacent Surfaces

The design information shown in Figures 3.6 and 3.7 applies to unshielded surfaces. If two surfaces are too close together, as shown in Figure 3.5D, the air flow from the surfaces interferes and the design curves no longer apply. For most surface dimensions used in electronic equipment, surfaces or fins can be brought to within 1/2 in. of each other without much mixing of the air currents, and the design information of Figures 3.6 and 3.7 can be used down to this spacing. The effects of spacing between adjacent surfaces is therefore much less critical for natural convection cooling than for radiation cooling.

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Altitude

The amount of natural convection cooling that can be obtained depends on air density, and so is a function of the altitude at which the equipment is operated.

Figure 3.8 shows the reduction of convective heat transfer as a function of altitude. This is the factor R_2 of Equation 3.2. As Figure 3.8 shows, convective heat transfer is only 80% as effective at an altitude of 10,000 ft as it is at sea level. At 35,000 ft it is only half as effective.

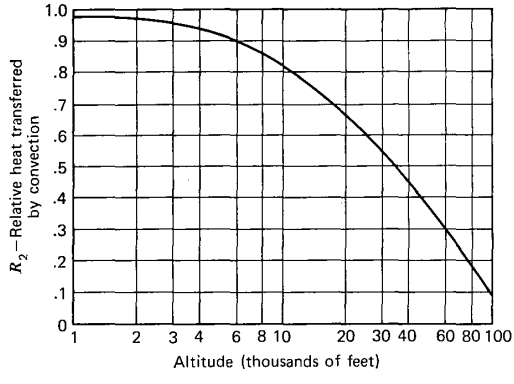

FIGURE 3.8
Relative effectiveness of natural convection cooling as a function of altitude

3.4 EXAMPLES

The amount of heat that can be transferred by radiation and natural convection can be calculated from the equations and figures of this chapter. The correct use of this design information is illustrated in this section by the following examples:

- The finned heat sink described in Section 3.1.
- A radiation cooled high power vacuum rectifier tube.
- A rack of card mounted components.
- Packaged airborne electronics equipment.

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EXAMPLE 1
A Finned Heat Sink

A typical heat sink for electronic components, cooled by radiation and natural convection, was described in Section 3.1 and was shown in Figure 3.1.

The power transferred from this heat sink when its fins are at 100°C and the surroundings are at 20°C will be calculated in this example. The important dimensions of the heat sink are shown in Figure 3.9. (The orientation of the heat sink in Figure 3.9 was chosen to best show its dimensions. In actual use, the heat sink would be mounted with the fins vertical.)

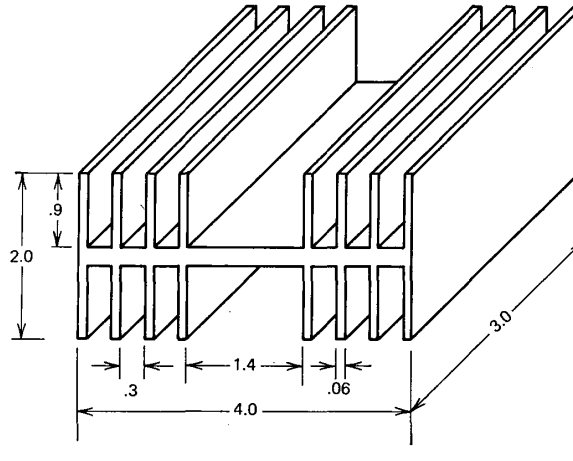

FIGURE 3.9
Dimensions of heat sink shown in Figure 3.1

The first step is to calculate the heat transferred by radiation, using Equation 3.1 and Figures 3.3 and 3.4. From Figure 3.3, when the heat sink surface is at 100°C and the surroundings are at 20°C, the radiated power is .46 W/in^2. The heat sink is painted black and, using Table 3.1, emissivity (ε) is .95.

The radiating area of the fins consists of the end surfaces which are not shielded and the U-shaped troughs between the fins which are shielded. For the 12 troughs where the fins are spaced .3 in. apart, the ratio of fin height to fin spacing is .9 in./.3 in. = 3 and the shielding factor R from Figure 3.4 is, therefore, .35 for rectangular fins whose length is three times their height.

For the two troughs in the center of the heat sink, the ratio of fin height to fin spacing is .9 in./1.4 in. = .64, and the shielding factor is .8.

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The effective radiating area is:

$$\begin{aligned} &12 \text{ troughs} \times (.9 + .3 + .9) \text{ in.} \times 3 \text{ in.} \times .35 \\ &+ 2 \text{ troughs} \times (.9 + 1.4 + .9) \text{ in.} \times 3 \text{ in.} \times .80 \\ &+ 2 \text{ ends} \times 2 \text{ in.} \times 3 \text{ in.} \times 1 \\ &= 54 \text{ in}^2. \end{aligned}$$

The effective radiating is 54 in^2, whereas the total surface area is 107 in^2. The power transferred by radiation, using Equation 3.1 is:

$$\begin{aligned} Q &= Q_r \varepsilon R S \\ &= .46 \text{ W/in}^2 \times .95 \times 54 \text{ in}^2 \\ &= 24 \text{ W}. \end{aligned}$$

The second step is to calculate the heat transferred by convection using Equation 3.2 and Figures 3.6 and 3.7.

The temperature difference between the fins and the surroundings is 100°C − 20°C = 80°C. From Figure 3.6, the heat transferred by convection is .53 W/in^2 for a 1 in. long vertical fin. The heat sink should be mounted with all fins vertical. The effective length along which the air moves is therefore 3 in. From Figure 3.7, the reduction factor R_1, which takes into account fin length and orientation, is .76. Assuming sea level operation, $R_2 = 1$. The spacing between fins of .3 in. is so small that some error will be introduced because of mixing of the air streams from adjacent fins. This error will be ignored in this calculation.

The heat transferred by convection, calculated from Equation 3.2, is therefore:

$$\begin{aligned} Q &= Q_c R_1 R_2 A \\ &= .53 \text{ W/in}^2 \times .76 \times 1 \times 107 \text{ in}^2 \\ &= 42 \text{ W}. \end{aligned}$$

The final step is to combine the effects of radiation and convection to find the total heat transferred from the heat sink.

Radiation	24 W
Convection	42 W
Total	66 W

The above calculations assume a perfectly conducting fin. The fin temperature must be corrected to account for the finite thermal conductivity of the aluminum fin. From Equation 2.3 of Chapter 2:

$$\left(\frac{\Delta T}{Q}\right)_{\text{conduction}} = \frac{z}{kyL} = \frac{.9 \text{ in.}}{5.5 \text{ W/in.°C} \times .06 \text{ in.} \times 3 \text{ in.}} = 0.87\text{°C/W}$$

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where the thermal conductivity of aluminum from Table 2.1 of Chapter 2 is 5.5 W/in. °C.

$(\Delta T/Q)_{\text{transfer}}$ is 80°C/66 W = 1.2°C/W for the heat sink of 16 fins, and, is therefore, 16 times greater or 19°C/W for a single fin.

$$\frac{\left(\frac{\Delta T}{Q}\right)_{\text{conduction}}}{\left(\frac{\Delta T}{Q}\right)_{\text{transfer}}} = \frac{.87°\text{C/W}}{19°\text{C/W}} = .046$$

and from Figure 2.13 of Chapter 2, fin efficiency is 98%. Therefore, the total power transferred from the fins is .98 × 66 W = 65 W.

The measured performance of this heat sink was shown in Figure 3.2. The calculated performance is in excellent agreement with the measured results.

EXAMPLE 2
A Radiation Cooled High Power Vacuum Rectifier Tube

In most electronic cooling designs, the temperature of the cooling fins must be limited to less than 300°C to prevent damage to electronic components and oxidation of the components and the fins. An exception to this rule is the radiation cooling of a high power vacuum rectifier tube. A photograph of such a tube is shown in Figure 3.10.

The power dissipated at the anode of the tube is radiated directly through the glass bulb to the surroundings. The vacuum environment around the anode protects it from oxidation so it can operate "red hot" at 650°C, and take advantage of the high radiation heat transfer that occurs at high temperatures. The anode is a 2 in. diameter × 4 in. long cylinder of molybdenum.

Molybdenum is used for the anode because of its good mechanical and vacuum properties at high temperature. The emissivity of any shiny metal is less than .1, so the outer radiating surface of the anode is coated with zirconium, which, as Table 3.1 shows, raises emissivity to .65.

Only radiation need be considered in this problem. Natural convection does not occur because the anode is in a vacuum.

The amount of heat transferred by radiation is calculated from Equation 3.1. From Figure 3.3, the power radiated from a perfect unshielded radiator is 28 W/in^2 when the radiating surface is at 650°C, and the surroundings are at 20°C.

Emissivity of the coated molybdenum surface is .65, and R is 1 since the anode is unshielded. The area of the anode is 2 in. × π × 4 in. = 25 in^2. From Equation 3.1:

$$\begin{aligned} Q &= Q_r \varepsilon R S \\ &= 28 \text{ W/in}^2 \times .65 \times 1 \times 25 \text{ in}^2 \\ &= 455 \text{ W.} \end{aligned}$$

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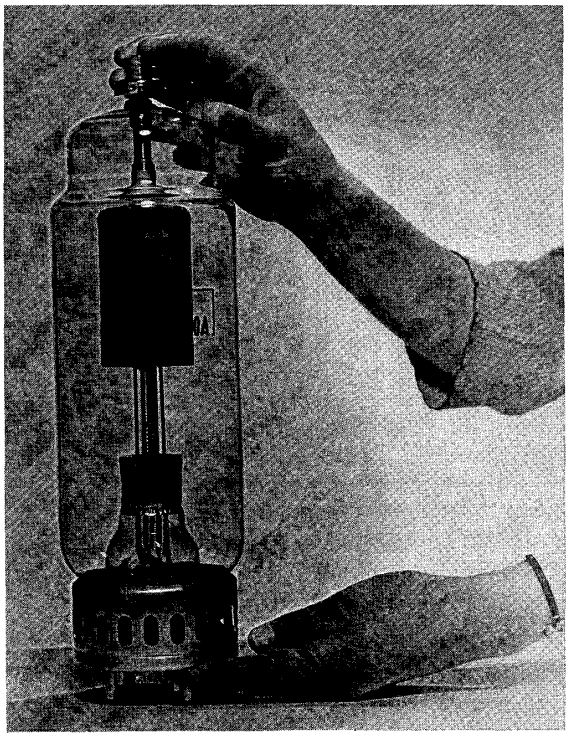

FIGURE 3.10
Radiation cooled high power vacuum rectifier tube (Photo courtesy of Varian Associates)

Therefore, at the rated anode dissipation of the tube of 450 W, the anode operates at about 650°C.

EXAMPLE 3
A Rack of Card-Mounted Components

Figure 3.11 shows a typical rack of card-mounted electronic components. One end of the rack has been removed to show the details of the components on the cards. The components are mounted on one side of 4 in. × 6 in. × 1/16 in. thick circuit board cards. Connections between components are made with printed circuitry on the opposite sides of the card.

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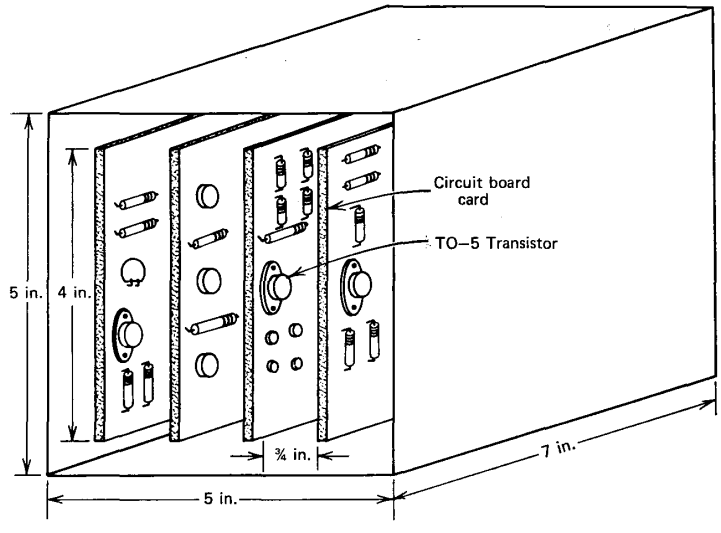

FIGURE 3.11
Rack of card-mounted electronic components cooled by radiation and natural convection

Four cards are mounted in a rack 5 in. wide × 7 in. long × 5 in. high. The top and bottom of the rack are open to allow natural convection cooling of the components. The spacing between the cards is approximately 3/4 in. when allowance is made for the height of the components mounted on the card surface.

Forced air cooling of this same configuration is discussed in Example 3 of Chapter 4. The same configuration will be considered again in Example 2 of Chapter 6, where cooling will be by liquid evaporation in a liquid filled package. By using the same configuration for each of these examples, the relative advantages and disadvantages of each cooling method can be compared.

In this present example, only cooling by natural convection needs to be considered. Radiation can be neglected because of the mutual shielding of the circuit board cards. Considering the cards as an array of fins, the ratio of their height to spacing is 4 in. divided by 3/4 in. or 5.3, and from Figure 3.4, the effective radiated power is only 25% that of an unshielded radiator. Also, the emissivity of the transistor cases and other components mounted on the cards is low. Therefore, radiation cooling can be neglected.

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The power that can be dissipated by a TO-5 transistor mounted in one of the cards will be calculated.

The transistor case temperature will be 70°C, and the temperature of the surrounding air will be 20°C. The temperature difference between the transistor and the surrounding air is therefore 50°C.

The convective air flow pattern will be controlled primarily by the card dimensions themselves. The length over which the air moves is the vertical height of the card of 4 in.

From Equation 3.2:

$$\begin{aligned} Q &= Q_c F_1 F_2 S \\ &= .3\ \text{W/in}^2 \times .7 \times 1 \times .30\ \text{in}^2 \\ &= .06\ \text{W}. \end{aligned}$$

Q_c is the power that would be transferred by natural convection from a 1 in. high vertical surface. The temperature difference between the transistor and the surrounding air is 50°C, and from Figure 3.6, Q_c is .3 W/in^2. F_1 is the correction factor for surface length and orientation. For a 4 in. high vertical surface, $F_1 = .7$ from Figure 3.7. At sea level, $F_2 = 1$. The surface area of the TO-5 transistor is:

$$\begin{array}{lll} \text{Top} & (\pi/4)\ (.3\ \text{in.})^2 & = .07\ \text{in}^2 \\ \text{Side} & \pi \times .3\ \text{in.} \times 1/4\ \text{in.} & = .23\ \text{in}^2 \\ \hline & & .30\ \text{in}^2 \end{array}$$

Therefore, the power that can be transferred from the transistor with a case temperature of 70°C by natural convection when it is mounted on a circuit board card in the rack is only .06 W. If more power must be transferred by natural convection from this card mounted transistor, its surface area must be increased by the use of a small finned heat sink.

EXAMPLE 4
A Packaged Airborne Electronics Equipment

The package which contains the electronic equipment often serves as an effective radiator for transferring heat by radiation and natural convection.

In this example, the heat that can be transferred from an airborne electronics equipment will be calculated under the following conditions.

Temperature of box surface	50°C
Temperature of surroundings	20°C
Altitude	30,000 ft
Dimensions	5 in. × 5 in. × 8 in. long

The first step is to calculate the heat transferred by radiation, using Equation 3.1 and Figure 3.3. From Figure 3.3B, when the package surface

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is at 50°C, the radiated power is .14 W/in^2. The box should be painted to increase its emissivity to .9.

The total surface area is:

$$2 \text{ sides} + \text{top} + \text{bottom} + 2 \text{ ends} = 2 \times 5 \text{ in.} \times 8 \text{ in.} + 5 \text{ in.} \times 8 \text{ in.} + 5 \text{ in.} \times 8 \text{ in.} + 2 \times 5 \text{ in.} \times 5 \text{ in.} = 210 \text{ in}^2.$$

Using Equation 3.1:

$$\begin{aligned} Q &= Q\,\varepsilon RS \\ &= .14 \text{ W/in}^2 \times .9 \times 1 \times 210 \text{ in}^2 \\ &= 26.5 \text{ W}. \end{aligned}$$

The next step is to calculate the heat transferred by natural convection using Equation 3.2 and Figures 3.6, 3.7, and 3.8.

The temperature difference between the package surface and the surroundings is 50°C − 20°C = 30°C. From Figure 3.6, the heat transferred by convection is .15 W/in^2 for a 1 in. vertical surface at sea level.

The effective length along which air moves is 5 in. for the vertical sides and ends of the package. From Figure 3.7, the relative effectiveness factor due to surface dimensions and orientation is .67.

The effective length for the top surface is (5 in. × 8 in.)/(5 in. + 8 in.) = 3 in. and from Figure 3.7, $R_1 = .76$. For the bottom surface, $R_1 = .38$.

The effect of 30,000 ft altitude operation is given by the altitude reduction factor R_2, which is .55 from Figure 3.8 at an altitude of 30,000 ft. The power transferred by natural convection from each surface should now be calculated using Equation 3.2.

For the top surface:

$$\begin{aligned} Q &= Q_c F_1 F_2 S \\ &= .15 \text{ W/in}^2 \times .76 \times .55 \times 5 \text{ in.} \times 8 \text{ in.} \\ &= 2.5 \text{ W}. \end{aligned}$$

Calculations made in the same manner for the other surfaces give the following results.

Surface	Q_c (W/in^2)	F_1	F_2	Area (in^2)	Power (W)
Top	.15	.76	.55	5 × 8	2.5
Bottom	.15	.38	.55	5 × 8	1.3
2 Sides	.15	.67	.55	2(5 × 8)	4.4
2 Ends	.15	.67	.55	2(5 × 5)	2.8
				TOTAL	11.0

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The total heat transferred from the package at 50°C is therefore:

Radiation	26.5 W
Convection	11.0 W
	37.5 W

The components inside the package must, of course, be hotter than the package surface to be able to transfer their heat to the package. The heat may be transferred from the components to the package walls by conduction, radiation and natural convection, or any of the heat transfer means. In Example 2 of Chapter 6, the transfer of heat from the components to the package walls by evaporation will be discussed.

3.5 REFERENCES

Useful references for cooling of electronic equipment by radiation and natural convection are as follows:

1. Krauss, A. D., *Cooling Electronic Equipment*, Prentice Hall, Inc., Englewood Cliffs, N.J., 1954, Chapter 2, pp. 43–51, Chapter 3, pp. 52–69.
2. *Guide Manual of Cooling Methods for Electronic Equipment*, NAVSHIPS 900,190, U.S. Government Printing Office, Washington, D.C., pp. 28–39.
3. McAdams, W. H., *Heat Transmission*, McGraw-Hill, New York, 1954, Chapter 4, pp. 55–82, Chapter 7, pp. 165–183.
4. Goldman, W., *A Primer on Heat Sinking*, Wakefield Engineering, Inc., Wakefield, Mass., 1966.
5. *Heat Sinks*, Thermalloy, Inc., Dallas, Texas, 1972.

Reference 1 provides the basic equations from which the formulas and graphs of this Chapter are derived. Chapter 2 covers natural convection. Chapter 3 covers radiation.

Reference 2 presents the same information as Reference 1, and, in addition, presents some particularly useful nomographs for calculation.

Reference 3 provides the same basic equations of References 1 and 2.

Reference 4 provides practical rules for the use of radiation and natural convection cooled heat sinks.

Reference 5 provides application information and a detailed listing of representative heat sinks for electronic equipment.

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4

Forced Air Cooling

An order of magnitude increase in heat transfer can be achieved by blowing air over the electronic component, rather than relying on radiation and natural convection. The price that must be paid for this increased cooling is:

- Increased system complexity, because a fan and its associated equipment (such as ducting, dust filters, and interlocks) are required to force the air over the component.
- Reduced electrical efficiency for the system, because the fan requires electrical power.
- Increased vibration and acoustical noise.

Obviously, heat transfer by radiation and natural convection should be used wherever possible, to avoid the increased system complexity of forced air cooling. However, when the heat that must be transferred is greater than 50 W, the size of the fins required for radiation and natural convection becomes excessive, and forced air cooling with all of its complexity often provides the best system design.

Although more complicated than radiation and natural convection, forced air cooling is certainly simpler than forced liquid cooling, because a supply of cooling air is readily available and air does not have the freezing, boiling, or dripping problems of liquids.

The design of forced air cooling poses two problems:

1. Design of the cooling fins on the electronic component.
2. Choice of the fan or blower.

These two problems must be solved jointly. The amount of air flow that a particular fan can provide is determined by the pressure into which the fan must work. Both the amount of heat transfer that can be obtained from forced air cooling and the pressure required to force air through the cooling

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fins depends on air flow and fin geometry. Consequently, the fin design must be made in conjunction with the choice of fan. This tradeoff is illustrated in Section 4.1, where a typical forced air cooling design is described.

Design information for calculating the temperature rise of the component with forced air cooling and the pressure drop required to force the air through the fins is presented in Sections 4.2 through 4.5. The temperature rise of the cooling air itself, which is independent of fin size or geometry, is discussed in Section 4.2. Information for designing air cooling fins is given in Section 4.3. Cooling of electronic components in racks and cabinets is described in Section 4.4.

The effect of altitude and air temperatures different from the standard sea level and room temperature environment is considered in Section 4.5.

Fans for the forced air cooling of electronic equipment are discussed in Section 4.6. This discussion includes:

1. The pressure versus airflow characteristics of the fans.
2. The size, electrical power, acoustic noise, life, and cost of the fans.
3. Auxiliary equipment such as dust filters, ducting, and interlocks.

Examples of forced air cooling designs of electronic equipment are presented in Section 4.7 to illustrate the use of the equations, figures and fan specifications of this chapter.

4.1 A TYPICAL FORCED AIR COOLING DESIGN

Figure 4.1 shows a typical forced air cooled heat sink. The conduction of heat from the electronic component to the fins has been discussed in detail in Chapter 2. A fan blows air through the fins, and the heat is transferred from the fins to the air stream. The temperature of the air rises as it absorbs heat, and the fin temperature rises to be hotter than the air in order to transfer heat.

The temperature rise of the fins above the inlet air temperature is shown in Figure 4.2 for the particular heat sink shown in Figure 4.1. This curve shows the measured temperature rise at the base of the fins per watt of power transferred as a function of air flow rate in cubic feet per minute (CFM).

At a flow rate of 18 CFM, Figure 4.2 shows that the temperature rise of the fins above the inlet air temperature is .5°C/W. If the inlet air from the surroundings is at 20°C and the fins are at 100°C, the temperature difference is 80°C and 160 W can be transferred.

If the air flow through the fins is increased to 45 CFM, the heat transfer rate doubles. With an 80°C temperature difference between the fins and the inlet air, 320 W can be transferred. This is significantly more heat than can be transferred by radiation and natural convection from a heat sink of the same size.

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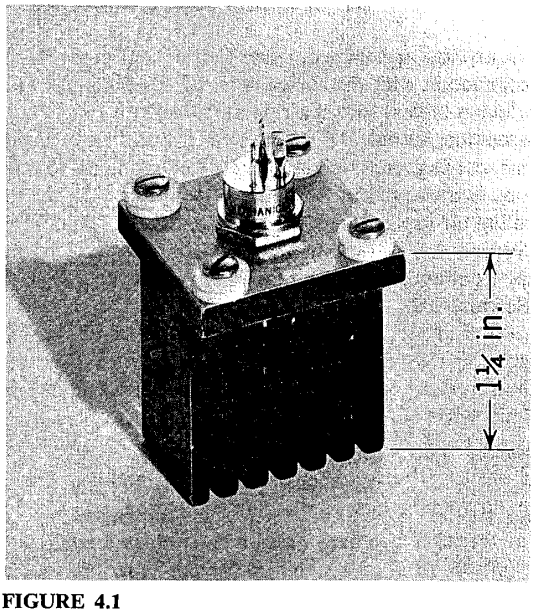

FIGURE 4.1
Forced air cooled heat sink (Photo courtesy of Wakefield Engineering, Inc.)

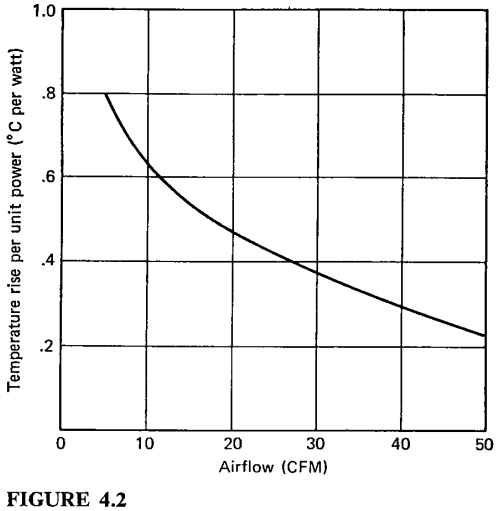

FIGURE 4.2
Temperature rise of forced air cooled heat sink as a function of airflow

For example, the heat sink discussed in Section 3.1 of Chapter 3 and shown in Figure 3.1, which is designed for cooling by radiation and natural convection, occupies three times the volume of the forced air cooled heat sink and fan combination. At the same temperature difference of 80°C, it can transfer only 60 W, or less than one-fifth the power of the forced air cooled sink.

The foregoing discussion suggests that as great an air flow as possible should be used to maximize the heat transfer capability of the fins. However, the greater the air flow, the greater the pressure required to force the air through the fins. Pressure as a function of air flow is shown in Figure 4.3

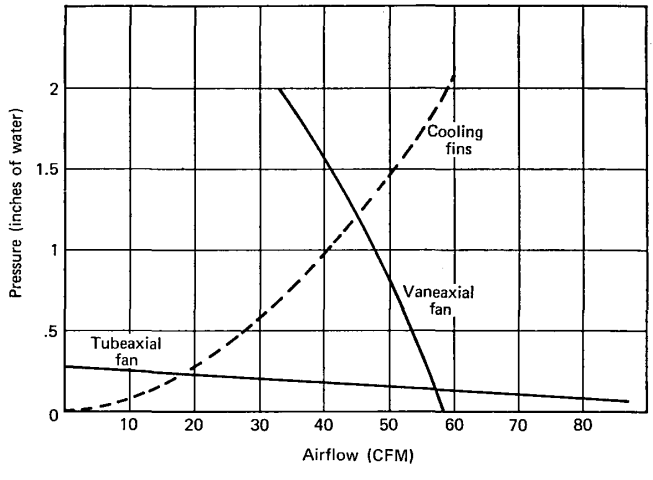

FIGURE 4.3
Fan characteristics and pressure required to force air through the heat sink as a function of airflow

for the particular heat sink shown in Figure 4.1. Pressure is expressed in "inches of water," which is the height of a column of water that the pressure will support. This choice of unit is used because fan pressure capability is commonly expressed in this unit, and because pressure is easily measured with a water column manometer (as described in Chapter 10).

As Figure 4.3 shows, at an air flow of 18 CFM, the pressure required to force the air through the cooling fins is only .2 in. of water. The required pressure increases rapidly with flow rate. At 45 CFM, the required pressure is 1.2 in. of water.

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Pressure versus air flow characteristics of two fans which are commonly used in electronic equipment are superimposed over the fin characteristics of Figure 4.3. These particular fans are described in detail in Section 4.6, and are illustrated in Figures 4.13 and 4.14. The intersection of the pressure versus flow characteristics of the cooling fins with the pressure versus flow characteristics of the fans determines the actual air flow through the fins and thereby determines their heat transfer capability.

The higher pressure capabilities of the vaneaxial fan allow it to force 45 CFM of air through the cooling fins, and with this air flow the fin assembly has a heat transfer capability of .25°C/W.

In contrast, the tubeaxial fan has a much lower pressure capability and can force only 18 CFM through the cooling fins. Therefore, when the same fin assembly is used with the tubeaxial fan, its heat transfer capability is only one-half as good as when it is used with the vaneaxial fan.

From a consideration of only pressure versus air flow capabilities, the higher pressure vaneaxial fan would be the best choice. However, the choice of cooling fan is dictated by other requirements besides pressure versus air flow characteristics, namely:

1. Size and weight.
2. Power consumption.
3. Acoustical noise.
4. Life.
5. Cost.

As will be discussed in detail in Section 4.6, the vaneaxial fan has a better pressure versus air flow characteristic and is smaller in size and weight, but the tubeaxial fan offers the advantages of less power consumption, lower acoustical noise, longer life, and lower cost.

The effectiveness of the various methods of heat transfer were compared in Figure 1.2 of Chapter 1. The value shown for forced air cooling was taken from the cooling fin design shown in Figure 4.1. At an air flow of 45 CFM, which can be supplied by the vaneaxial fan, the temperature rise of the fins is .25°C/W. The total surface area of the fins is 13 ducts × 2 sides/duct × 1.5 in. × 1 in. = 40 in^2. If the fin temperature is 100°C and the inlet air temperature is 20°C, the heat transferred is 8 W/in^2.

Air cooling fins can be spaced much closer together than fins for radiation and natural convection cooling because there is no shielding problem. Consequently, forced air cooling designs can achieve much more fin surface in a given volume as well as greater heat transfer per unit fin area than radiation and natural convection designs.

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4.2 TEMPERATURE RISE OF THE COOLING AIR

The temperature rise of the cooling fins above the inlet air temperature consists of two parts, as shown by the following formula:

$$T\,(\text{fin}) = \Delta T\,(\text{air}) + \Delta T\,(\text{fin-air}) + T\,(\text{surroundings}) \tag{4.1}$$

where:

T (fin)	is the temperature of the cooling fin surface (°C).
ΔT (air)	is the temperature rise of the air as it absorbs heat from the cooling fins (°C).
ΔT (fin-air)	is the temperature rise of the cooling fin above the air in the cooling duct (°C).
T (surroundings)	is the temperature of the inlet air, which is equal to the temperature of the surroundings (°C).

The air temperature increases as the air passes along the fins because the air is absorbing power. This is the term ΔT (air) of Equation 4.1. The fin must be hotter than the cooling air to transfer heat. This is the term ΔT (fin-air).

The temperature rise of the air as it absorbs heat from the cooling fins is independent of the geometry or dimensions of the fins. The air temperature rise depends only on the power that is transferred and the total air flow rate, and is given by the following formula:

$$\frac{\Delta T\,(\text{air})}{Q} = \frac{1.73}{f} \tag{4.2}$$

where:

ΔT (air)	is the temperature rise of the air in absorbing heat from the cooling fins (°C).
Q	is the total power that is being transferred (watts).
f	is the total air flow through the fins (CFM).

Equation 4.2 applies only to air at sea level at a temperature of 20°C. Modifications of the constant of the equation to include other altitudes and air temperatures are discussed in detail in Section 4.5.

Equation 4.2 is illustrated graphically in Figure 4.4, where the temperature rise of the cooling air is shown as a function of the power that is being transferred with air flow as a parameter. Note that with an air flow of 50 CFM, 500 W of power can be transferred with an air temperature rise of only 17°C. As will be discussed in Section 4.6, fans are available which can provide air flows as high as 500 CFM. With this air flow rate, 10 kW of heat can be transferred with an air temperature rise of only 35°C.

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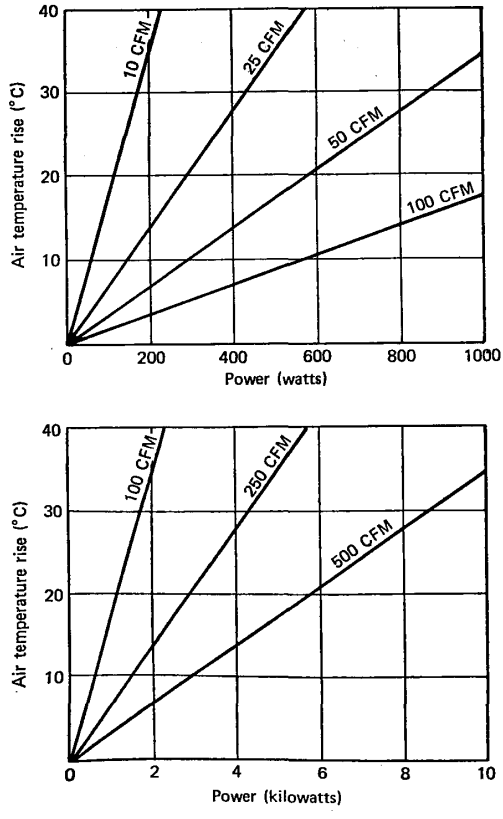

FIGURE 4.4
Temperature rise of cooling air as a function of power absorbed

4.3 DESIGN EQUATIONS FOR AIR COOLING FINS

The design of forced air cooling of electronic equipment has two parts. The cooling fin dimensions must be selected to obtain sufficient cooling and at the same time keep the pressure drop through the fins low enough so that the required air flow can be supplied by a suitable fan. The fan must be chosen not only for its pressure versus air flow characteristics but also to meet other system requirements of size, weight, power consumption, acoustical noise, life and cost.

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In this section, design formulas are presented for estimating the heat transfer capability and the pressure requirements as a function of fin geometry and air flow under room temperature and sea level operating conditions. Modifications to the formulas to account for the effect of varying air density at other temperatures and altitudes are presented in Section 4.5. The problem of fan selection is discussed in Section 4.6.

As shown in Equation 4.1, the temperature rise of the cooling fins above the inlet air temperature consists of two parts—the temperature rise of the air as it absorbs power and the temperature rise of the fins above the air. The temperature rise of the air, as discussed in the previous section, is independent of fin geometry. In any fin design, the fins will get hot enough to transfer the heat being generated in the electronic components to the air stream. However, unless the fins are designed properly, this temperature rise may be excessive.

Fin Temperature Rise

The temperature rise required to transfer a given amount of heat from the fins depends upon the velocity of the air moving past the fins, and the air velocity in turn depends on the fin geometry and dimensions and on the air flow rate. The temperature rise is related to these factors by the following formula:

$$\frac{\Delta T\,(\text{fin-air})}{Q} = \frac{140\,w}{n^{.2} z^{.2} f^{.8} L} \tag{4.3}$$

where:

ΔT (fin-air)	is the temperature rise of the fin surface above the air flowing through the cooling ducts between the fins (°C).
Q	is the total power that is being transferred (watts).
w	is the width of the ducts, i.e. the spacing between fins (inch).
z	is the height of the fins above the base (inch).
L	is the length of the fins along the direction of air flow (inch).
n	is the number of ducts through which the air flows.
f	is the total air flow through all the ducts (CFM).

The geometry of the cooling fins is shown in Figure 4.5, and this figure defines the critical fin dimensions w, z, and L of Equation 4.3.

Forced air flow is a complicated hydrodynamic problem which is affected by many factors. Several simplifying assumptions have been made in deriving Equation 4.3, so it gives only an approximate result. In spite of its approximate nature, the equation is useful in estimating cooling capability and in

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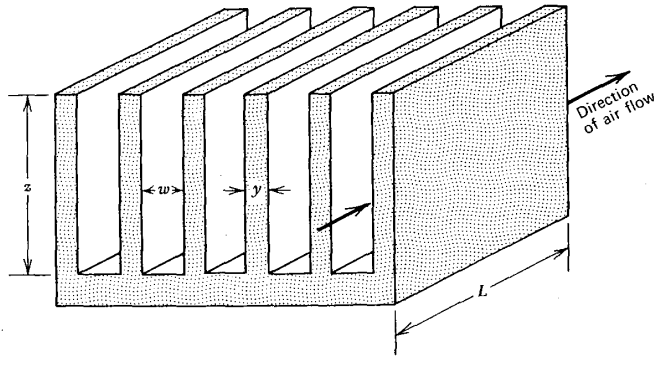

FIGURE 4.5
Geometry of forced air cooled heat sink and important dimensions

deciding how changes in fin dimensions will affect cooling capability. The degree of correlation between Equation 4.3 and measured results is shown in the examples of Section 4.7. Ultimately a cooling design obtained by the use of Equation 4.3 must be built and experimentally evaluated. Suitable measurement techniques for forced air cooling designs are described in detail in Chapter 10.

The major assumptions used in deriving Equation 4.3 are as follows:

1. The average air temperature is 20°C and the altitude is at sea level.
2. The air flow is turbulent.
3. The ducts are narrow compared to their height above the base.

The first assumption of sea level operation and 20°C air affects only the coefficient of the equation. Corrections to this coefficient for other altitudes and air temperatures are presented in Section 4.5.

Is is useful to examine Equation 4.3 to determine the major factors affecting heat transfer. The temperature rise of the fins above the air is directly proportional to the duct width, i.e., the spacing between the fins, so reducing this dimension will correspondingly reduce the temperature difference between the fin surface and the air. In contrast, the temperature rise of the fin is almost independent of duct height.

The temperature rise of the fin is inversely proportional to the length of the fin in the direction of air flow, so, for example, doubling the length of the fin would cut the temperature rise in half.

The fin temperature rise is inversely proportional to the .8 power of air flow. Consequently, to reduce fin temperature rise by a factor of two, the air

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flow rate must be more than doubled. This fact is evident from the experimental results shown in Figure 4.2, where air flow must be increased from 18 CFM to 45 CFM, i.e., 2.5 times, to reduce the temperature rise per watt by a factor of two. The number of ducts that the air flow is divided into has very little effect on heat transfer capability as long as the duct width (i.e., the fin spacing) is constant, since heat transfer capability depends on the number of ducts to the .2 power. The total air flow could all be forced through a single duct or it could be divided up into many ducts, with only a small change in fin temperature rise.

Equation 4.3 assumes a perfectly conducting fin. The fin temperature rise calculated from Equation 4.3 must be corrected to account for the finite thermal conductivity of the fin, as outlined in Section 2.6 of Chapter 2.

Pressure

Unfortunately, the fin geometry and dimensions cannot be selected arbitrarily to minimize temperature rise, because they also affect the pressure required to force air through the fins. The minimizing of the air pressure drop is an important step in reducing fin temperature. As shown by the formulas of the previous paragraphs, the temperature rise of the fins depends on the amount of air flow. However, the amount of air flow that can be forced through the fins by a particular fan depends on the pressure versus air flow characteristics of the design. Therefore, reducing the air pressure drop through the equipment is extremely important.

The total pressure into which the fan must work is determined not only by the cooling fins themselves but by all of the following elements:

- Entry port.
- Dust filter.
- Inlet ducting.
- Cooling fins.
- Outlet ducting.
- Exit port.

Of all of these elements that are in the air flow path, only the cooling fins contribute to cooling, but all contribute to the pressure drop.

An exact calculation of the pressure drops through each element of the system is extremely difficult. However, the application of three general rules allow unnecessary pressure drops to be minimized. These rules are:

1. The pressure drop through any structure is inversely proportional to the square of its cross-sectional area. For example, if the cross-sectional area of a duct is doubled, the pressure drop through it is reduced to one-fourth.

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2. The pressure drop through any element is approximately directly proportional to the air flow squared. This fact is evident from the pressure versus air flow characteristics shown in Figure 4.3. At 18 CFM the pressure drop is approximately .2 in. of water. When the air flow is doubled to 36 CFM, the pressure drop increases to .8 in. of water, a four times increase.
3. The pressure drop through any structure is proportional to the length of the structure (except that for short structures the fixed entry and exit pressure drops may be much larger than the pressure drop due to air flow through the structure).

Applying the first of the above rules, the cross-sectional area of the entry port, inlet ducting, outlet ducting, and the exit port should be made large enough and without sharp bends or abrupt transitions so that the pressure drop through all of these elements (which do not contribute to cooling) can be negligible. This first step must be taken in any forced air cooling design.

In a well designed equipment, the major part of the pressure drop will be the cooling fins themselves. The geometry and dimensions of the fins affect both the heat transfer and the pressure drop, and a compromise design between the two must often be used. An approximate formula for estimating the pressure drop through cooling fins as a function of their dimensions is as follows:

$$\Delta p = \frac{\left(\frac{f}{n}\right)^2}{(wz)^2}\left[1 + .01\,\frac{L}{w}\right] \times 10^{-3} \tag{4.4}$$

where:

Δp is the pressure drop through the fins (inches of water).
All other terms are as defined for Equation 4.3.

Equation 4.4 is based on the same assumptions that were used for Equation 4.3, that is, sea level operation with 20°C inlet air and turbulent flow. A multiplying constant which permits Equation 4.4 to be used at other altitudes and inlet air temperatures is discussed in Section 4.5.

As would be expected, Equation 4.4 illustrates the three general rules for air pressure drop stated above. The pressure drop is inversely proportional to the square of the cross-sectional area of the ducts (which is wz) and directly proportional to the square of the air flow rate per duct. The cross-sectional area of the ducts cannot be made arbitrarily large to reduce pressure drop because duct dimensions affect both heat transfer as well as pressure. Note, however, by comparing Equations 4.3 and 4.4, that the fin temperature depends directly on duct width but only slightly on duct height, whereas pressure drop is equally affected by both. Therefore, fin height can be

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increased to reduce the pressure drop without significantly degrading the heat transfer capability of the fins.

Note also that although the fin temperature rise is fairly insensitive to the number of fins used, the pressure drop is very sensitive, varying inversely as the square of the number of fins.

The first term in the brackets accounts for the transition into and out of the fin structure. The second term accounts for the pressure drop through the fins themselves. Note that until the length of the fins is 100 times their spacing, the pressure drop due to the transitions into and out of the finned structure is greater than the pressure drop due to the air flowing through the fins.

Certain additional simplifying assumptions have been used for the terms in the bracket of Equation 4.4. The first term is actually a function of the ratio of fin thickness to duct width, but a fixed value for the case when the thickness to width ratio is one-third has been used. The second term depends slightly on flow rate and this variation has been ignored. The principal use of Equation 4.4 should be in obtaining a rough estimate of pressure drop through the fins and in determining how changes in fin dimensions will affect the pressure drop. In any actual equipment, the pressure drop must be measured using the techniques described in Chapter 10.

4.4 FORCED AIR COOLING OF RACKS AND CABINETS

A common means of cooling electronic equipment is by blowing air through a rack or cabinet to cool all of the electronic components mounted in the cabinet. Figure 4.6 shows a typical electronic equipment cabinet with a blower mounted in the bottom. The air flow is shown schematically.

The air temperature increases as it absorbs power from the components, and therefore the air is the hottest at the top of the cabinet. The temperature rise of the exit air is independent of component shapes, sizes or locations and depends only on the total power absorbed from the components and the total air flow rate. This air temperature rise can be calculated from Equation 4.2, which was shown graphically in Figure 4.4.

The electronic components must, of course, be hotter than the cooling air in order to transfer heat to the air. Because the air temperature increases as it circulates through the cabinet, the components near the top of the cabinet will be hotter than the components near the bottom of the cabinet even if their power dissipation is the same. Enough air should therefore be blown through the cabinet to limit the air temperature rise to about 25°C. From Equation 4.2, if a total power of 1000 W is dissipated in the cabinet, the air flow should be 70 CFM.

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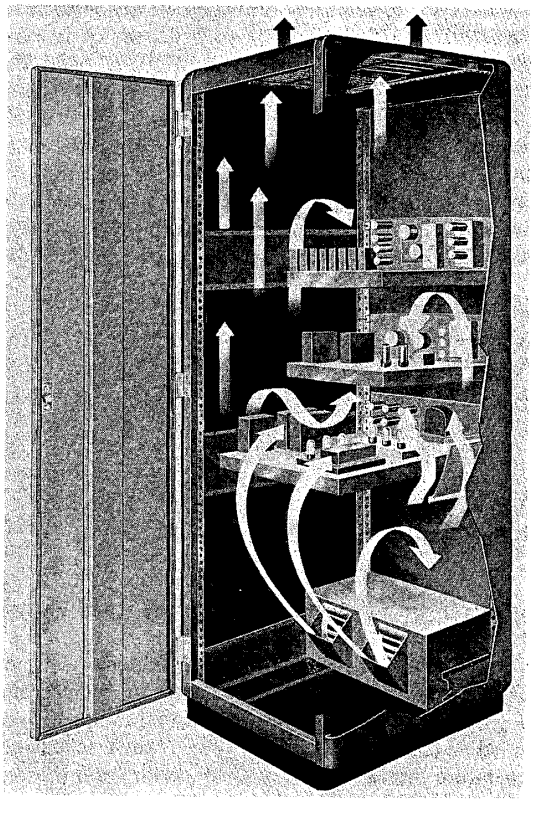

FIGURE 4.6
Airflow in a forced air cooled rack of electronics equipment (Photo courtesy of McLean Engineering Laboratories)

The amount of air that can be blown through the cabinet depends on the pressure versus air flow characteristics of the cabinet and the pressure versus air flow capabilities of the fan. A simple formula is not available for calculating the air pressure required to flush a given amount of air through an electronics cabinet because of the complicated nature of the air flow. However, an estimate can be made from the empirical data presented in Figure 4.7, which shows pressure versus air flow characteristics for five typical electronics racks and cabinets.

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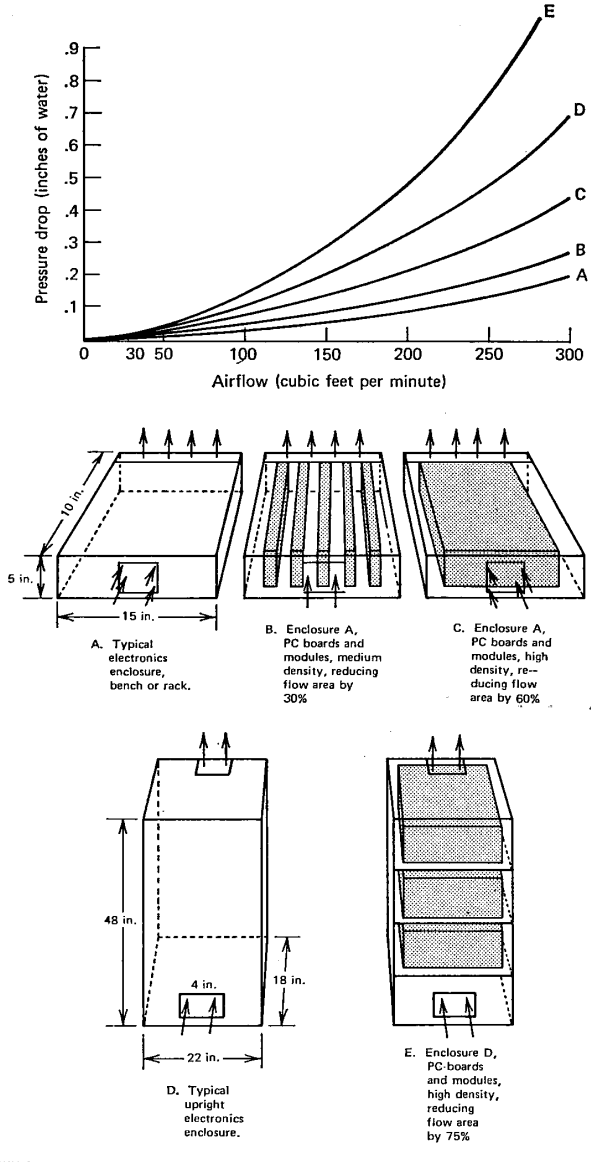

FIGURE 4.7
Pressure drop as a function of airflow for typical electronic enclosures (Courtesy of Pamotor, Inc.)

Likewise, a simple calculation of the temperature rise of each component as it transfers its heat to the air stream is not possible because the air flow is so complex. However, an estimate of the temperature rise can be obtained by approximating the actual situation for each component by the idealized geometry shown in Figure 4.8.

In this idealized case, the electronic component is assumed to be in the shape of a circular cylinder of diameter d and is located in a duct with a cross section A which is much larger than the component diameter. Figure 4.8 shows the amount of power transferred per degree of temperature difference between the component surface and the air stream as a function of the air flow rate and the component and duct dimensions. The curve of Figure 4.8 is for a 1 in. high cylinder, but it can be directly applied to components of other heights because the power that is transferred is directly proportional to cylinder height. The data can be applied to components of any shape by approximating their actual shape by an appropriate circular cylinder. The amount of heat transfer depends not only on the dimensions of the component being cooled but also on the velocity of the cooling air, which

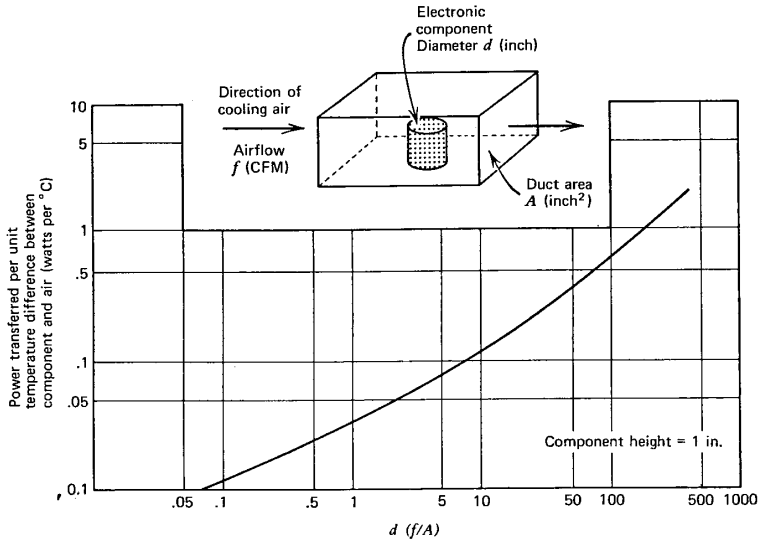

FIGURE 4.8
Power transferred from a cylindrical component in an airstream

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is determined by the volume of air flow and the cross-sectional area of the duct in the cabinet through which the air is blown.

The use of Figure 4.8 is illustrated by the following example. Assume the component to be cooled is an electron tube with a diameter of 3/4 in. and a height of 2 in. An air flow of 25 CFM is blown through a 5 in. × 5 in. duct in which the tube is located. $d(f/A) = 3/4$ in. × 25 CFM/5 in. × 5 in. = .75 and from Figure 4.8:

$$\frac{Q}{\Delta T} \text{ is } .03 \text{ W/°C.}$$

Since the cylinder is 2 in. high and Figure 4.8 is for a 1 in. high cylinder, twice as much power will be transferred, that is, .06 W/°C. If the air temperature is 20°C and the tube surface is at 100°C, the temperature difference is 80°C, and approximately 5 W of power will be transferred.

This type of air cooling is less effective than if the component were mounted on a finned heat sink such as was shown in Figure 4.1, but it is more effective than if only natural convection were used.

For comparison, the amount of heat transferred from the tube by natural convection can be calculated from the design information of Chapter 3. When the tube surface is at 100°C and the surroundings are at 20°C, .4 W/in^2 will be transferred by natural convection from a 2 in. vertical surface. The area of the tube surface is 5 in^2, so the total power transferred by natural convection is 2 W or only 40% as much as when air is flushed past the tube. Note, however, that if the air flow were less, or if the tube were located in a larger duct with the same volume of air flow, the cooling that could be obtained would be less.

In most practical cases, the cooling that can be obtained for any individual component by blowing air through a rack or cabinet is only slightly better than can be obtained by natural convection. Any attempt to duct the air around the components will raise the air pressure requirements so that a reasonable fan cannot blow enough total air through the cabinet. A good general rule therefore is to design each chassis so that all components on it are satisfactorily cooled by conduction, radiation, and natural convection, and depend on the fan only to bring sufficient air into the cabinet. In other words, the individual chassis should be designed to operate on the bench without forced air cooling. The fan is then used to insure that enough air is provided so that several chassis can operate together inside the cabinet.

Air should always be blown into the cabinet, as shown in Figure 4.6, rather than sucked in, to keep dust out of the cabinet. A dust filter should be mounted at the fan inlet.

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4.5 THE EFFECT OF ALTITUDE AND INLET AIR TEMPERATURE

The design equations and curves given in the previous sections were all for the case of sea level operation with an inlet air temperature of 20°C. At other altitudes and air temperatures, the amount of cooling and the pressure drop obtained with a given volume of air flow will be different, because the air density and the thermal properties of air are different.

For Equations 4.2 through 4.4, which give the temperature rise of the cooling air as it absorbs power, the temperature rise of the cooling fins above

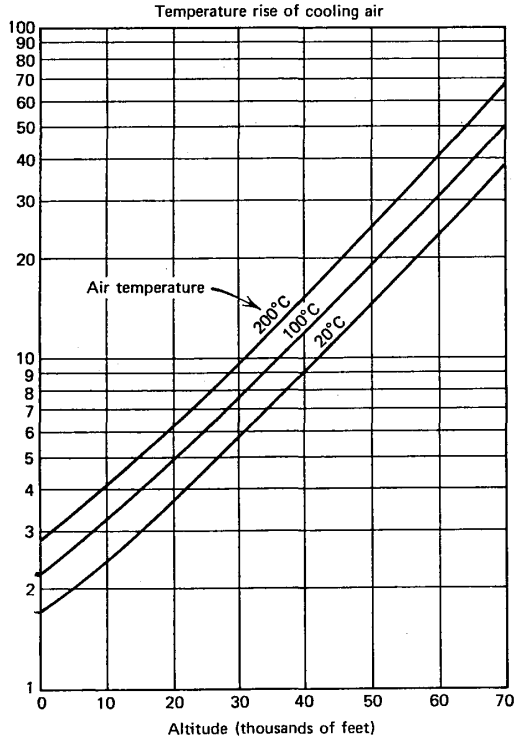

FIGURE 4.9A
Coefficient of Equation 4.2 as a function of altitude and air temperature

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the air, and the pressure drop through the fins, respectively, the value of air density and the other air thermal properties affect only the constant multiplying factor of the equations. Consequently, Equations 4.2 through 4.4 can be used at any altitude or air inlet temperature simply by using a different constant. Appropriate values of these constants as a function of altitude, with inlet air temperature as a parameter, are shown in Figure 4.9. The value of these constants at sea level and an inlet air temperature of 20°C from Figure 4.9 is, of course, the same as given with the equations in the previous section.

As altitude and inlet air temperature increase, the temperature rise of both the cooling air and the fins increases. For example, when altitude increases to 35,000 ft, the temperature rise of a given volume flow of cooling air increases four times over the temperature rise that occurs at sea level. The temperature rise of the fins above the cooling air increases three times.

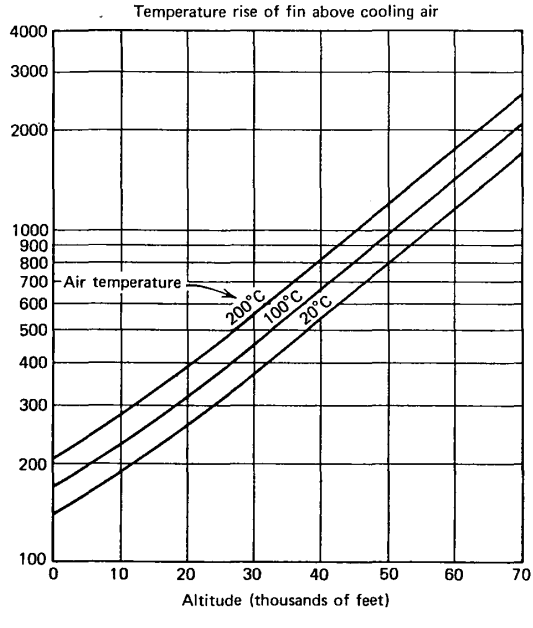

FIGURE 4.9B
Coefficient of Equation 4.3 as a function of altitude and air temperature

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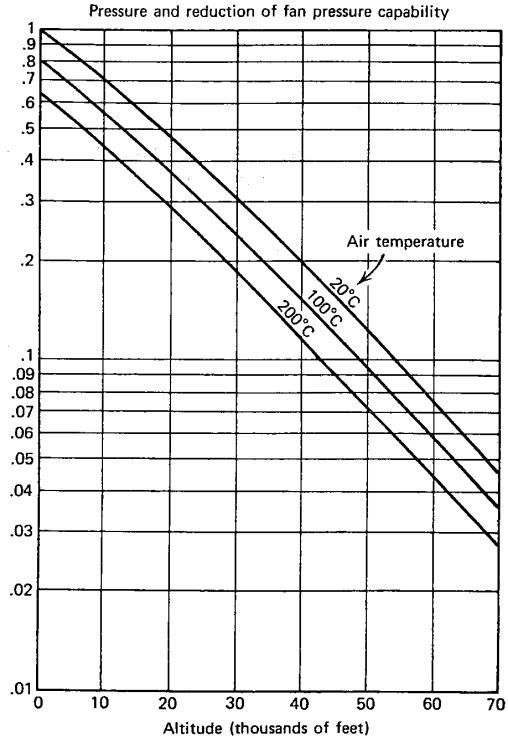

FIGURE 4.9C
Coefficient of Equation 4.4 as a function of altitude and air temperature

The pressure required to force a given volume of air through the cooling fins decreases with increasing altitude and inlet air temperature, as shown by the curves of Figure 4.9. However, the pressure versus air flow capability of most fans decreases by the same ratio with increasing altitude and inlet air temperature, with the result that the air flow that a particular fan can blow through a particular fin assembly is approximately constant at any altitude or inlet air temperature.

The effect of altitude on the performance of the forced air cooled heat sink described in Section 4.1 is shown in Figure 4.10. The temperature rise of the fins and the pressure required to force air through them is shown as a function

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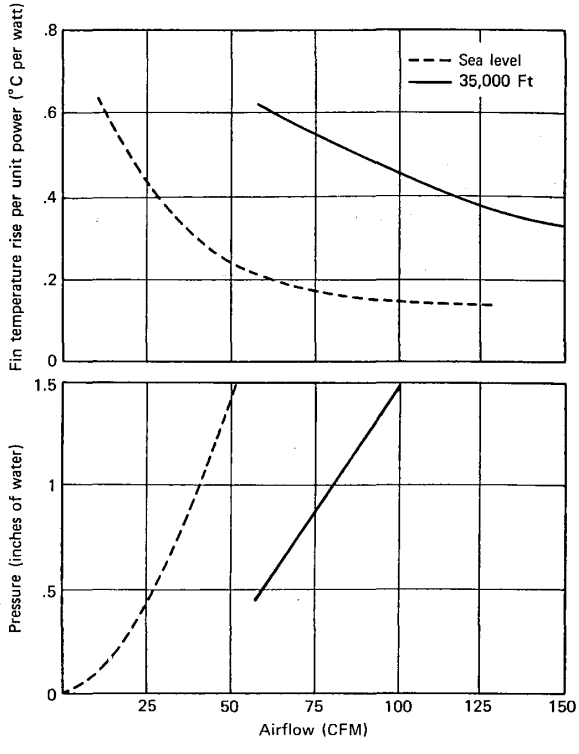

FIGURE 4.10
A comparison of cooling fin temperature and pressure at sea level and at an altitude of 35,000 ft

of air flow at both sea level and at an altitude of 35,000 ft. (The sea level curves are the same as were shown previously in Figures 4.2 and 4.3.) As expected, the fin temperature rise is greater at 35,000 ft than at sea level for a given air flow and the pressure required to force air through the fins is less.

Estimating High Altitude Performance from Sea Level Measurements

The equations presented in Sections 4.2 and 4.3 for temperature rise and pressure give only approximate answers. Ultimately a forced air cooling design obtained by the use of these equations must be built and experimentally measured. The experimental measurements at sea level and with

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room temperature inlet air are easy to make, and are described in detail in Chapter 10.

These same measurements are much more difficult to make for high altitude conditions and for inlet air temperatures different from room temperature. For these cases a special temperature-altitude chamber must be used.

Fortunately, this experimental complication can be avoided by measuring the performance of the forced air cooling design at sea level with room temperature air and then extrapolating the measured results to the desired altitude and inlet air temperature using the correction factors of Figure 4.9. The correction factors involve only the density and thermal properties of air and these are well established as a function of altitude and air temperature. The effect of the air flow pattern on cooling and pressure drop, which can only be approximately calculated from the geometry and dimensions of the fins, can be exactly determined from the sea level, room temperature measurements.

The performance of the forced air cooled heat sink described in Section 4.1 at an altitude of 35,000 ft, which is shown in Figure 4.10, was deduced in just this way. For example, the cooling fin temperature rise at 60 CFM at sea level was measured to be .2°C/W. From Figure 4.9, operation at 35,000 ft would increase the fin temperature by 450/140 = 3 times, so the fin temperature would be .6°/W. Similarly, the measured sea level pressure at 60 CFM is 2 in. of water. From Figure 4.9, this pressure would be reduced to .25 × 2 in. of water or .5 in. of water at an altitude of 35,000 ft.

One caution should be exercised in using Figure 4.9 to extrapolate measured results to other altitudes and inlet air temperatures. The extrapolation assumes that the air flow remains turbulent. Greater air flow is required for turbulent flow at 35,000 ft than at sea level. For this reason, the extrapolation for 35,000 ft operation has not been extended to low air flow rates in Figure 4.10.

Fan Selection for High Altitude Operation

The pressure that a particular fan can supply at a given air flow is proportional to the air density and so decreases with increasing altitude and air inlet temperature as shown in Figure 4.9.

Fan pressure versus air flow characteristics are normally supplied by fan manufacturers for sea level and room temperature operating conditions. The performance of a fan at any altitude and air inlet temperature may be extrapolated from the manufacturer's data by multiplying the pressure at each flow rate by the fan pressure reduction factor shown in Figure 4.9. The use

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of this approach is illustrated in Figure 4.11, which shows the pressure versus air flow characteristics of the finned heat sink described in Section 4.1 and the pressure versus air flow characteristics of a 3 in. diameter vaneaxial fan at an altitude of 35,000 ft and at sea level. The pressure versus air flow characteristics of the finned heat sink at 35,000 ft were shown previously in Figure 4.10 and were deduced from sea level measurements as described in the previous paragraph. The problem is to select an appropriate fan for use with these fins at 35,000 ft. The dotted curve shows the sea level characteristics of the 3 in. diameter vaneaxial fan and is taken from the fan manufacturers published data. The performance of this fan at 35,000 ft is shown by the solid curve and is obtained by multiplying its sea level pressure capability at each flow rate by the appropriate reduction factor from Figure 4.9. As Figure 4.11 shows, at 35,000 ft the pressure versus air flow characteristics of the fan and the cooling fins intersect at an air flow of 70 CFM. This is the air flow at which the combination will operate at 35,000 ft, and from Figure 4.10 the temperature rise of the fins will be .57°C/W.

Airborne electronics equipment must operate over a range of altitudes from sea level to the maximum altitude capability of the aircraft. The most difficult cooling problem usually occurs at the maximum altitude, and so the cooling fins must be designed and the fan selected for this altitude as described in this section. At lower altitudes the forced air cooling system will be overdesigned, and the fan will consume an excessive amount of power, because the power requirements of a fan are approximately proportional to

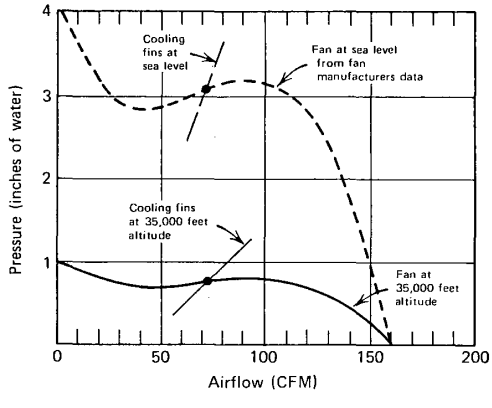

FIGURE 4.11
A comparison of fan characteristics at sea level and at an altitude of 35,000 ft

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its pressure-air flow rate product. For these variable altitude applications, special fans are available which reduce their rotation speeds and thereby their air flow rate at low altitude, where sufficient cooling is obtained with low air flows. In this way the power requirements of the fan are kept reasonably constant with altitude.

4.6 FANS FOR COOLING ELECTRONIC EQUIPMENT

The selection of the proper fan for a particular electronic cooling application is as critical as the design of the cooling fins themselves. The amount of cooling that can be obtained depends on the amount of air that can be forced through the fins, and the amount of air is determined by the pressure versus airflow characteristics of both the cooling fins and the fan or blower. Therefore, the choice of fan and the cooling fin design must be done jointly.

The four types of fans that are commonly used for cooling electronic equipment are illustrated in Figures 4.12 through 4.15.

Figure 4.12 shows a "propeller fan." This type of fan provides high air flow volume at low pressures and is best suited for flushing operations in which a large volume of air is blown through a rack or cabinet to directly cool all of the components. The free delivery (i.e., the maximum air flow at zero pressure) that can be obtained from propeller fans depends on the propeller diameter and varies from about 25 CFM for a 3 1/2 in. diameter propeller up to about 1000 CFM for a 10 in. diameter propeller. The particular model shown in Figure 4.12 has a 6 1/2 in. diameter propeller and as shown provides a free delivery of 160 CFM. The maximum pressure that can be developed by these fans with any propeller diameter is only about .5 in. of water.

Figure 4.13 shows a "tubeaxial fan." Its motor is mounted in the fan hub. As a consequence, a tubeaxial fan can develop better pressure and occupies much less axial length than a propeller fan of the same diameter. The fan illustrated in Figure 4.13 is widely used for cooling commercial electronic equipment. It weighs only 1 lb, is quiet, has an expected operating life without servicing of 10 yr, and is low in cost. To make use of all these advantages, the cooling fins that are used with the fan must be carefully designed because the fan has a pressure capability of only about .1 in. of water at high flow rates. A design of cooling fins to optimally use this particular tubeaxial fan is given in Example 2 of Section 4.7.

Figure 4.14 shows a "vaneaxial fan" and pressure versus air flow curves for a 2 in. diameter model and for a 3 in. diameter model. These vaneaxial fans operate at high rotation speeds and consequently have high pressure capabilities and very small sizes.

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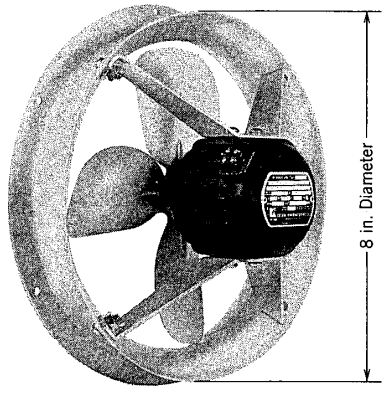


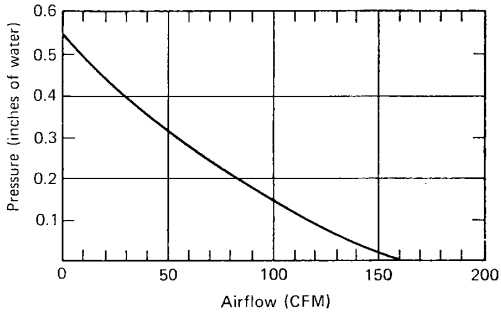


FIGURE 4.12
Propeller fan and its pressure—airflow characteristics (Photo courtesy of Rotron, Inc.)

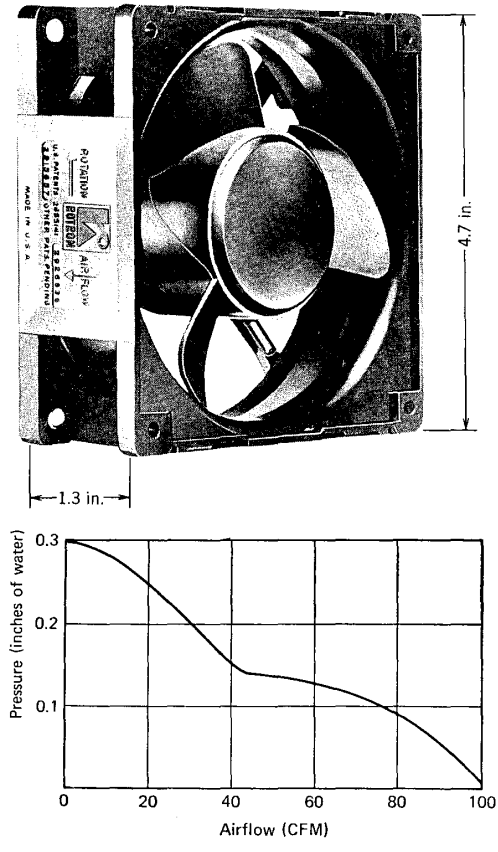
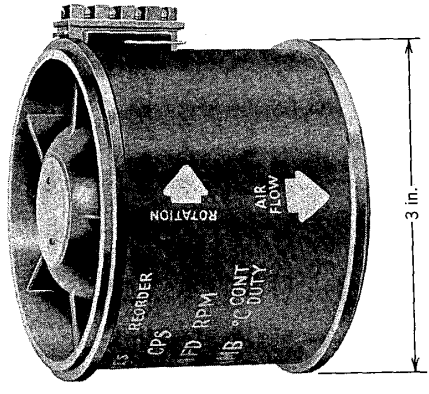


FIGURE 4.13
Tubeaxial fan and its pressure—airflow characteristics (Photo courtesy of Rotron, Inc.)



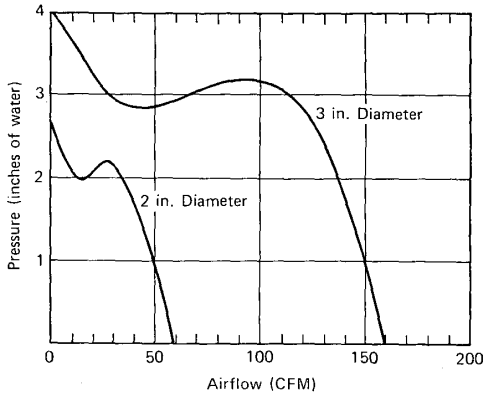


FIGURE 4.14
Vaneaxial fan and pressure—airflow characteristics for two different fan diameters (Photo courtesy of Rotron, Inc.)

At an air flow of 100 CFM, the 3 in. diameter vaneaxial fan has more than ten times the pressure capability of the propeller fan of Figure 4.12 and yet occupies less than one-tenth the volume. This high performance capability is obtained at the expense of high acoustic noise, short life, and the need for 400 cycle power.

The air cooling fins described in Section 4.1 were designed to use the 2 in. diameter vaneaxial fan at sea level and the 3 in. diameter fan at an altitude

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of 35,000 ft. The performance of the 3 in. diameter vaneaxial fan was compared at sea level and at an altitude of 35,000 ft in Figure 4.11.

Figure 4.15 shows a "squirrel cage centrifugal blower." This blower provides high air flow and high pressure capabilities at low rotation speeds. Squirrel cage blowers with wheel diameters from 1.5 in. to 7.5 in. provide free deliveries of 10 to 700 CFM and operating pressures of 1 to 2 in. of water. The particular squirrel cage blower shown in Figure 4.15 has a 5 in. diameter wheel, and as shown, provides 160 CFM at free delivery. A design of cooling fins on a high power tube to match this blower is given in Example 4 of Section 4.7.

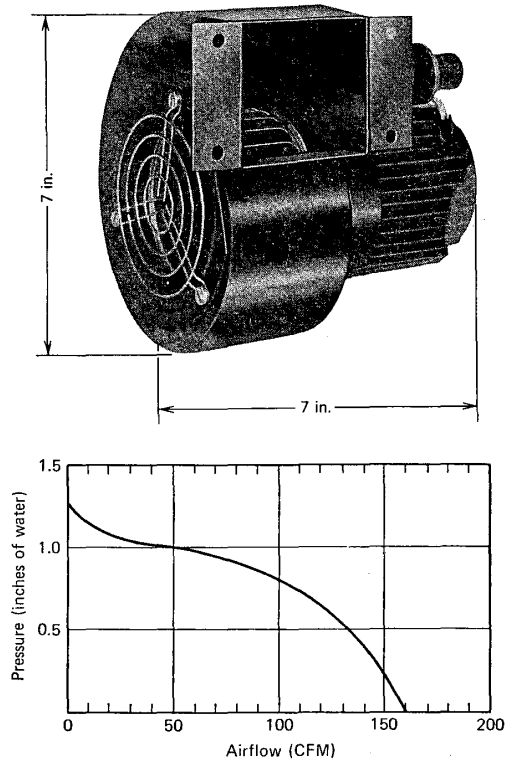

FIGURE 4.15
Squirrel cage blower and its pressure—airflow characteristics (Photo courtesy of Rotron, Inc.)

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Selection of the Most Appropriate Fan for a Particular Electronic Cooling Application

The following factors must be considered in selecting the most appropriate fan for a particular electronic cooling application:

1. Pressure versus air flow characteristics.
2. Size and weight.
3. Power requirements.
4. Acoustic noise.
5. Life.
6. Cost.

To give some insight into the tradeoffs between these factors, the pressure versus airflow characteristics of the four fans shown in Figures 4.12 through 4.15 are compared in Figure 4.16, and the other important factors for fan selection are tabulated in Table 4.1.

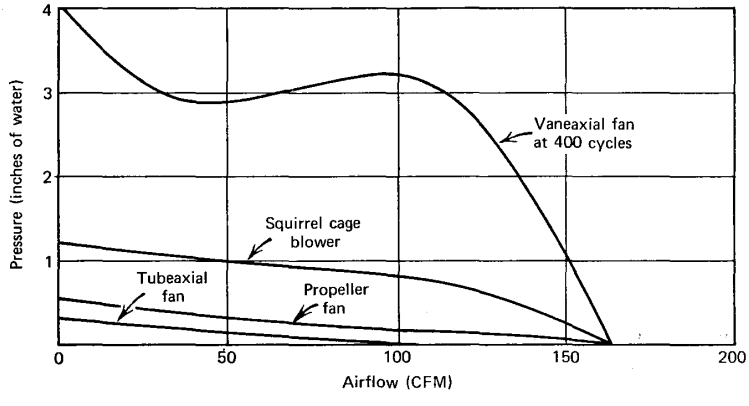

FIGURE 4.16
Comparison of the pressure—airflow characteristics of propeller fan, tubeaxial fan, vaneaxial fan, and squirrel cage blower

Many different levels of performance can be obtained from a particular type of fan by using different wheel diameters, different rotation speeds, different motors, bearings, etc. Therefore, Table 4.1 does not really provide a comparison between types of fans, but only a comparison between the four particular fans illustrated in Figures 4.12 through 4.15.

The propeller fan, the squirrel cage blower, and the vaneaxial fan all provide about the same free delivery of 160 CFM. The squirrel cage blower

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TABLE 4.1
Comparison of Fan Types

Fan Type	Propeller Fan	Tubeaxial Fan	Vaneaxial Fan	Squirrel Cage Blower
Specific Model	Rotron MF 5501	Rotron Muffin	Rotron Aximax 3	Rotron VS38A2
Size	8 in. Dia. × 5 in.	4.7 in. × 4.7 in. ×	3 in. Dia. × 2.5 in.	7 in. × 7 in. × 6 in.
Weight (lbs)	2	1	1	7
Power (watts)	21	14	130	100
Acoustic noise (above speech interference level)	50 dB	45 dB	68 dB	52 dB

and the vaneaxial fan both provide their high air flow rates at high pressures. Both of these fan types require over 100 W of power for their operation. The power required by a fan is approximately proportional to the product of the fan's pressure and air flow. Therefore, a fan that has one-tenth the pressure capability at a given air flow will require only one-tenth the power. For this reason the low pressure propeller and tubeaxial fans shown require much less power than the vaneaxial fan or the squirrel cage blower.

The expected life of the tubeaxial fan is 10 years when the inlet air is at room temperature. In contrast, the life of the 400 cycle vaneaxial fan is only about 5000 hours because of its high rotation speed.

The acoustic noise characteristics of the tubeaxial fan, the vaneaxial fan and the squirrel cage blower are compared in Figure 4.17. This standard method of describing the acoustic noise of fans shows the sound pressure level in decibels above 3×10^{-9} lb/in^2 as a function of frequency. The reference of 3×10^{-9} lb/in^2 is the threshold of hearing of a tone at 1000 Hz.

The "speech interference level" rating of a fan is the average noise in the three octave bands centered around 500, 1000, and 2000 Hz. The speech interference level can be determined from Figure 4.17 for the particular fans shown and has been tabulated in Table 4.1.

Low frequency acoustic noise does not sound as loud to the human ear as high frequency noise. The dotted curves of Figure 4.17 are equal loudness contours which include this effect. Note that a 37 dB pressure level at 2000 Hz "sounds" just as loud as a 40 dB level at 1000 Hz. The upper equal loudness

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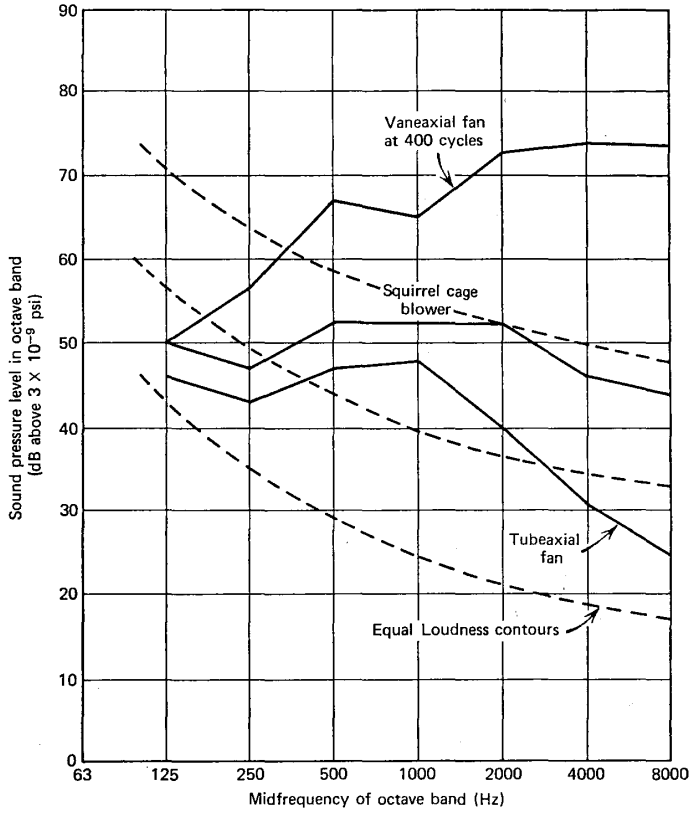

FIGURE 4.17
Comparison of the acoustic noise of tubeaxial fan, vaneaxial fan, and squirrel cage blower

contour in Figure 4.17 is the noise level in a typical business office. The lower contour is the loudness of a whisper.

The noise of the tubeaxial fan is about 10 dB below the noise level of a typical office. A special version of this fan (which can supply only 50 CFM) has its noise lowered another 20 dB to almost the level of a whisper.

The noise of the squirrel cage blower is almost as loud as the total noise of an office. The noise of the vaneaxial fan is even higher, and sounds even

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louder than the sound pressure level curve of Figure 4.17 suggests because the noise is concentrated at high frequencies.

Fan Accessories

Forced air cooling of electronic equipment requires the use of several accessories besides just the fan and the cooling fins.

In many designs a filter must be used to keep dust out of the electronic equipment. The filter increases the pressure that the fan must overcome to force the air through the cooling fins. Figure 4.18 shows a filter mounted on the tubeaxial fan that was shown in Figure 4.13, and also the pressure versus air flow characteristics of the fan with and without the filter. The necessity of using the filter reduces the free air flow capability of the fan from 100 CFM

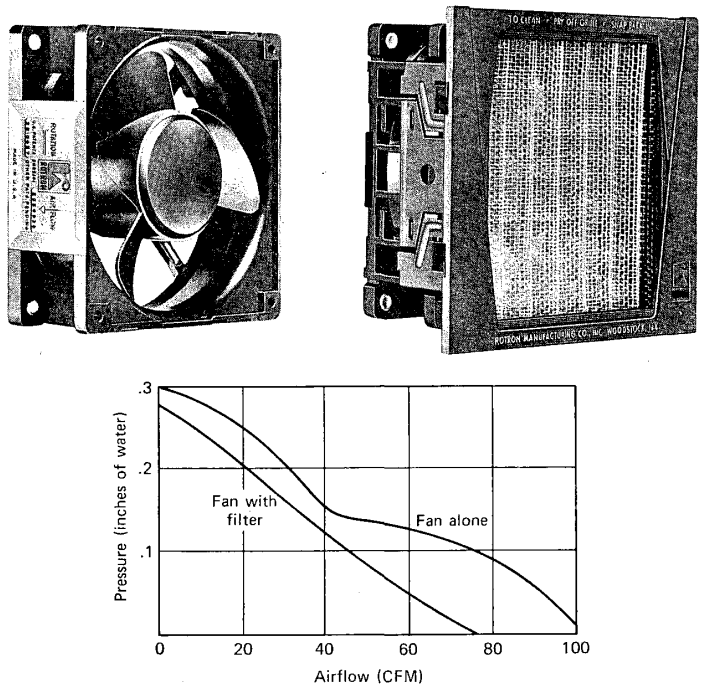

FIGURE 4.18
Vaneaxial fan with air filter and the effect of the air filter on the pressure —airflow characteristics (Photo courtesy of Rotron, Inc.)

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to 75 CFM, and at 50 CFM reduces the pressure capability of the fan almost to one-half of its capability without the filter.

In some designs, the cooling fan can be mounted directly on the cooling fins. In other designs the air must be brought to the cooling fins through a supply duct. The supply duct must be chosen so that it introduces only a negligible pressure drop. This can be achieved by keeping the cross sectional area of the supply duct several times larger than the total area of the cooling ducts between the fins, and by avoiding sharp bends or changes of cross section in the supply duct.

An air flow interlock is shown in Figure 4.19. If air cooling is necessary to keep the temperature of the electronic components below their safe operating value, and if the required air flow is then not supplied because the ducting is clogged or the fan fails or is not turned on, the electronic components will be damaged. The air flow interlock, which is a pressure sensitive device actuated by the air flow itself, prevents this problem by turning off the equipment when the required flow of air is not present.

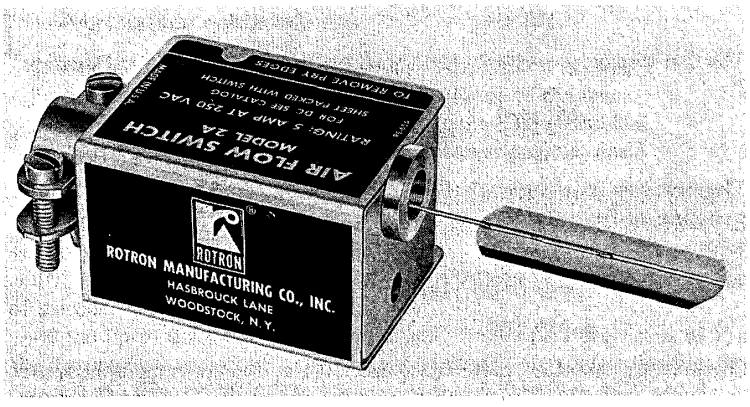

FIGURE 4.19
Airflow interlock switch (Photo courtesy of Rotron, Inc.)

4.7 EXAMPLES

The optimum design of fins for forced air cooling and the choice of the most appropriate fan can be made from the equations, figures, and tables of this chapter. The correct use of this design information is illustrated in this section by the following examples:

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- The air cooled heat sink described in Section 4.1.
- A forced air cooled semiconductor heat sink.
- A forced air cooled rack of card-mounted components.
- An air cooled, high power gridded tube.

EXAMPLE 1
A Forced Air Cooled Heat Sink

The important dimensions of the forced air cooled heat sink described in Section 4.1 and shown in Figure 4.1 are as follows:

Fin spacing (w)	.090 in.
Fin height (z)	1.5 in.
Fin length (L)	1.0 in.
Fin thickness (y)	.033 in.
Fin material	Copper
Number of ducts (n)	13
Total air flow (f)	45 CFM
Inlet air temperature	20°C
Altitude	Sea level

The first step is to calculate the temperature rise of the air from Equation 4.2:

$$\frac{\Delta T\,(\text{air})}{Q} = \frac{1.73}{f} = \frac{1.73}{45\ \text{CFM}} = .04°\text{C/W}.$$

The next step is to calculate the temperature rise of the fin surface above the cooling air from Equation 4.3:

$$\frac{\Delta T\,(\text{fin-air})}{Q} = \frac{140\ w}{n^{.2} z^{.2} f^{.8} L} = \frac{140 \times .090\ \text{in.}}{(13)^{.2} \times (1.5\ \text{in.})^{.2} \times (45\ \text{CFM})^{.8} \times 1\ \text{in.}}$$
$$= .33°\text{C/W}.$$

Equation 4.3 assumes a perfectly conducting fin. The fin temperature must be corrected to account for the finite thermal conductivity of the fin. From Equation 2.3 of Chapter 2,

$$\left(\frac{\Delta T}{Q}\right)_{\text{conduction}} = \frac{z}{kyL}$$
$$= \frac{1.5\ \text{in.}}{10\ \text{W/in. °C} \times .033\ \text{in.} \times 1.0\ \text{in.}}$$
$$= 4.5°\text{C/W}$$

where the thermal conductivity of copper from Table 2.1 of Chapter 2 is 10 W/in. °C.

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$(\Delta T/Q)_{\text{transfer}}$ is .33°C/W for the heat sink of 13 fins, and is therefore 13 times greater or 4.3°C/W for a single fin.

$$\frac{\left(\frac{\Delta T}{Q}\right)_{\text{conduction}}}{\left(\frac{\Delta T}{Q}\right)_{\text{transfer}}} = \frac{4.5°\text{C/W}}{4.3°\text{C/W}} = 1.05.$$

From Figure 2.13 of Chapter 2, fin efficiency is 76%. The temperature at the base of the fin is increased by 1/.76 over the value calculated by Equation 4.3 which assumes a perfectly conducting fin.

$$\frac{\Delta T\ (\text{fin-air})}{Q} = \frac{.33°\text{C/W}}{.76} = .43°\text{C/W}.$$

The temperature difference between the tip of the fin and the air is 65% of the temperature difference between the base of the fin and the air.

These calculations can now be used to estimate the temperature at the following locations on the heat sink:

1. At the air inlet end of the heat sink at the base of the fins.
2. At the air inlet end of the heat sink at the tip of the fins.
3. At the air outlet end of the heat sink at the base of the fins.
4. At the air outlet end of the heat sink at the tip of the fins.

An air inlet temperature of 20°C and a power of 100 W will be assumed. At the inlet end of the heat sink the air temperature rise is zero because the air is just starting through the fins. The fin temperature rise is .43°C/W × 100 W = 43°C, so the fin temperature is 20°C + 43°C = 63°C at the base of the fins.

At the tip of the fins on the air inlet end, the temperature rise is 65% of the temperature rise of the base, because of the requirement to conduct heat through the fins, or 28°C. The fin temperature is therefore 48°C. At the air outlet end of the fins, the air temperature has increased by .04°C/W × 100 W = 4°C in absorbing the 100 W of power. The fin temperatures at the base and tip of the fins are correspondingly 4°C higher at the air outlet end, neglecting any axial conduction along the heat sink.

The temperature rise of the fins above the inlet air temperature at their base in the center of the heat sink from these calculations should be .43°C/W + .02°C/W = .45°C/W. Half of the total air temperature rise was taken as this would be the air temperature halfway through the fins. The measured value, as shown in Figure 4.2, is .26°C/W, and this degree of correlation is typical of the accuracy of calculations made with Equation 4.3.

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TABLE 4.2
The effect of design changes on heat sink performance

Change	Units	Reference	Material	Fin Spacing	Fin Height	Fin Length	Fin Thickness
Fin spacing (w)	inches	.090	.090	.060	.090	.090	.090
Fin height (z)	inches	1.5	1.5	1.5	3.0	1.5	1.5
Fin length in direction of air flow (L)	inches	1.0	1.0	1.0	1.0	2.0	1.0
Fin thickness (y)	inches	.033	.033	.033	.033	.033	.063
Material	—	Copper	Aluminum	Copper	Copper	Copper	Copper
$\Delta T/Q$ (without fin efficiency)	°C/watt	.33	.33	.22	.29	.17	.33
Fin efficiency	—	.76	.65	.68	.60	.76	.86
$\Delta T/Q$	°C/watt	.43	.51	.32	.49	.22	.39
Pressure	inches of water	.73	.73	1.75	.18	.81	.73

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Pressure drop is calculated from Equation 4.4 as follows:

$$\Delta p = \frac{\left(\frac{f}{n}\right)^2}{(wz)^2}\left[1 + .01\,\frac{L}{w}\right] \times 10^{-3}$$

$$= \frac{\left(\frac{45\text{ CFM}}{13}\right)^2}{(.090\text{ in.} \times 1.5\text{ in.})^2}\left[1 + .01\,\frac{1\text{ in.}}{.090\text{ in.}}\right] \times 10^{-3}$$

$$= .73\text{ in. of water.}$$

The measured value of pressure drop shown in Figure 4.3 at this flow rate of 45 CFM is 1.2 in. of water. This degree of correlation is typical of the accuracy of calculations made with Equation 4.4.

The calculated effect of varying the material of the heat sink, fin spacing, fin height, fin length, and fin thickness are shown in Table 4.2. These calculations were made in the same manner as outlined above.

The average temperature rise per watt of power and the pressure required to force air through the fins for each design change are compared on the last two lines of the table. Note that increasing fin height significantly reduces pressure requirements and that increasing fin length in the direction of air flow significantly decreases the temperature rise with only a very small increase in pressure.

EXAMPLE 2
A Forced Air Cooled Semiconductor Heat Sink

A forced air cooled heat sink with several semiconductors mounted on it is shown in Figure 4.20 along with its important dimensions. This heat sink has been carefully designed to use the tubeaxial fan that was shown in Figure 4.13, and this fan can be seen mounted on the back end of the heat sink.

The individual quadrants of the heat sink are thermally isolated, and so the calculations can be made for a single quadrant. Each quadrant uses one-fourth of the air flow supplied by the fan. The calculations will be made for an air flow of 12 CFM/quadrant, or 48 CFM total.

The first step is to calculate the temperature rise of the air above the inlet air temperature. The temperature rise of the air is calculated from Equation 4.2 as follows:

$$\frac{\Delta T\text{ (air)}}{Q} = \frac{1.73}{f} = \frac{1.73}{12\text{ CFM}} = .14°\text{C/W}.$$

Halfway through the fins, at a point directly under the component, the air temperature rise is .07°C/W.

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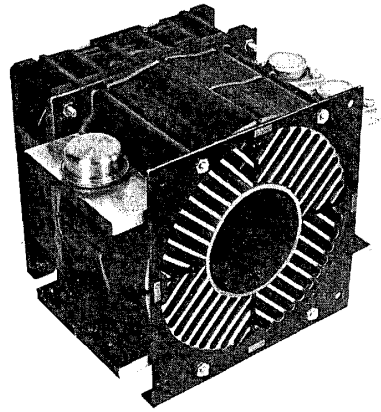

DIMENSIONS OF ONE QUADRANT

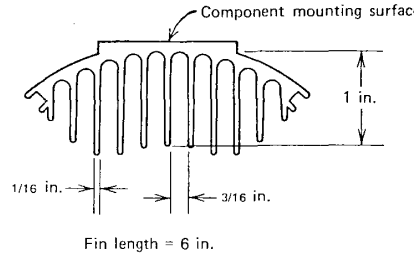

FIGURE 4.20
Forced air cooled semiconductor heat sink (Photo courtesy of Wakefield Engineering, Inc.)

The temperature rise of the fins above the cooling air is calculated from Equation 4.3. The duct dimensions are not uniform, so to use Equation 4.3 the small ducts at each end of the structure will be neglected. The number of ducts used in the calculation is therefore eight, and their average height is 1 in.

$$\frac{\Delta T\,(\text{fin-air})}{Q} = \frac{140\,w}{n^{.2}z^{.2}f^{.8}L}$$

$$= \frac{140 \times .187\text{ in.}}{(8)^{.2}(1\text{ in.})^{.2}(12\text{ CFM})^{.8} \times 6\text{ in.}}$$

$$= .39°\text{C/W}.$$

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Fin efficiency should be calculated next. The fins are made of aluminum, whose thermal conductivity from Table 2.1 of Chapter 2 is 5.5 W/in. °C.

From Equation 2.3 of Chapter 2:

$$\left(\frac{\Delta T}{Q}\right)_{\text{conduction}} = \frac{z}{kyL}$$

$$= \frac{1 \text{ in.}}{5.5 \text{ W/in. °C} \times .063 \text{ in.} \times 6 \text{ in.}}$$

$$= .5\text{°C/W}.$$

$(\Delta T/Q)_{\text{transfer}}$ is .39°C/W for the heat sink of eight fins, and is therefore eight times higher for a single fin.

$$\frac{\left(\frac{\Delta T}{Q}\right)_{\text{conduction}}}{\left(\frac{\Delta T}{Q}\right)_{\text{transfer}}} = \frac{.5\text{°C/W}}{8 \times .39\text{°C/W}} = .16.$$

Fin efficiency from Figure 2.13 of Chapter 2 is .94, so the temperature rise at the base of the fins is .39/.94 = .41°C/W.

The total temperature rise above the inlet air temperature at the center of the heat sink is obtained from Equation 4.1 by adding the temperature rise of the air, which is .07°C/W, to the temperature rise of the fins above the air, which is .41°C/W, for a total of .48°C/W. This calculated value is in excellent agreement with the measured value of .5°C/W.

The pressure drop through the heat sink can be estimated from Equation 4.4 as follows:

$$\Delta p = \frac{\left(\frac{f}{n}\right)^2}{(wz)^2}\left[1 + .01\frac{L}{w}\right] \times 10^{-3}$$

$$= \frac{\left(\frac{12 \text{ CFM}}{8}\right)^2}{(.187 \text{ in.} \times 1 \text{ in.})^2}\left[1 + .01 \times \frac{6 \text{ in.}}{.187 \text{ in.}}\right] \times 10^{-3}$$

$$= .09 \text{ in. of water.}$$

The measured pressure drop through this forced air cooled heat sink is .1 in. of water with an air flow of 12 CFM/quadrant.

EXAMPLE 3
A Forced Air Cooled Rack of Card-Mounted Components

Figure 4.21 shows a typical rack of card-mounted components. Cooling of this rack by radiation and natural convection was considered in Example 3 of Chapter 3. In this present example, forced air cooling of the same configuration will be calculated.

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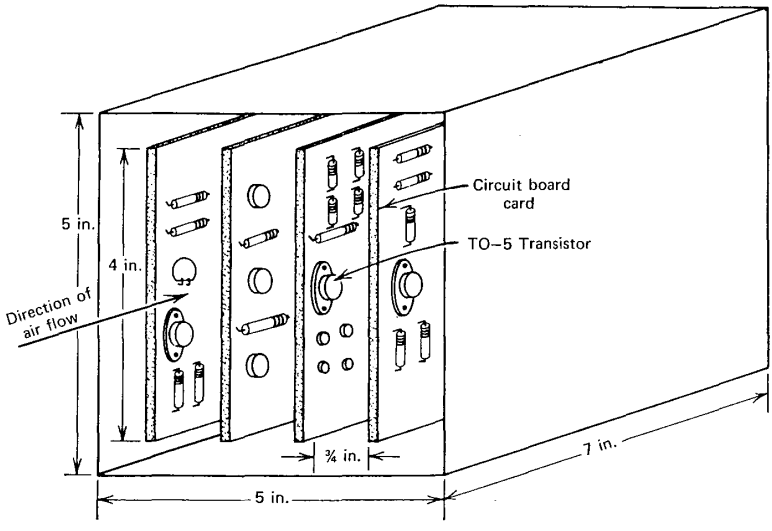

FIGURE 4.21
Forced air cooled rack of card-mounted components

The equipment shown in Figure 4.21 consists of four 4 in. × 6 in. × 1/16 in. thick circuit boards. The electronic components are mounted on one side of each of the circuit boards, and the connections between components are made with printed circuitry on the opposite side of the boards.

The four cards are mounted in a rack 5 in. wide × 7 in. long × 5 in. high. The ends of the rack are open to allow air to be blown through the cards. The spacing between the cards is approximately 3/4 in., when allowance is made for the height of the components mounted on the card surface.

The pressure versus air flow characteristics of this rack can be estimated from Curve B of Figure 4.7. At 60 CFM the pressure is about .04 in. of water. This airflow could be provided by the tubeaxial fan shown in Figure 4.18 with its filter.

An estimate of the temperature of any of the components mounted on the card can be made using Figure 4.8. A sample calculation will be made for a TO-5 transistor. The important dimensions for this calculation are:

Transistor diameter (d):	.3 in.
Cross sectional-area of duct (A):	3/4 in. × 5 in.
Airflow per duct (f):	60/6 = 10 CFM.
Height of transistor in air stream:	1/4 in.

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With these dimensions

$$d\left(\frac{A}{f}\right) = \frac{.3 \text{ in.} \times 10 \text{ CFM}}{3/4 \times 5 \text{ in}^2} = .8$$

and the power transferred, as shown in Figure 4.8, is .03 W/°C for a 1 in. high component. Since the TO-5 transistor is only 1/4 in. high, the power transferred is only 1/4 × .03 W/°C = .0075° W/°C.

If the transistor case temperature is 70°C and the air temperature is 20°C, so that the temperature difference is 50°C, a power of .4 W can be transferred by this air cooling design. This power is 6.5 times more than can be transferred with the same temperature difference by natural convection, as was shown in Example 3 of Chapter 3.

The same rack of card-mounted components will be considered again in Chapter 6, where cooling will be by liquid evaporation in a liquid filled package.

EXAMPLE 4
An Air Cooled, High Power Gridded Tube

A high power gridded tube used for industrial radio frequency heating equipment is shown in Figure 4.22 along with its important dimensions. The tube is specifically designed for forced air cooling. When the tube is operated with a plate dissipation of 3000 W, the manufacturer recommends a total air flow of 100 CFM to keep the ceramic seals below 200°C and the air cooling fins below 250°C, assuming an inlet air temperature of 50°C. At this flow rate, the measured pressure drop is .7 in. of water. The squirrel cage blower shown in Figure 4.15 is a good fan for cooling this tube.

Because the ceramic seals must be kept at a lower temperature than the cooling fins, the cooling air should be blown over the seals first and then through the fins.

The first step in calculating the forced air cooling of this tube is to calculate the temperature rise of the cooling air from Equation 4.2, as follows:

$$\frac{\Delta T\,(\text{air})}{Q} = \frac{1.9}{f} = \frac{1.9}{100 \text{ CFM}} = .019°\text{C/W}.$$

The multiplying constant of Equation 4.2 and all the other equations used in this example have been modified by the use of Figure 4.9 to account for the inlet air temperature of 50°C. When 3000 W are being absorbed, ΔT (air) = 3000 W × .019°C/W = 57°C. If the inlet air temperature is 50°C, the outlet air temperature is 107°C.

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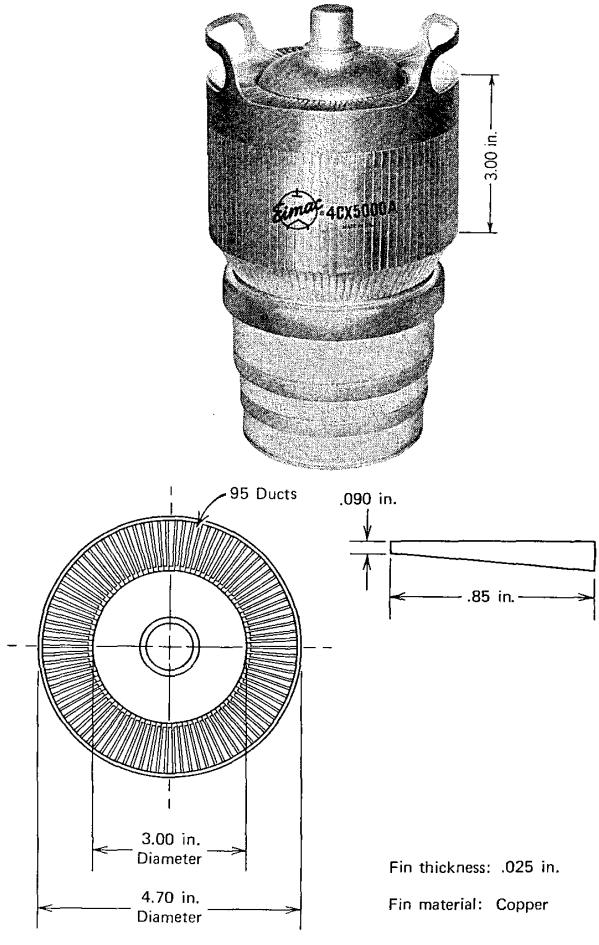

FIGURE 4.22
A forced air cooled high power gridded tube (Photo courtesy of Varian Associates)

The next step is to calculate the temperature rise of the fins above the cooling air temperature. There are 95 ducts. The shape of the ducts is not quite rectangular; the minimum dimension of .090 in. is used for the duct width. From Equation 4.3:

$$\frac{\Delta T\,(\text{fin-air})}{Q} = \frac{150\,w}{n^{.2}z^{.2}f^{.8}L}$$

$$= \frac{150 \times .090 \text{ in.}}{(95)^{.2} \times (.85 \text{ in.})^{.2} \times (100 \text{ CFM})^{.8} \times 3 \text{ in.}}$$

$$= .047°\text{C/W}.$$

Fin efficiency should be calculated next. From Equation 2.3 of Chapter 2:

$$\left(\frac{\Delta T}{Q}\right)_{\text{conduction}} = \frac{z}{kyL}$$

$$= \frac{.85 \text{ in.}}{10 \text{ W/in. °C} \times .025 \text{ in.} \times 3 \text{ in.}}$$

$$= 1.3 \text{ °C/W}.$$

$(\Delta T/Q)_{\text{transfer}}$ is .047°C/W for the total heat sink of 95 fins and is therefore 95 times greater or 4.5°C/W for a single fin.

$$\frac{\left(\frac{\Delta T}{Q}\right)_{\text{conduction}}}{\left(\frac{\Delta T}{Q}\right)_{\text{transfer}}} = \frac{1.3°\text{C/W}}{4.5°\text{C/W}} = .3$$

and from Figure 2.13 of Chapter 2, fin efficiency is 91%, and

$$\frac{\Delta T\,(\text{fin-air})}{Q} = \frac{.047°\text{C/W}}{.91} = .052°\text{C/W}.$$

When 3000 W is being transferred the temperature rise of the base of the cooling fins is 3000 W × .052°C/W = 156°C. At the inlet side of the fins the calculated fin temperature is 156°C + 50°C since the entering cooling air is at 50°C. At the outlet end of the fins the calculated fin temperature is 263°C, because the air temperature has increased 57°C in absorbing the 3000 W of power.

The pressure drop through the fins is calculated from Equation 4.5 as follows:

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$$\Delta p = \frac{\left(\frac{f}{n}\right)^2}{(wz)^2}\left[1 + .01\frac{L}{w}\right] \times .9 \times 10^{-3}$$

$$= \frac{\left(\frac{100\text{ CFM}}{95}\right)^2}{(.090\text{ in.} \times .85\text{ in.})^2}\left[1 + .01\frac{3\text{ in.}}{.090\text{ in.}}\right] \times .9 \times 10^{-3}$$

$$= .2\text{ in. of water.}$$

The calculated pressure is considerably lower than the measured value of .7 in. of water, but the measured value includes the additional pressure required to force the air through the tube mounting socket.

4.8 REFERENCES

Useful references for forced air cooling of electronic equipment are as follows.

1. Krauss, A. D., *Cooling Electronic Equipment*, Prentice Hall, Inc., Englewood Cliffs, N.J., 1965, Chapter 2, pp. 30–51, Chapter 9, pp. 194–213, Chapter 12, pp. 248–269.
2. *Guide Manual of Cooling Methods for Electronic Equipment*, NAVSHIPS 900, 190, U.S. Government Printing Office, Washington, D.C., 1955, pp. 74–89.
3. *How to Select the Optimum Fan for Your Applications*, Application Bulletins 7041–7046, Pamotor, Inc., Burlingame, Calif., 1971.
4. *Fan Selection for High Altitude Applications*, Application Note B 750, Rotron, Inc., Woodstock, New York, 1969.
5. Dietz, H. G., *Forced Air Cooling Primer for the Electronics Engineer*, Henry G. Dietz Co., Long Island City, New York, 1964.
6. Sutherland, R. I., *Care and Feeding of Power Grid Tubes*, Eimac Division of Varian, San Carlos, Calif., 1967.
7. *Air Moving Devices for Commercial, Industrial and Military Applications*, Rotron, Inc., Woodstock, New York, 1970.
8. Hay, A. D., *Air Cooling Electronic Enclosures*, McLean Engineering Laboratory, Princeton Junction, N.J., 1966.

Reference 1 provides, in Chapters 2 and 12, the basic hydro-dynamic equations from which the formulas and graphs of this chapter are derived. It also provides, in Chapter 9, a detailed discussion of the theory of fan operation and the effect of ducting dimensions on air pressure drop.

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Reference 2 provides the same information as Reference 1 in somewhat less detail.

Reference 3 describes the tradeoff between the factors that must be considered in fan selection, and presents a particularly good discussion of fan acoustic noise.

Reference 4 discusses the procedure for extrapolating sea level measurements to high altitude, including the choice of fan.

Reference 5 provides practical rules for designing effective forced air cooling systems.

Reference 6 provides practical rules for applying forced air cooling to high power transmitter tubes, including the scaling of sea level designs to high altitudes and high inlet air temperatures.

Reference 7 provides application information and a detailed listing of representative fans and blowers for cooling electronic equipment.

Reference 8 discusses the forced air cooling of racks and cabinets.

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