Cooling Techniques for Electronic Equipment

DAVE S. STEINBERG

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Cooling Techniques for Electronic Equipment

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SEC et al. v. MRI
SEC Exhibit 1013.0004
IPR2023-00221

To My Wife Annette
And To My Two Daughters Cori and Stacie

SEC et al. v. MRI
SEC Exhibit 1013.0005
IPR2023-00221

Preface

Electronic equipment is slowly reaching into almost every phase of modern living, from sewing machines and washing machines to mass transportation and atomic energy control systems. The reliability of these systems is of great importance to our comfort and safety. If a transistor fails in a television set, it may only cause a minor inconvenience. However, if an electronic control system should malfunction because it has overheated, it may result in substantial property damage and possible injury.

Electronic systems are rapidly shrinking in size, while their complexity and capability continue to grow at an amazing rate. As the power has been increasing, the volume has been decreasing, resulting in a dramatic increase in the heat density. As a result, the temperatures in many electronic systems have been rising rapidly, producing a large increase in the number of failures.

High failure rates in electronic boxes may also be caused by high thermal stresses in the solder joints of electronic components mounted on circuit boards. This is usually due to a high thermal expansion coefficient with insufficient strain relief in the component lead wire. This area is discussed in detail, together with recommendations for mounting components to prevent this type of failure.

The purpose of this book is to show designers and engineers quick methods for designing electronic hardware to withstand severe thermal environments without failing. Techniques are presented that will permit the development of many different types of reliable electronic systems *without* the aid of the powerful new high speed digital computers.

This book was developed from a series of seminars, lectures, and short courses for the cooling of electronic equipment, which I have presented at the University of Wisconsin–Extension every year since 1975. The book was influenced by my industrial experience in the mechanical design, packaging, and testing of many different types of sophisticated electronic components and systems during the past 25 years.

Mathematical modeling techniques using analog resistor networks are also included. For those who wish to use high speed digital computers to

SEC et al. v. MRI
SEC Exhibit 1013.0006
IPR2023-00221

solve thermal problems, these techniques can be used to break up a complex system into many individual thermal resistors and nodes.

In an attempt to keep the book as simple as possible, the derivations of equations were minimized. The emphasis is always on the real electronic hardware that is used in today's sophisticated electronic systems. Many sample problems are presented, to demonstrate practical and cost effective methods for designing efficient, reliable cooling systems.

Although the metric system for weights and measures is widely used in Europe, at the present time it is used by only a few American companies. However, there appears to be a strong movement toward conversion from the present English system to the metric system. This book was therefore written with dual units, English and metric, to permit electronic equipment designers and engineers to work with either set of units effectively.

I thank Mr. Joel Newberger for his contributions in the section on induced draft cooling, and Mr. Joel Sloan for proofreading several sections of the book.

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Westlake Village, California
August 1980

SEC et al. v. MRI
SEC Exhibit 1013.0007
IPR2023-00221

Contents

Preface vii

Symbols xvii

1 Evaluating the Cooling Requirements 1

- **1.1** Heat Sources, 1
- **1.2** Heat Transmission, 2
- **1.3** Steady State Heat Transfer, 4
- **1.4** Transient Heat Transfer, 5
- **1.5** Electronic Equipment for Airplanes, Missiles, Satellites, and Spacecraft, 6
- **1.6** Electronic Equipment for Ships and Submarines, 8
- **1.7** Electronic Equipment for Communication Systems and Ground Support Systems, 9
- **1.8** Minicomputers, Microcomputers, and Microprocessors, 10
- **1.9** Cooling Specifications for Electronics, 11
- **1.10** Specifying the Power Dissipation, 12
- **1.11** Dimensional Units and Conversion Factors, 14

2 Designing the Electronic Chassis 21

- **2.1** Formed Sheet Metal Electronic Assemblies, 21
- **2.2** Dip Brazed Boxes with Integral Cold Plates, 22
- **2.3** Plaster Mold and Investment Castings with Cooling Fins, 24
- **2.4** Die Cast Housings, 25

SEC et al. v. MRI
SEC Exhibit 1013.0008
IPR2023-00221

2.5 Large Sand Castings, 26
2.6 Extruded Sections for Large Cabinets, 26
2.7 Humidity Considerations in Electronic Boxes, 26
2.8 Conformal Coatings, 28
2.9 Sealed Electronic Boxes, 29
2.10 Standard Electronic Box Sizes, 33

3 Conduction Cooling for Chassis and Circuit Boards 35

3.1 Concentrated Heat Sources, Steady State Conduction, 35
3.2 Mounting Electronic Components on Brackets, 36
3.3 Sample Problem—Transistor Mounted on a Bracket, 39
3.4 Uniformly Distributed Heat Sources, Steady State Conduction, 41
3.5 Sample Problem—Cooling Integrated Circuits on a PCB, 44
3.6 Circuit Board with an Aluminum Heat Sink Core, 46
3.7 Sample Problem—Temperature Rise along a PCB Heat Sink Plate, 46
3.8 How to Avoid Warping on PCBs with Metal Heat Sinks, 47
3.9 Chassis with Nonuniform Wall Sections, 48
3.10 Sample Problem—Heat Flow along Nonuniform Bulkhead, 50
3.11 Two Dimensional Analog Resistor Networks, 53
3.12 Sample Problem—Two Dimensional Conduction on a Power Supply Heat Sink, 54
3.13 Heat Conduction across Interfaces in Air, 60
3.14 Sample Problem—Temperature Rise across a Bolted Interface, 64
3.15 Sample Problem—Temperature Rise across a Small Air Gap, 65
3.16 Heat Conduction across Interfaces at High Altitudes, 66
3.17 Outgassing at High Altitudes, 69
3.18 Circuit Board Edge Guides, 70

SEC et al. v. MRI
SEC Exhibit 1013.0009
IPR2023-00221

3.19 Sample Problem—Temperature Rise across a PCB Edge Guide, 72
3.20 Heat Conduction through Sheet Metal Covers, 72
3.21 Radial Heat Flow, 73
3.22 Sample Problem—Temperature Rise through a Cylindrical Shell, 74

4 Mounting and Cooling Techniques for Electronic Components 77

4.1 Various Types of Components, 77
4.2 Mounting Components on PCBs, 78
4.3 Sample Problem—Hot Spot Temperature of an Integrated Circuit on a Plug-in PCB, 81
4.4 How to Mount High Power Components, 87
4.5 Sample Problem—Mounting High Power Transistors on a Heat Sink Plate, 89
4.6 Electrically Isolating High Power Components, 91
4.7 Sample Problem—Mounting a Transistor on a Heat Sink Bracket, 92
4.8 Potted Modules, 94
4.9 Sample Problem—Temperature Rise in a Potted Module, 95
4.10 Component Lead Wire Strain Relief, 98

5 Practical Guides for Natural Convection and Radiation Cooling 106

5.1 How Natural Convection Is Developed, 106
5.2 Natural Convection for Flat Vertical Plates, 109
5.3 Natural Convection for Flat Horizontal Plates, 109
5.4 Heat Transferred by Natural Convection, 110
5.5 Sample Problem—Vertical Plate Natural Convection, 111
5.6 Turbulent Flow with Natural Convection, 113
5.7 Sample Problem—Heat Lost from an Electronic Box, 114
5.8 Finned Surfaces for Natural Convection Cooling, 117
5.9 Sample Problem—Cooling Fins on an Electronic Box, 119
5.10 Natural Convection Analog Resistor Networks, 121
5.11 Natural Convection Cooling for PCBs, 123

SEC et al. v. MRI
SEC Exhibit 1013.0010
IPR2023-00221

5.12 Natural Convection Coefficient for Enclosed Air Space, 125
5.13 Sample Problem—PCB Adjacent to a Chassis Wall, 126
5.14 High Altitude Effects on Natural Convection, 129
5.15 Sample Problem—PCB Cooling at High Altitudes, 130
5.16 Radiation Cooling of Electronics, 132
5.17 Radiation View Factor, 136
5.18 Sample Problem—Radiation Heat Transfer from a Hybrid, 142
5.19 Sample Problem—Junction Temperature of a Dual FET Switch, 145
5.20 Radiation Heat Transfer in Space, 147
5.21 Effects of α/e on Temperatures in Space, 148
5.22 Sample Problem—Temperatures of an Electronic Box in Space, 150
5.23 Simplified Radiation Heat Transfer Equation, 151
5.24 Sample Problem—Radiation Heat Loss from an Electronic Box, 152
5.25 Combining Convection and Radiation Heat Transfer, 154
5.26 Sample Problem—Electronic Box in an Airplane Cockpit Area, 155
5.27 Equivalent Ambient Temperature for Reliability Predictions, 157
5.28 Sample Problem—Equivalent Ambient Temperature of an RC07 Resistor, 159
5.29 Increase in Effective Emittance on Extended Surfaces, 160

6 Forced Air Cooling for Electronics 164

6.1 Forced Cooling Methods, 164
6.2 Cooling Air Flow Direction for Fans, 165
6.3 Static Pressure and Velocity Pressure, 166
6.4 Losses Expressed in Terms of Velocity Heads, 170
6.5 Sample Problem—Air Flow Loss at a Fan Entrance, 171
6.6 Establishing the Flow Impedance Curve for an Electronic Box, 172

SEC et al. v. MRI
SEC Exhibit 1013.0011
IPR2023-00221

6.7 Sample Problem—Fan Cooled Electronic Box, 173
6.8 Hollow Core PCBs, 189
6.9 Cooling Air Fans for Electronic Equipment, 192
6.10 Air Filters, 194
6.11 Cutoff Switches, 195
6.12 Static Pressure Loss Tables and Charts, 195
6.13 High Altitude Conditions, 196
6.14 Sample Problem—Fan Cooled Box at 30,000 Feet, 199
6.15 Other Convection Coefficients, 203
6.16 Sample Problem—Cooling A TO-5 Transistor, 205
6.17 Conditioned Cooling Air from an External Source, 207
6.18 Sample Problem—Generating a Cooling Air Flow Curve, 208
6.19 Static Pressure Losses for Various Altitude Conditions, 209
6.20 Sample Problem—Static Pressure Drop at 65,000 Feet, 212
6.21 Total Pressure Drop for Various Altitude Conditions, 218
6.22 Sample Problem—Total Pressure Loss through an Electronic Box, 219
6.23 Finned Cold Plates and Heat Exchangers, 219
6.24 Pressure Losses in Multiple Fin Heat Exchangers, 221
6.25 Fin Efficiency Factor, 223
6.26 Sample Problem—Hollow Core PCB with a Finned Heat Exchanger, 225
6.27 Undesirable Air Flow Reversals, 239

7 Cooling Minicomputers, Microcomputers, and Microprocessors 242

7.1 Introduction, 242
7.2 Minicomputer Systems, 242
7.3 Sample Problem—PCB Mounted above a Minicomputer Floppy Disk, 244
7.4 Sample Problem—Fan Cooled Minicomputer, 249
7.5 Microcomputer Systems, 252
7.6 Sample Problem—Cooling a Microcomputer, 253

SEC et al. v. MRI
SEC Exhibit 1013.0012
IPR2023-00221

7.7 Microprocessor Development, 255
7.8 Mounting Microprocessors to Resist Vibration Fatigue, 256
7.9 Microprocessors in Severe Thermal Environments, 257
7.10 Sample Problem—Cooling a Microprocessor Mounted on a PCB, 257

8 Effective Cooling for Large Racks and Cabinets 260

8.1 Induced Draft Cooling for Large Consoles, 260
8.2 Air Flow Losses for Large Cabinets, 261
8.3 Flotation Pressure and Pressure Loss, 262
8.4 Sample Problem—Induced Draft Cooling for a Large Cabinet, 262
8.5 Natural Cooling for Large Cabinets with Many Flow Restrictions, 267
8.6 Sample Problem—Temperature Rise of Cooling Air in a Cabinet with an Induced Draft, 268
8.7 Warning Note for Induced Draft Systems, 272
8.8 Tall Cabinets with Stacked Card Buckets, 273
8.9 Sample Problem—Induced Draft Cooling of a Console with Seven Stacked Card Buckets, 274
8.10 Electronics Packaged within Sealed Enclosures, 278
8.11 Small Enclosed Modules within Large Consoles, 281
8.12 Sample Problem—Small PCB Sealed within an RFI Enclosure, 283
8.13 Test Data for Small Enclosed Modules, 290
8.14 Pressure Losses in Series and Parallel Air Flow Ducts, 293
8.15 Sample Problem—Series and Parallel Air Flow Network, 294

9 Transient Cooling for Electronic Systems 300

9.1 Simple Insulated Systems, 300
9.2 Sample Problem—Transient Temperature Rise of a Transformer, 301
9.3 Thermal Capacitance, 302

SEC et al. v. MRI
SEC Exhibit 1013.0013
IPR2023-00221

9.4 Time Constant, 303
9.5 Heating Cycle Transient Temperature Rise, 304
9.6 Sample Problem—Transistor on a Heat Sink, 304
9.7 Temperature Rise for Different Time Constants, 308
9.8 Sample Problem—Time for Transistor to Reach 95% of its Stabilized Temperature, 310
9.9 Cooling Cycle Transient Temperature Change, 310
9.10 Sample Problem—Transistor and Heat Sink Cooling, 310
9.11 Transient Analysis for Temperature Cycling Tests, 312
9.12 Sample Problem—Electronic Chassis in a Temperature Cycling Test, 317
9.13 Sample Problem—Methods for Decreasing Hot Spot Temperatures, 322
9.14 Sample Problem—Transient Analysis of an Amplifier on a PCB, 324

10 Special Applications for Tough Cooling Jobs 331

10.1 New Technology—Approach with Caution, 331
10.2 Heat Pipes, 331
10.3 Degraded Performance in Heat Pipes, 333
10.4 Typical Heat Pipe Performance, 334
10.5 Heat Pipe Applications, 336
10.6 Direct and Indirect Liquid Cooling, 340
10.7 Forced Liquid Cooling Systems, 341
10.8 Pumps for Liquid Cooled Systems, 342
10.9 Storage and Expansion Tank, 343
10.10 Liquid Coolants, 343
10.11 Simple Liquid Cooling System, 344
10.12 Mounting Components for Indirect Liquid Cooling, 344
10.13 Basic Forced Liquid Flow Relations, 347
10.14 Sample Problem—Transistors on a Water Cooled Cold Plate, 351

References 361

Index 365

SEC et al. v. MRI
SEC Exhibit 1013.0014
IPR2023-00221

1
Evaluating the Cooling Requirements

1.1 HEAT SOURCES

Electronic equipment relies on the flow and control of electrical current to perform a fantastic variety of functions, in virtually every major industry throughout the world. Whenever electrical current flows through a resistive element, heat is generated in that element. An increase in the current or resistance produces an increase in the amount of heat that is generated in the element. The heat continues to be generated as long as the current continues to flow. As the heat builds up, the temperature of the resistive element starts to rise, unless the heat can find a flow path that carries it away from the element. If the heat flow path is poor, the temperature may continue to rise until the resistive element is destroyed and the current stops flowing. If the heat flow path is good, the temperature may rise until it stabilizes at a point where the heat flowing away from the element is equal to the heat generated by the electrical current flowing in the element.

Heat is generated by the flow of electrical current in electronic component parts such as resistors, diodes, integrated circuits (ICs), hybrids, transistors, microprocessors, relays, dual inline packages (DIPs), large scale integrated circuits (LSIs), and very large scale integrated circuits (VSIs).

Figure 1.1 shows an electronic chassis that has heat exchangers (cold plates) on the top and bottom surfaces which rely on conditioned cooling air for controlling temperatures on plug-in circuit boards.

Electronic components and electronic systems are rapidly shrinking in size while their complexity and capability continue to grow at an amazing rate. In addition, the power has been increasing while the volume has been decreasing. This has produced a dramatic increase in the power density, resulting in rapidly rising temperatures and a large increase in the number of failures.

The temperatures must be controlled on every component to ensure a reliable electronic system. If the operating temperatures become too high, electronic malfunctions may occur. Malfunctions may produce a simple out

SEC et al. v. MRI
SEC Exhibit 1013.0015
IPR2023-00221

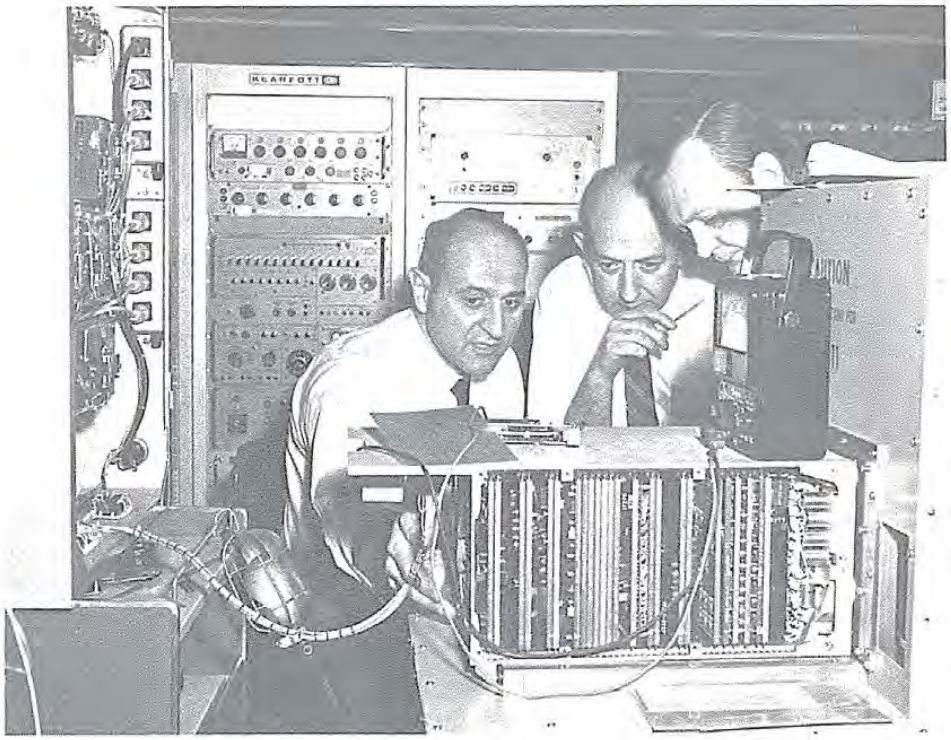


Figure 1.1 The author discussing design details with two associates. (Courtesy Kearfott Division, The Singer Co.)

of tolerance condition for a minor temperature increase, or a catastrophic failure for a major temperature increase.

Heat always flows from the hot area to the cool area. Since the electronic components are usually the source of the heat, the electronic components will usually be the hottest spots in an electronic system. (During transient conditions and temperature cycling tests, the electronic components may not necessarily be the hottest points in the system.) The basic heat transfer problem in electronic systems is, therefore, the removal of internally generated heat by providing a good heat flow path from the heat sources to an ultimate sink, which is often the surrounding ambient air.

1.2 HEAT TRANSMISSION

There are three basic methods by which heat can be transferred: conduction, convection, and radiation. The laws relating to these methods of heat transmission are of primary importance in the design and operation of electronic equipment.

SEC et al. v. MRI
SEC Exhibit 1013.0016
IPR2023-00221

Conduction is the transfer of kinetic energy from one molecule to another. In an opaque solid it is the only method of heat transfer, where heat flows from the hot areas to the cooler areas of the solid. Heat conduction also occurs in gases and liquids, but the amount of heat transferred is usually smaller for the same geometry.

Convection is the transfer of heat by the mixing action of fluids. When the mixing is due entirely to temperature differences within the fluid, resulting in different densities, the action is known as natural convection. When the mixing is produced by mechanical means, such as fans and pumps, the action is known as forced convection.

Thermal radiation is the transfer of energy by electromagnetic waves that are produced by bodies because of their temperature. A hot body radiates energy in all directions. When this energy strikes another body, the part that is absorbed is transformed into heat.

Most electronic systems make use of all three basic methods of heat transfer to some extent, even though one method may dominate the design. For example, an electronic box cooled by forced convection might utilize a fan to draw air over electronic components mounted on printed circuit boards (PCBs), as shown in Figure 1.2.

The greatest amount of heat is picked up by forced convection as the cooling air passes over the individual electronic components that are mounted on the PCB. However, some of the heat from the electronic components is conducted directly to the PCB under the component body, and some of the heat is conducted to the back side of the PCB, through the components electrical lead wires, as shown in Figure 1.3. Since the cooling

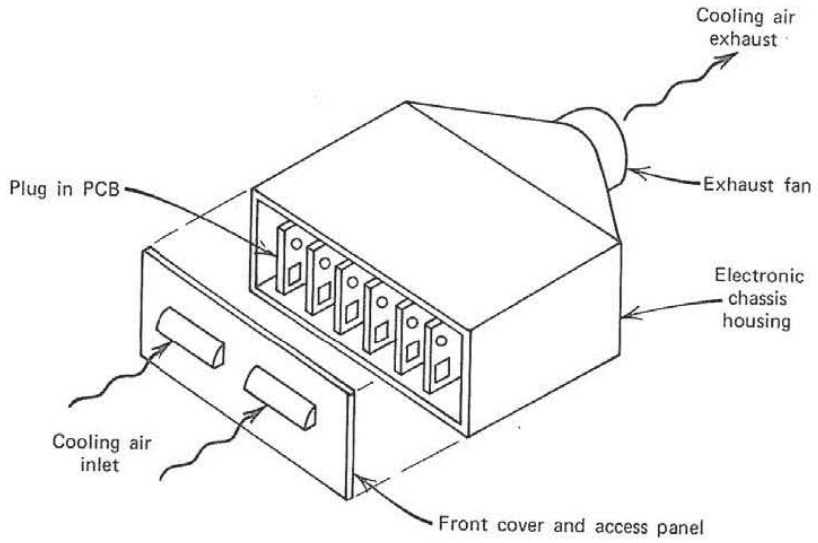


Figure 1.2 Electronic box cooled with an exhaust fan.

SEC et al. v. MRI
SEC Exhibit 1013.0017
IPR2023-00221

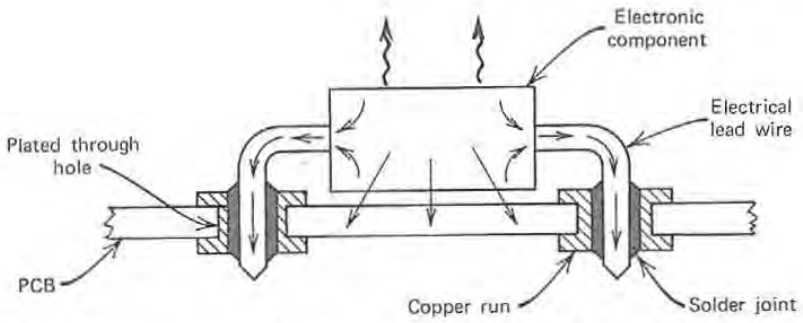


Figure 1.3 Heat conduction path from component, with heat flow through lead wires, to back side of PCB.

air passes over both surfaces of the PCB, the conduction of heat to the back side of the PCB provides additional surface area for improved cooling.

In addition, some of the heat is radiated from the hot components to the surrounding chassis walls and to the cooler spots on adjacent PCBs. This helps to reduce the component hot spot temperatures.

1.3 STEADY STATE HEAT TRANSFER

If an electronic system is turned on and left running for a very long period of time, and if the power requirements remain constant during that period, the temperatures of the electronic components and their mounting structures, such as PCBs, will usually become stable. Minor fluctuations in the line voltages, small changes in the physical properties of the individual components, and slight variations in the outside ambient conditions may have some small effects on the temperatures within the electronic system. For all practical purposes, however, the heat gained (or the power dissipated) by the electronic components is equal to the heat lost, so that the system has reached thermal equilibrium. The internal heat has found one or more thermal paths from the heat source to the ultimate heat sink. Usually, all three methods of heat transfer—conduction, convection, and radiation—are involved. When the thermal equilibrium condition has been reached, the rate of heat being transferred by each of the three methods remains constant. The temperature gradients are now fixed with the heat flowing from the hotter parts of the system to the cooler parts of the system, until the heat finally reaches the ultimate sink. These characteristics indicate that the system has reached the steady state heat transfer condition. Steady state conditions may develop in a matter of minutes for small components such as transistors and diodes. However, for large electronic consoles, it may take a full day of operation before steady state heat transfer conditions are reached.

SEC et al. v. MRI
SEC Exhibit 1013.0018
IPR2023-00221

1.4 TRANSIENT HEAT TRANSFER

When the rate of heat flow changes within an electronic system, it will normally produce a temperature change somewhere in that system. Also, when there is a temperature change within an electronic system, there will normally be a change in the heat flow rate somewhere in the system. Changes of these types are defined as transient heat transfer conditions, because the thermal equilibrium of the system is unbalanced. Transient heat transfer conditions develop, for example, when the power is first turned on in an electronic system. As the current flows through the electronic components, heat is generated and the temperatures within the components begin to rise, resulting in a transient, or changing condition.

Transient conditions will also occur in an electronic system when it is subjected to temperature cycling tests. Consider a system sitting in an environmental chamber where the ambient temperature is slowly being cycled between −54°C (−65°F) and +71°C (+160°F). In this case, the outside temperature often increases more rapidly than do the temperatures within the electronic box. Heat then flows from the outside of the box toward the interior, because heat always flows from the hot body to the cold body.

A satellite in orbit around the earth experiences transient heat transfer due to the constantly changing angle with respect to the sun and the earth. The intensity of the solar radiation may be constant, but the heat absorbed will vary along the surface because the angle the sun makes with respect to the surface is changing.

Sometimes it is necessary to use an auxiliary cooling device or technique for a short period, until the regular cooling system is available to take over the job. Consider the case where a missile is carried under the wing of an airplane. An auxiliary cooling cart is available to supply cooling to the electronic system within the missile while the airplane engine is being started and checked. The missile electronic system is normally cooled by the ram air during the captive flight phase and during the free flight phase after the missile is released from the airplane. No cooling air is provided for the missile electronics during the taxi and takeoff period, because of the extra weight and cost. Instead, the electronics must rely upon the thermal capacity or thermal inertia of the system to absorb the heat without developing excessive temperatures during this period. When a number of airplanes are lined up, waiting to take off, delays of 30 min may occur. This may cause the electronics to overheat. If the weight of the electronic system is increased, it will increase the thermal inertia and permit cooler operation for longer periods of time. For higher power systems, however, a very large mass may be required to keep the electronics cool for 30 min, so that a more sophisticated technique may be required.

Sometimes it is desirable to use the change of state from a solid to a liquid, or from a liquid to a gas, to absorb heat. A large amount of heat can

SEC et al. v. MRI
SEC Exhibit 1013.0019
IPR2023-00221

be absorbed under these conditions. It is often possible to use hollow wall construction for the electronic chassis, which could be filled with wax that melts at a predetermined temperature. The change of state from a solid to a liquid may absorb enough heat to permit the electronics to survive a 30 min period with no cooling air. Once the airplane is flying, the ram air cools the wax, which returns to the solid state. If the missile is not fired, the melting wax permits the delaying cycle to be repeated over and over again.

1.5 ELECTRONIC EQUIPMENT FOR AIRPLANES, MISSILES, SATELLITES, AND SPACECRAFT

Electronic boxes used in airplanes, missiles, satellites, and spacecraft often have odd shapes that permit them to make maximum use of the volume available in odd-shaped structures. An odd-shaped box may require more time to design, because it is usually more difficult to provide the circuit cards with an efficient heat flow path, regardless of the cooling method used.

The trend in military and commercial airplanes and helicopters is toward a series of several standard sizes for plug-in types of electronic boxes that fit in racks. These are called ATR (air transport rack) boxes. They are of various widths, which are known as one quarter, one half, three quarters, and full width, each with a short and a long length. The electrical interface connectors are often at the rear of the box, with quick release fasteners at the front [1].*

Many of the electronic boxes are cooled by forced convection with bleed air from the jet engine compressor section. Since this air is at a high temperature and pressure, it is throttled (passed through the cooling turbine), cooled, and dried with a water separator before it is used. This air often enters the electronic box at the rear, adjacent to the electrical connectors. Rubber gaskets are used around the inlet ports at the air interface to provide an effective plug-in connection, which reduces the leakage at the cooling air interface.

Sometimes the conditioned cooling air is not completely dry because of excessive moisture in the air from humidity or a rainstorm. Small drops of water will often be carried into the electronics section together with the cooling air. If this water accumulates on PCBs or their plug-in connectors, electrical problems may develop. Therefore, many specifications do not permit external cooling air to come into direct contact with electronic components or circuits.

Air cooled heat exchangers, commonly called air cooled cold plates, which are being used more and more in airplanes, provide conditioned air for cooling the electronics. These heat exchangers are usually dip brazed

* *Numbers in brackets refer to references at the end of the book.*

SEC et al. v. MRI
SEC Exhibit 1013.0020
IPR2023-00221

when many thin [0.006 to 0.008 in (0.15 to 0.20 mm)] aluminum plate fins are used. Pin fin aluminum castings are becoming very popular because of their low cost. There is usually a slight weight increase with pin fins because the walls and fins have to be thicker to permit the molten aluminum to flow [2, 3].

Electronic systems for missiles generally have two cooling conditions to consider, captive and free flight. If the missile flight duration is relatively short, the electronics can be precooled during the captive phase so that the system can function with no additional cooling during the flight phase. The electronic support structure would act as the heat sink, soaking up the heat as it is generated, to permit the electronic system to function during the free flight phase.

Some missiles, such as the Cruise missiles, have a very long free flight phase, so that the cooling system must be capable of cooling the electronics for several hours. If ram air is used at speeds near Mach 1, the ram temperature rise of the cooling air may exceed 100°F (55°C). Since Cruise missiles fly at low altitudes, where the surrounding ambient air temperatures can be as high as 100°F, the cooling air temperatures could reach values of 200°F (93°C) even before the cooling process begins. Since the maximum desirable component mounting surface temperature is about 212°F (100°C), the outside ambient air cannot be used directly for cooling.

Cruise missiles must carry a large supply of fuel for their long flights. The fuel is often pumped through liquid cooled cold plates to provide cooling for the electronics. Toward the end of the flight mission, when the fuel supply runs low, the temperatures may increase. At this point it may be necessary to use the thermal inertia in the electronics structure to keep the system cool enough to finish its flight.

Electronic systems for satellites and spacecraft generally rely upon radiation to deep space for all their cooling. Deep space has a temperature of absolute zero, −460°F or 0°Rankine (−273°C or 0°Kelvin). Temperatures this low can provide excellent cooling if the proper surface finishes are used [4].

Special surface finishes and treatments may be required for satellites and spacecraft to prevent them from absorbing large quantities of heat from the sun. This heat may be direct solar radiation plus solar radiation reflected from the various planets and their moons (reflected radiation is called albedo) [4, 5].

Liquid cooled cold plates are often used to support electronic systems. Pumps then circulate the cooling fluid from the cold plates, where the heat is picked up, to the space radiators, where the heat is dumped to space.

Conduction heat transfer is used extensively for cooling electronic equipment in space environments. In the hard vacuum conditions of outer space, flat and smooth surfaces must be utilized with high contact pressures, to minimize the temperature rise across each interface. Although air is not normally considered to be a good heat conductor, its presence will sharply

SEC et al. v. MRI
SEC Exhibit 1013.0021
IPR2023-00221

reduce the contact resistance at most interfaces. Thermal greases are sometimes used to reduce the interface resistance in hard vacuum environments, by filling the small voids that would otherwise develop when the air is evacuated.

Air cooling can still be provided in a hard vacuum environment if a sealed and pressurized box is used. An internal fan can be used to circulate internal cooling air through a liquid cooled cold plate, which would carry away the heat. The heat from the fan must be added to the total heat load of the system. Also, a sealed box will have a large pressure differential across the surfaces of the box in the vacuum of outer space. The outer surfaces of the box must therefore be thick and stiff to resist excessive deformations and stresses caused by these pressures.

Natural convection cannot be used to transfer heat in satellites and spacecraft electronics. Natural convection requires a gravity field to permit the heated air to rise, because of its reduced density. In satellites and spacecraft, the effects of gravity are neutralized by the velocity and the continuous free fall characteristics of the flight path. Therefore, only radiation, conduction, and forced convection (in sealed boxes) should be considered for cooling electronic systems in space.

1.6 ELECTRONIC EQUIPMENT FOR SHIPS AND SUBMARINES

Large cabinets, consoles, or enclosures are normally used to support the electronic equipment used on ships and submarines. These cabinets are usually heavy and rugged, to provide protection for the electronics during storms and rough seas. Some cabinets may dissipate more than 2 kilowatts of heat, so liquid cooled cold plates and heat exchangers are often used to cool the equipment. The electronic components are often mounted on panels and sliding drawers. Panels are used to support displays, and drawers are used to support the heavy power supply units. Completely enclosed cabinets are often used by the Navy with special radio frequency interference (RFI) and electromagnetic interference (EMI) gaskets on heavy doors. This protects and shields the electronics from undesirable radiated and conducted electromagnetic waves, which can interfere with the operation of the equipment. Because these systems are completely enclosed, cooling problems may become quite severe [6].

Water is usually available on ships and submarines, so that it is natural to utilize it for cooling. Water cooled heat exchangers are often used, with external fins to permit cooling with forced air. Fans are used to force the air through the heat exchanger fins to cool the air, which is then circulated through the console. This type of forced convection cooling can be used with both closed loop and open loop systems. With a closed loop system, a cooling air supply plenum and a return plenum may be established within the side walls of the consoles. The side walls are often several inches deep,

SEC et al. v. MRI
SEC Exhibit 1013.0022
IPR2023-00221

with ribs to provide rigidity from high shock loads, so that they can easily carry the cooling air to and from the electronics. If an open loop system is used for cooling, the air entering the console would be forced through the water cooled heat exchanger, which is usually at the base of the console. The conditioned air would be circulated through the electronics and then exhausted at the top of the console.

When the heat loads are not too high, natural convection techniques can often be used to cool the electronics. This works well on tall cabinets, which can use chimney effects to force the air through the system without the use of fans or pumps. Air enters at the bottom of this cabinet, where it first picks up heat from the electronics. The warmer air has a reduced density, so that it starts to rise through the chassis, picking up more heat as it rises. The cooling air finally exits at the top of the cabinet.

Large open spaces must be provided in a chassis that is cooled by natural convection. The flow resistance must be minimized to permit the cooling air to pass freely through the cabinet. This requires a low packaging density, with very few turns in the air flow path, to ensure the proper cooling air flow [7, 8].

Often designers package electronic equipment in a console and completely ignore the cooling air flow path. A large fan is then placed at the bottom of the unit to "blow air at the electronics," with the hope that somehow, somewhere, the air will find a path to the hot spots so that they will be adequately cooled. This brute force technique is usually justified on the grounds that it has been used many times by many people and it "seems to work." Needless to say, this is a poor policy, because there is no assurance that the blowing air will ever reach the spots that require cooling, and there is no way to determine what the cooling effects will be. It is best to plan ahead so that the air can be directed to the most critical electronic components for more efficient cooling. This will result in a smaller, lighter, and less expensive electronic system, with higher reliability.

1.7 ELECTRONIC EQUIPMENT FOR COMMUNICATION SYSTEMS AND GROUND SUPPORT SYSTEMS

Communication systems and ground support systems must both be capable of continuous operation for extended periods of time at high altitudes, in hot desert areas, in arctic areas, and in rain, sleet, and snow. Large systems have their electronics completely enclosed within shelters that resemble small barns and can hold several people for long periods of time. Small systems are often enclosed in transit cases, which may be carried on the back seat of a Jeep type of vehicle for rapid mobility. Small panel trucks are very popular for transporting moderate sized units.

Communication shelters are used all over the world by military personnel and by commercial television networks, to transmit and receive all forms

SEC et al. v. MRI
SEC Exhibit 1013.0023
IPR2023-00221

of data. The shelters are usually lined from wall to wall with large electronic control consoles that can easily dissipate many kilowatts of heat.

Such shelters are insulated to protect them from both heat and cold. In hot climates the shelters are often equipped with exhaust fans to flush out hot air that has been trapped. This is important because the electronic consoles within the shelters use the local ambient air for cooling. This is done using forced convection or natural convection, depending upon the power dissipation and the location of the console. If the internal ambient temperatures within the shelter become too hot, refrigeration units may be provided, with auxiliary power units, to keep the shelters cool.

Ground support systems are generally used to perform functional checks of the electronic equipment for airplanes, missiles, ships, submarines, trains, trucks, and automobiles to ensure that they are operating properly. If malfunctions are discovered, replacements can be made to ensure successful completion of the mission or trip.

Ground support equipment is also used to supply auxiliary power or auxiliary cooling for electronic equipment that requires temporary boosters. These units could contain gasoline powered engines, with generators and regulated power supplies or large batteries as the power source.

Lightweight controls and displays are generally supported by light, dripproof, sheet metal enclosures. If heavy items such as large transformers and power supply components are required, they are mounted at the bottom of a rugged electronic enclosure, to reduce the dynamic loads that may be developed in a shock and vibration environment.

1.8 MINICOMPUTERS, MICROCOMPUTERS, AND MICROPROCESSORS

The computer industry has been changing very rapidly since the introduction of miniature semiconductor silicone chips. The first chips, which were introduced in about 1962, contained only about 15 to 20 diodes, transistors, and resistors on a substrate that measured about 0.150 by 0.200 in (0.381 by 0.508 cm). Component densities have since increased sharply, so that the same size chip can now incorporate over 30,000 components, and the costs have dropped just as fast. Large computers utilizing discrete diode, transistor, and resistor components were soon replaced with much smaller machines which had the same power at a much lower cost because they utilized the new semiconductor chip technology. These new, smaller machines became known as minicomputers because they were so small compared to the older and larger machines but had the same capability.

Minicomputers have found exciting new applications in many areas. Their size, flexibility, reduced costs, and improved memory storage have permitted more small businesses to make use of them. The power requirements of these machines have also dropped, so that exotic and expensive techniques are not required to cool the electronics.

SEC et al. v. MRI
SEC Exhibit 1013.0024
IPR2023-00221

Microcomputers are even smaller than minicomputers. A typical minicomputer will occupy one drawer of a filing cabinet. A typical microcomputer will fit on one plug-in PCB about 5 × 7 in (12.70 × 17.78 cm). Microcomputers are not as fast as minicomputers and are generally used where flexibility, size, and cost are more important than speed.

Microcomputers have been replacing systems that utilize custom-made large scale integrated circuits (LSIs) and hybrid circuits. Although these circuits have a high density and low power dissipation, they lack flexibility and cannot be modified or adapted for different applications.

Both the minicomputers and the microcomputers require some type of central processing unit (CPU) to control input and output, to perform mathematical operations, to decode, and to move information in and out of the memory. This CPU, which would normally require the mounting surface area of one plug-in PCB, can now be placed on a single small chip. This chip is called a microprocessor, and it is the most expensive part of the microcomputer.

Microprocessors are available in rectangular cases about 2.5 × 0.75 × 0.20 in (6.35 × 1.905 × 0.508 cm) with about 40 external wires, which perform all of the CPU functions. Microprocessors must be used with memory systems, which can also be expensive. Many memory systems use dual inline package (DIP) components, which are fabricated on silicon chips in much the same way that microprocessors are made. The semiconductor memories require less power than do core memories. However, core-type memories are able to operate at higher temperatures.

Floppy disk memories are rapidly becoming available with very large capacities. These thin disks are made of a soft or floppy material (hence the name) that is coded magnetically on one or both sides and enclosed in a square cardboard cover. These disks provide a large amount of memory at a very low price.

Bits are recorded on the surface of the rotating disk with a movable head. The surface of the disk is divided into sectors, shaped like a slice of pie, and tracks, shaped like concentric rings. The disk requires 7 to 15 watts of power for its operation. Many systems use two disks because the drive mechanism for two disks is only slightly more expensive than that for one disk.

Cassette tape memories are quite common in low cost systems. The cassette memory is the cheapest mass memory available. This type of memory can store large amounts of information, but the system is very slow and usually must be turned on and off manually.

1.9 COOLING SPECIFICATIONS FOR ELECTRONICS

A number of government and civil agencies issue documents that relate to the cooling of electronic equipment. A few of these are described below.

SEC et al. v. MRI
SEC Exhibit 1013.0025
IPR2023-00221

1. *MIL-E-16400 (Navy) Electronic, Interior Communication and Navigation Equipment, Naval Ship and Shore, General Specification.* This specification covers the general requirements applicable to the design and construction of electronic interior communication and navigation equipment intended for naval ship or shore applications. The specification defines the environmental conditions within which the equipment must operate satisfactorily and reliably. This includes the selection and application of materials and parts and means by which the assembled system will be tested to ensure that it is acceptable to the U.S. Navy.

2. *MIL-STD-202, Test Methods for Electronic and Electrical Component Parts.* This specification establishes uniform methods for testing electronic and electrical component parts. This includes basic environmental tests to determine resistance and capability of withstanding natural elements and conditions surrounding military operations. The specification includes items such as capacitors, resistors, switches, relays, transformers, and jacks. Only small parts are considered in this specification. The maximum weight of transformers and inductors is limited to 300 pounds or having a rms test voltage up to 50,000 volts.

3. *MIL-STD-810, Environmental Test Methods.* This document describes all test methods of a similar character that have appeared in the various joint or single service specifications. It also describes newly developed test methods that are feasible for use by more than one service.

4. *MIL-E-5400, Electronic Equipment, Airborne, General Specification.* This specification covers the general requirements for airborne electronic equipment for operation primarily in piloted aircraft. It includes various temperature and altitude conditions for different classes of equipment, with details on selecting materials, components, and fasteners.

5. *MIL-STD-781B, Reliability Tests: Exponential Distribution.* This standard outlines test levels and test plans for reliability qualification, production acceptance, and longevity tests. Included are temperature cycling tests for isolating weak components and poor designs.

1.10 SPECIFYING THE POWER DISSIPATION

Power dissipations in electronic systems should be evaluated carefully, because the reliability and the mean time between failures (MTBF) can be sharply reduced by excessive component temperatures. It is better to be slightly conservative (slightly higher) in estimating power dissipations, to provide for future growth. Also, some compensation must be made for the normal tolerance variations in the component ratings and in the line voltages, so that a little safety factor should be added to the power dissipation in the system.

A safety factor will also provide a cushion to fall back upon when it is suddenly discovered that some critical components will not be available for

SEC et al. v. MRI
SEC Exhibit 1013.0026
IPR2023-00221

production, or that substitute components will dissipate 10% more power. It is very expensive to change the cooling technique in a system after the design has been completed.

It is possible, of course, to be too conservative and to specify power dissipations that are far higher than would normally be expected. This condition will result in added expense, weight, and size. In some cases being too conservative may result in the use of a large fan when none is needed. A large and heavy liquid cooling system may be installed instead of a simple fan because someone was too conservative in specifying the power dissipation.

Some contracts may require a thermal analysis to be completed 3 months before the qualification test program. In order to prove the validity of the analysis, the contract may require the analysis to agree within ±10% of the actual temperatures achieved during the qualification test. This is a difficult situation to deal with, because the thermal design may be based upon a worst-case duty cycle or test condition, where high line voltages and component tolerance variations have been included in the total power dissipation. Higher power dissipations will result in higher component temperatures. During the actual qualification test, on the other hand, a completely different power dissipation may be used because a different duty cycle may be specified for the tests. This may result in poor agreement between the analysis and the tests [9].

It may be necessary to investigate many different sets of conditions to prove the thermal design of an electronic system. For example, a high altitude, high temperature environment may produce the highest temperatures in the logic section of the system rather than in the power supply section. Instead, a high altitude, low temperature environment may be the most severe for power supplies, because this condition draws the maximum heater power for the warm-up period. High current demands, even for short periods, may create hot spots in the electronic components that control the power [9].

Many electronic systems require special test sets to simulate test conditions similar to those that will be experienced in actual field operation. These test sets may be designed to exercise the electronic systems in many different ways, depending upon the manner in which the equipment designers interpret the test specifications. This can result in a test power dissipation that is completely different from the dissipation experienced in actual field operation. Therefore, it is important to make sure that the test equipment designers and the electronic system designers agree on the power requirements. This will reduce the chances of having a big difference in the design power dissipation and the test power dissipation.

The engineers responsible for determining the power dissipation in an electronic system should take the time required to determine these values accurately. The power dissipations may dictate the methods of heat removal, which, in turn, will determine the hot spot component temperatures.

SEC et al. v. MRI
SEC Exhibit 1013.0027
IPR2023-00221

It is worth the extra time required for a more accurate estimate of the power dissipation, because the reliability, size, weight, and cost of the system may depend upon it.

1.11 DIMENSIONAL UNITS AND CONVERSION FACTORS

The metric system for measuring is rapidly gaining acceptance in the United States, but the English system is still used as the standard by industry, by most lay persons, by many engineers. Both sets of dimensional units are therefore used in this book to permit engineers to use the system that is more convenient for them and to permit them to convert easily from one system to the other.

Power dissipation, or heat (Q), for electronic systems is expressed in three standard sets of units: watts, British thermal units per hour (Btu/hr), and calories per second (cal/sec). The watt is the standard unit of heat and power measure for electronic equipment in most countries throughout the world, including the United States.

The British thermal unit (Btu) is the quantity of heat required to raise the temperature of 1 pound mass of water 1 degree Fahrenheit (1°F) at standard pressure [10].

The calorie (cal) is the quantity of heat required to raise the temperature of 1 gram mass of water 1 degree Celsius (1°C) at standard pressure. This is sometimes written as gram-calorie (g-cal), which is called the small calorie [10].

A consistent set of units is always used in this book. When the unit of heat is the Btu, the length dimension will be feet (ft) and the time unit will be hours (hr). When the unit of heat is the calorie, the length dimension will be centimeters (cm) and the time unit will be seconds (sec).

Conversions can be made between the various units with the use of Table 1.1.

Table 1.1 Power/Heat Rate Conversions[a] Q

Multiply Number of → / By → / to Obtain ↓	watts	Btu/hr	cal/sec
watts	1	0.293	4.187
Btu/hr	3.413	1	14.285
cal/sec	0.239	0.070	1

[a] *1 watt = 1 joule/sec.*
1 Btu = 1055 joules = 252 cal.

SEC et al. v. MRI
SEC Exhibit 1013.0028
IPR2023-00221

For example, to convert 10 Btu/hr to cal/sec using Table 1.1:

$$\left(10\,\frac{\text{Btu}}{\text{hr}}\right)(0.070) = 0.70\,\frac{\text{cal}}{\text{sec}} \tag{1.1}$$

Without the table, using dimension analysis,

$$\left(10\,\frac{\text{Btu}}{\text{hr}}\right)\left(225\,\frac{\text{cal}}{\text{Btu}}\right)\left(\frac{\text{hr}}{3600\ \text{sec}}\right) = 0.70\,\frac{\text{cal}}{\text{sec}} \tag{1.2}$$

To convert 30 watts to Btu/hr using Table 1.1:

$$(30\ \text{watts})(3.413) = 102.39\,\frac{\text{Btu}}{\text{hr}} \tag{1.3}$$

Thermal conductivity is expressed in English units and in metric units.

$$\text{English units:}\quad K = \frac{\text{Btu}}{\text{hr ft}^2\ ^\circ\text{F/ft}} = \frac{\text{Btu ft}}{\text{hr ft}^2\ ^\circ\text{F}} = \frac{\text{Btu}}{\text{hr ft}\ ^\circ\text{F}}$$

$$\text{Metric units:}\quad K = \frac{\text{g cal}}{\text{sec cm}^2\ ^\circ\text{C/cm}} = \frac{\text{cal cm}}{\text{sec cm}^2\ ^\circ\text{C}} = \frac{\text{cal}}{\text{sec cm}\ ^\circ\text{C}}$$

Conversions can be made between English units and metric units using Table 1.2.

Table 1.2 Thermal Conductivity Conversions *K*

Multiply Number of → / By → / to Obtain ↓	$\frac{\text{Btu ft}}{\text{hr ft}^2\ ^\circ\text{F}}$ or $\frac{\text{Btu}}{\text{hr ft}\ ^\circ\text{F}}$	$\frac{\text{cal cm}}{\text{sec cm}^2\ ^\circ\text{C}}$ or $\frac{\text{cal}}{\text{sec cm}\ ^\circ\text{C}}$
$\frac{\text{Btu}}{\text{hr ft}\ ^\circ\text{F}}$	1	241.9
$\frac{\text{cal}}{\text{sec cm}\ ^\circ\text{C}}$	0.00413	1

For example, to convert 88 Btu/hr ft °F to cal/sec cm °C using Table 1.2:

$$\left(88\,\frac{\text{Btu}}{\text{hr ft}\ ^\circ\text{F}}\right)(0.00413) = 0.363\,\frac{\text{cal}}{\text{sec cm}\ ^\circ\text{C}} \tag{1.4}$$

SEC et al. v. MRI
SEC Exhibit 1013.0029
IPR2023-00221

The coefficient of heat transfer is expressed in English units and metric units in Table 1.3.

$$\text{English units:} \quad h = \frac{\text{Btu}}{\text{hr ft}^2\,^\circ\text{F}}$$

$$\text{Metric units:} \quad h = \frac{\text{cal}}{\text{sec cm}^2\,^\circ\text{C}}$$

Other conversion values are given in Tables 1.4 through 1.11.

Table 1.3 Coefficient of Heat Transfer Conversions *h*

Multiply Number of → / By → / to Obtain ↓	$\frac{\text{Btu}}{\text{hr ft}^2\,^\circ\text{F}}$	$\frac{\text{cal}}{\text{sec cm}^2\,^\circ\text{C}}$
$\frac{\text{Btu}}{\text{hr ft}^2\,^\circ\text{F}}$	1	7373
$\frac{\text{cal}}{\text{sec cm}^2\,^\circ\text{C}}$	0.0001356	1

Table 1.4 Viscosity Conversions[a] μ

Multiply Number of → / By → / to Obtain ↓	$\frac{\text{lb}}{\text{ft hr}}$	$\frac{\text{g}}{\text{cm sec}}$ (poise)	$\frac{0.01\ \text{g}}{\text{cm sec}}$ (centipoise)
$\frac{\text{lb}}{\text{ft hr}}$	1	242	2.42
$\frac{\text{g}}{\text{cm sec}}$ (poise)	0.00413	1	100
$\frac{0.01\ \text{g}}{\text{cm sec}}$ (centipoise)	0.413	0.01	1

[a] *English units: μ = lb/ft hr; metric units: μ = g/cm sec = poise (100 centipoise = 1 poise).*

SEC et al. v. MRI
SEC Exhibit 1013.0030
IPR2023-00221

Table 1.5 Specific Heat Conversions *Cp*

Multiply Number of → / By → / to Obtain ↓	$\frac{\text{Btu}}{\text{lb °F}}$	$\frac{\text{cal}}{\text{g °C}}$
$\frac{\text{Btu}}{\text{lb °F}}$	1	1
$\frac{\text{cal}}{\text{g °C}}$	1	1

Table 1.6 Density Conversions ρ

Multiply Number of → / By → / to Obtain ↓	$\frac{\text{lb}}{\text{ft}^3}$	$\frac{\text{lb}}{\text{in}^3}$	$\frac{\text{g}}{\text{cm}^3}$
$\frac{\text{lb}}{\text{ft}^3}$	1	1728	62.43
$\frac{\text{lb}}{\text{in}^3}$	0.000579	1	0.036
$\frac{\text{g}}{\text{cm}^3}$	0.0160	27.68	1

SEC et al. v. MRI
SEC Exhibit 1013.0031
IPR2023-00221

Table 1.7 Pressure Conversions

Multiply Number of → / By → / to Obtain ↓	$\frac{lb}{in^2}$	in H_2O	$\frac{g}{cm^2}$	mm Hg
$\frac{lb}{in^2}$	1	0.0361	0.0142	0.0193
in H_2O	27.7	1	0.394	0.535
$\frac{g}{cm^2}$	70.37	2.538	1	1.361
mm Hg	51.71	1.869	0.735	1

Table 1.8 Weight Conversion

Multiply Number of → / By → / to Obtain ↓	lb	g	kg
lb	1	0.00220	2.205
g	453.6	1	1000
kg	0.4536	0.001	1

SEC et al. v. MRI
SEC Exhibit 1013.0032
IPR2023-00221

Table 1.9 Length Conversion

Multiply Number of → / By → / to Obtain ↓	in	ft	cm
in	1	12	0.3937
ft	0.0833	1	0.0328
cm	2.54	30.48	1

Table 1.10 Area Conversions

Multiply Number of → / By → / to Obtain ↓	in^2	ft^2	cm^2
in^2	1	144	0.155
ft^2	0.00694	1	0.00107
cm^2	6.452	929	1

SEC et al. v. MRI
SEC Exhibit 1013.0033
IPR2023-00221

Table 1.11 Volume Conversions

Multiply Number of → / By → / to Obtain ↓	in^3	ft^3	cm^3
in^3	1	1728	0.0610
ft^3	5.787×10^{-4}	1	3.531×10^{-5}
cm^3	16.39	2.832×10^4	1

Figure 1.4 shows an investment cast aluminum chassis with the rear cover removed.

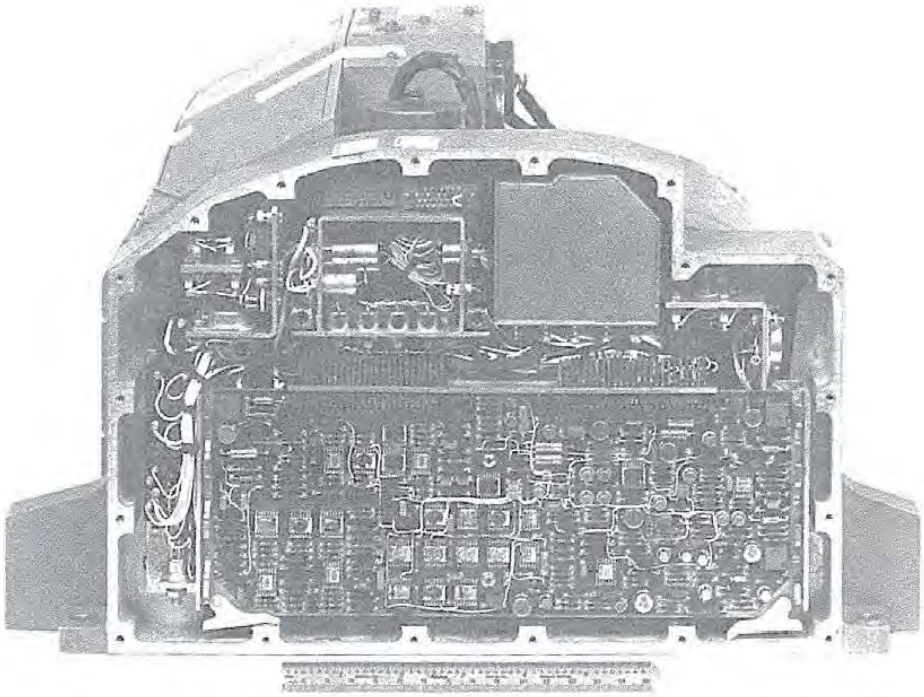


Figure 1.4 Investment cast electronic box, where the temperatures are controlled by conduction of heat to air cooled pin fin heat exchangers in the side walls. (Courtesy Litton Systems, Inc.)

SEC et al. v. MRI
SEC Exhibit 1013.0034
IPR2023-00221

2

Designing the Electronic Chassis

2.1 FORMED SHEET METAL ELECTRONIC ASSEMBLIES

Electronic systems normally consist of many different discrete electronic component parts, such as resistors, capacitors, diodes, transistors, hybrids, DIPs, LSIs, microprocessors, and transformers, which are enclosed within a support structure called the chassis. Power is dissipated by these different components, because the electronic system is not 100% efficient. Any power that is dissipated is rejected in the form of heat. This heat must be removed to prevent excessive temperatures from developing in these electronic components.

The purpose of the electronic chassis is to support the components while providing a low resistance thermal path to a heat sink, which will absorb this waste heat with a minimum rise in the temperature of the components. The heat sink may be the ambient air surrounding the chassis or a liquid cooled cold plate that is an integral part of the chassis wall.

Whatever method of heat transfer is selected, the technique should be as simple and as cost effective as possible. Many factors will have to be considered, such as the space available, the power requirements of the cooling system, the maximum allowable component temperatures, component sizes, power densities, and the heat sink temperature. Other factors, such as shock and vibration, may have to be considered together with the thermal environment to ensure an adequate chassis design [1, 11, 12].

Sheet metal structures are often used for many different types of electronic boxes because the manufacturing costs are so low. Thin gauge aluminum or steel sheets can be blanked and formed into a lightweight chassis with the use of rivets, spot welding, or arc welding. The final assembly is usually painted for protection and appearance. This type of construction can be used for a chassis 7 or 70 in high.

Light gauge steel sheet metal structures are usually not suitable for cooling high power electronic systems by means of conduction. The cross section areas are small because the metal is thin and the thermal conduct-

SEC et al. v. MRI
SEC Exhibit 1013.0035
IPR2023-00221

ance is low. This increases the thermal resistance, which also increases the component hot spot temperatures. Also, light gauge sheet metal structures are not generally capable of withstanding high vibration and shock levels, so that their use in these environments is very limited.

Lightweight sheet metal structures may tend to amplify any acoustic noise generated by cooling fans or pumps. If large, thin flat panels are used on an electronic box that is to be fan cooled, make sure that it is not too close to workers, who may object to the acoustic noise that is generated.

Structural epoxies are being used very successfully for assembling small electronic boxes. These epoxies have a high peel and shear strength, with a short cure time at 212°F (100°C), which makes them very cost effective. Lap joints with large surface areas must be used to obtain the necessary strength at the bonded interfaces.

Maintainability has recently become a big factor in the design of electronic equipment. Quick access is usually required to fault-isolate an electrical malfunction and to provide some means for a quick repair. This has resulted in the extensive use of plug-in types of circuit boards which can be removed and replaced very easily. These circuit boards are generally used to support small electronic components that do not dissipate a great deal of power. Large, heavy, or high power dissipating components are often mounted on the chassis walls with special support brackets.

A typical chassis would have two sheet metal dust covers on opposite sides of the chassis. One cover would provide access to the plug-in PCBs; the cover on the opposite side would provide access to the wiring harness or the master interconnecting board.

2.2 DIP BRAZED BOXES WITH INTEGRAL COLD PLATES

Dip brazed aluminum boxes are convenient to use for small, lightweight systems when the quantities are small, usually less than about eight or ten boxes. If the quantities are much larger, it is generally more cost effective to use aluminum investment castings or plaster mold castings, even though they require special tooling and pattern costs. Most electronic boxes will require shelves and brackets for mounting electronic components, ribs for stiffening the chassis to resist vibration, and cutouts for the cables and electrical harness. A dip brazed electronic chassis can often provide these features at a relatively low cost.

The size of the dip brazed assembly is usually limited by the size of the dip brazed tank, or salt bath, which is used to completely submerge the structure that is to be brazed. The individual parts of the chassis are joined together like a three dimensional jigsaw puzzle. Sometimes, a stainless steel fixture is used to hold the individual parts together during the dip brazing process. Sometimes the parts are tack welded, riveted, or screwed together with aluminum screws. An aluminum slurry or a brazing strip, with a melting

SEC et al. v. MRI
SEC Exhibit 1013.0036
IPR2023-00221

temperature slightly lower than the temperature of the salt bath, is used to join the individual structural members, which have a melting temperature slightly higher than that of the salt bath. As the slurry melts, capillary action draws the molten aluminum into the small voids between the individual parts and joins them rigidly.

Sometimes a structure may be too big to dip braze in one piece; in this case two separate dip brazed subassemblies are made. These can be joined later with screws and structural epoxy.

Thin plate fin types of heat exchangers are often used for the side walls of a chassis. The heat exchangers (or cold plates) are often dip brazed first as a subassembly and then cemented together to form a chassis. Sometimes the heat exchangers can be dip brazed as an integral part of the chassis without first forming a subassembly. The multiple fins provide a large surface area, which sharply increases the amount of heat that can be removed from the chassis with air or with liquids. A typical cross section through a chassis with finned side wall heat exchangers is shown in Fig. 2.1.

The plate fin material is usually about 0.006 in (0.0152 cm) or 0.008 in (0.0203 cm) thick so that many fins can be spaced close together. The typical spacing is about 14 to 18 fins per inch. Some companies can dip braze as many as 23 to 25 fins per inch.

When the fins get very close together, it becomes very difficult to clean out the salts left from the dip brazed tank, so that they may become trapped. This can block the fin passages and reduce the cooling effectiveness. Also, trapped salts can corrode the metal and weaken the structure.

It is desirable to have some method for checking the dip brazed heat exchangers to make sure that the fins are not blocked. A visual check is very valuable. If it is possible to look down the heat exchanger, blocked fins can be spotted and either rejected or cleaned. If a visual check is not possible, a pressure drop check should be made with a known flow passing

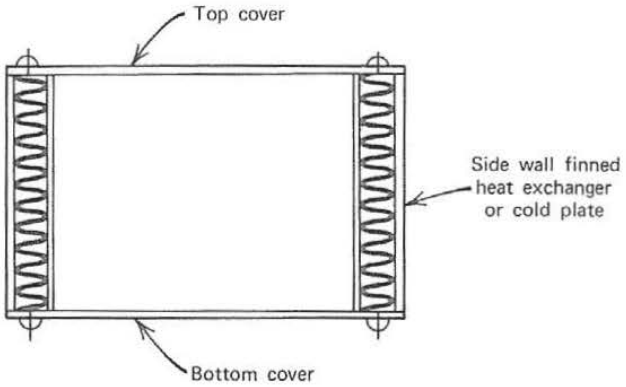


Figure 2.1 Chassis with side wall heat exchanger.

SEC et al. v. MRI
SEC Exhibit 1013.0037
IPR2023-00221

through the fins. If a blockage occurs, the pressure drop across the fins will be very high.

The fin material is generally very soft, so that it can be deformed easily. If fins in an air cooled cold plate are allowed to extend to the end of the chassis opening, they can become bent or deformed by foreign objects, such as bolts, nuts, screwdrivers, and even pencils. Therefore, for added protection, the ends of the fins should be recessed at least $\frac{1}{4}$ in.

2.3 PLASTER MOLD AND INVESTMENT CASTINGS WITH COOLING FINS

Plaster mold castings and investment castings are very popular for small electronic support structures or small electronic enclosures. Plaster mold castings and investment castings use plaster for the mold, into which a molten metal is poured. Typical metals are usually aluminum, magnesium, zinc, bronzes, and some steels. Wall thicknesses of 0.040 to 0.060 in (0.102 to 0.152 cm) can be readily obtained with both methods.

Investment piece part castings are generally slightly more expensive than plaster mold castings because an intermediate wax core is required. This core is made up in the exact detail required by the finished product. It is coated with several thin coats of plaster or refractory material and thoroughly dried. The coated core is heated to melt the wax, which is then drained, leaving the hollow mold. The mold is filled with molten metal while a vacuum is applied to remove tiny air pockets from the porous plaster or refractory outer shell. This permits the molten metal to fill in every small corner in the mold for excellent detail, surface finish, and accuracy. The outer shell must be broken away and destroyed to obtain the finished product.

Investment castings are somewhat limited in their size, depending upon the complexity. Intricate chassis castings 15 × 8 × 10 in (38.10 × 20.32 × 25.40 cm), with wall thicknesses of 0.070 in (0.178 cm), can be readily obtained.

Plaster mold casted piece parts are normally less expensive than investment castings because the process does not require the use of an intermediate disposable wax pattern. However, tooling costs for the plaster mold casting will be higher if permanent molds are used, because of the extra machining required.

Very large plaster mold castings can be made, up to 100 in (254 cm) in length if required, with considerable detail. However, plaster castings usually cannot produce small details as well as can investment castings.

Recent advances in casting technology have made it possible to produce thin wall aluminum castings that are pressure-tight for large molecule vapors such as nitrogen, or exotic liquids. The process, called the Antioch process, produces progressive freezing from the distant points in the casting toward

SEC et al. v. MRI
SEC Exhibit 1013.0038
IPR2023-00221

the risers. The process uses a plaster mold that has the permeability of a sand mold. Impregnation with an epoxy resin is not required to obtain an airtight box. This is important for applications that require electroplated nickel, electroless nickel, or some other superior plating process for superior corrosion resistance.

Investment castings and plaster castings methods can be used to make very efficient, lightweight heat exchangers and cold plates with integrally cast pin fins. Typical pin fins are about 0.062 in (0.157 cm) in diameter, 0.50 in (1.27 cm) long, and spaced on 0.200 in (0.508 cm) centers. Plate fin types of heat exchangers and cold plates are much more difficult to cast. Continuous plate fins require cores that must be supported as the molten metal is poured. These cores can shift and crack if the ribs are very long. If plate fins are desired, it might be better to cast them in short lengths instead of in a continuous length, to provide a means for supporting the cores.

Cemented construction techniques can often be combined with castings to provide a plate fin heat exchanger or cold plate. Extruded plate fin sections can be cemented to cast plate fin sections to provide a plate fin heat exchanger, as shown in Figure 2.2.

2.4 DIE CAST HOUSINGS

The die casting process is capable of providing the lowest cost piece parts, with high quality and excellent appearance. However, tooling costs are very high, tools take a long time to fabricate, and modifications or design changes are very expensive. Die castings should therefore not be considered for production runs of less than about 1000 piece parts. This requires long range planning, scheduling, and coordinating to ensure a satisfactory product.

For large production runs it may even be possible to use investment castings or plaster mold castings as a buffer until the die casting tools have been fabricated, installed, and proven out.

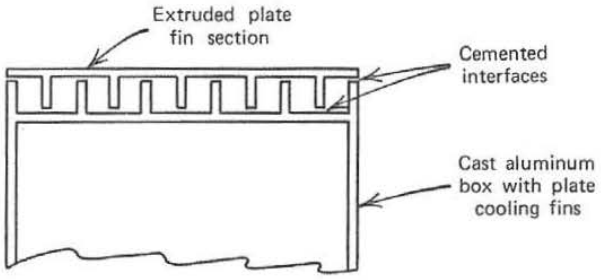


Figure 2.2 Plate fin extrusion cemented to a cast plate fin box to form a multiple fin heat exchanger.

SEC et al. v. MRI
SEC Exhibit 1013.0039
IPR2023-00221

2.5 LARGE SAND CASTINGS

Large electronic enclosures are often required for systems that must operate in severe vibration and shock environments. Rigid structures are required to withstand high acceleration levels, without fatigue failures, while providing an adequate support for electronic components. If weight is not too critical, fabrication costs can often be substantially reduced with the use of large aluminum sand castings. This type of cabinet is often suitable for ships and submarines, which must be capable of operating in rough seas during storms.

Mounting surfaces for the cabinet and for electronic subassemblies and components may have to be machined, because of poor casting tolerance control. However, with a little planning, the amount of machining can often be reduced to provide a cost effective structure.

2.6 EXTRUDED SECTIONS FOR LARGE CABINETS

Weight is often a problem with large cabinets that must be designed to withstand the Navy shock (MIL-S-901) and vibration (MIL-STD-167) requirements. Thick cast walls can provide the required rigidity, but with a high weight penalty. Under these circumstances extruded sections, with ribs or hollow cores, are capable of providing a rigid but relatively lightweight electronic enclosure. This type of structure can be welded, bolted, or riveted together to form a very rugged console. The hollow core type of extrusion is very convenient for ducting cooling air to various parts of the cabinet with fans or blowers. Large hollow core cross sections can carry large quantities of cooling air with a small pressure drop. Openings can be placed at various points in the extruded wall sections to direct the cooling air to hot spot areas. These openings can be blocked or reduced in size with plugs if the power distribution is changed at a later date.

Extruded side wall sections are convenient to use with closed forced air cooling systems, where a water cooled heat exchanger or a refrigeration unit cools the recirculated air. One vertical side wall can be used as the supply plenum, and the opposite side wall can be used as the return plenum, as shown in Figure 2.3.

2.7 HUMIDITY CONSIDERATIONS IN ELECTRONIC BOXES

Electronic equipment must often be capable of operating in very humid environments, where condensation will produce large amounts of water over a long period of time [13].

When high humidity environments are encountered, equipment designers must make a choice. They can seal the box against moisture or let the box

SEC et al. v. MRI
SEC Exhibit 1013.0040
IPR2023-00221

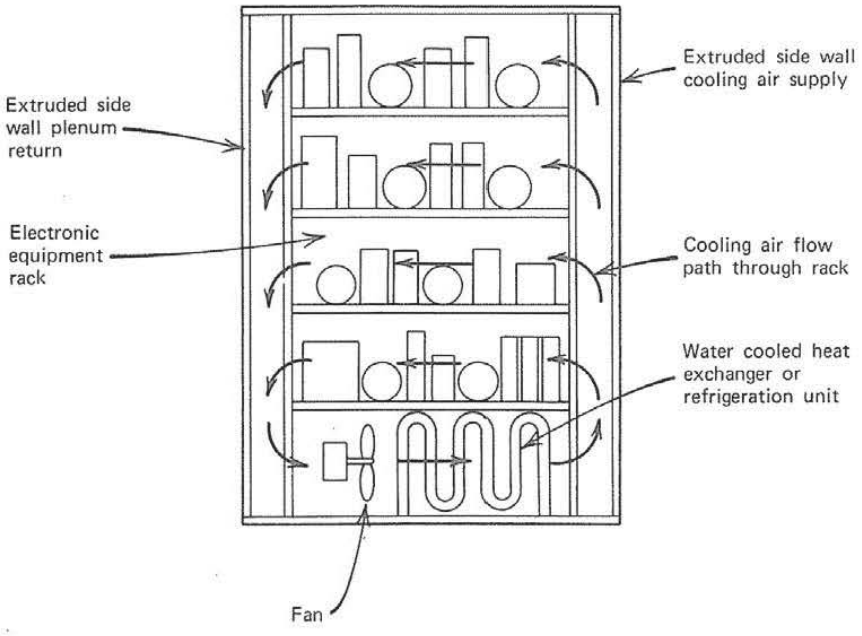


Figure 2.3 Closed cooling air recirculating system for electronic equipment rack.

breathe. Past experience with moisture problems shows that it is better to let the box breathe, because it is extremely difficult to seal an electronic box against moisture [14].

A sealed box is larger and heavier than a nonsealed box, especially if the system is required to operate at high altitudes. The walls of a sealed box must be capable of withstanding high forces due to high pressure differentials at high altitudes. The increased stiffness requirements will result in a weight increase and a size increase.

Humidity can cause serious problems in the electronic equipment when the internal circulating air is cooler than the outside ambient air. Moisture can condense on the electronic components, connectors, and circuit boards, producing short circuits or radical changes in the resistance between electronic components.

A moisture drainage path should be provided that permits the condensation to drain from the console walls, circuit boards, connectors, and wire harness to a drip pan at the bottom of the unit, where the moisture can evaporate or drain out of the system. Avoid moisture traps where the condensate can settle and cause corrosion. Drill holes, with a minimum diameter of 0.25 in, in the corners of horizontal bulkheads, away from the electronic components, to provide a moisture drainage path through the cabinet.

Use vertically oriented circuit boards and connectors, if possible, to

SEC et al. v. MRI
SEC Exhibit 1013.0041
IPR2023-00221

provide a natural moisture drainage path away from the boards to the bottom of the chassis.

Offset drain holes may be required in the base of the chassis to prevent foreign objects, such as screwdrivers, from being poked into these holes and causing internal damage. Drain holes similar to the one shown in Figure 2.4 will reduce the possibility of foreign objects entering the chassis.

2.8 CONFORMAL COATINGS

There are four popular types of conformal coatings: acrylic, epoxy, polyurethane, and silicone. A thin conformal coating 0.003 to 0.005 in (0.0076 to 0.0127 cm) thick may be required to protect the circuit boards from moisture. However, these coatings should be applied only if they are absolutely necessary. In general, coatings are expensive to apply. They require special cleaning processes, special tools to apply the coatings, and the circuit boards are difficult to rework. In addition, many coatings tend to crack, chip, and peel, and they will contaminate the connector contacts unless they are masked during the application. Also, water vapor tends to creep under the coatings and condense, so that coatings can change the electrical resistance between the circuits they are supposed to help. Therefore, if it becomes necessary to apply a conformal coating for moisture protection, make sure that the cleaning and application procedures are carefully followed, or the coatings may create more problems than they solve.

The conformal coat should not be applied so that it bridges the strain relief on the component lead wires. The purpose of the strain relief is to reduce stresses in the wire and in the solder joints. If the strain relief is bridged (completely filled), it will act like a short circuit and will not provide strain relief during vibration as the PCB flexes. Also, a filled strain relief will restrict the relative motion that results during temperature cycling tests. This will increase the stresses in the solder joints, which will increase the chance of failure.

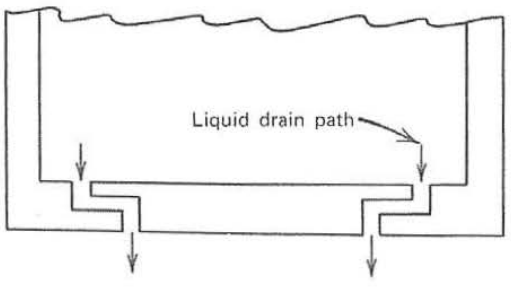


Figure 2.4 Offset drain holes in bottom of chassis.

SEC et al. v. MRI
SEC Exhibit 1013.0042
IPR2023-00221

2.9 SEALED ELECTRONIC BOXES

Electronic systems with high impedance circuits are usually very sensitive to humidity and moisture in the ambient air. Slight amounts of condensation on sensitive components, printed circuits, or electrical connectors can often produce large changes in the operating characteristics of the system. Therefore, to minimize potential problems resulting from humidity, moisture, and condensation, sensitive electronic systems are often packaged in sealed electronic boxes.

Sealed electronic boxes are also used for some air cooled electronic systems, which must be capable of operating in the hard vacuum environment of outer space. The sealed box is used to prevent the loss of air, which is required to cool the electronic components.

Many different types of seals can be used, depending upon the size of the unit, the cost, and the ease of repair. Solder seals are very effective for small covers on boxes, but repairs are inconvenient. O-ring seals are popular and easy to use for large or small boxes. A large, stiff, machined mounting flange with many high strength screws is required for the box and the cover, to provide an effective seal. A typical O-ring seal is shown in Figure 2.5.

Sealed electronic boxes, which are used for space applications, may require the use of a pressure relief valve to reduce the high internal pressure that will develop at high altitudes. If a relief valve is not used, the walls of the box will have to be made stiff enough to resist the high deflections and stresses that may result when the external atmosphere is no longer present.

The seal must do two things. First, it must prevent air (or any other gas, such as dry nitrogen) from leaking out of the box. Second, it must prevent water vapor from entering the box, because water vapor may condense and cause arcing or short circuits. Preventing air from leaking out of the box is

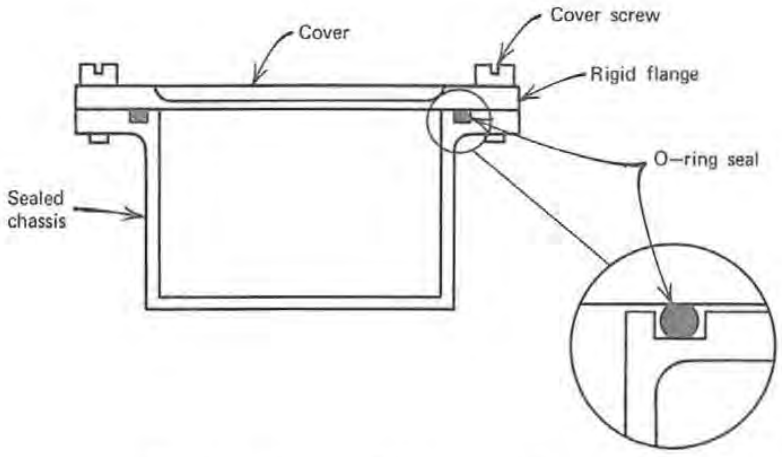


Figure 2.5 O-ring seal for electronic box.

SEC et al. v. MRI
SEC Exhibit 1013.0043
IPR2023-00221

not too difficult to accomplish, since many different types of gaskets will provide an effective seal for an airtight box. A vaportight box may be very difficult to achieve, however, if the wrong type of gasket is used.

Flat rubber gaskets are easy to use and cheap to fabricate. These gaskets are capable of providing an airtight seal for a large electronic box. A typical flat gasket seal is shown in Figure 2.6. A flat rubber gasket should not be used for boxes that require a water vapor seal. The interface pressure between the flat rubber gasket and its mating box and cover flanges is usually very low, because the rubber is free to deform as the cover screws are tightened. Although this gasket will seal in air, it will not seal out water vapor. The water vapor molecule is smaller than the air molecule, so that the water vapor module will pass through restricted openings more readily than the air molecule.

The water vapor molecule is smaller than the air molecule because the water vapor molecule has a smaller molecular weight. The size of the gas molecule is related to the molecular weight of the gas, so that a gas molecule with a high molecular weight will be larger than a gas with a low molecular weight. Air is primarily a mixture of oxygen and nitrogen gases. Oxygen (O_2) has a molecular weight of 32 and nitrogen (N_2) has a molecular weight of 28. Water vapor is still water (H_2O). Two atoms of hydrogen (H_2) have an atomic weight of 2, and one atom of oxygen (O) has an atomic weight of 16, for a total molecular weight of 18 [15].

The smaller size of the water vapor molecule therefore permits it to pass through openings that would normally restrict the flow of the air (oxygen and nitrogen) molecules. The O-ring type of gasket seal has a very high interface pressure, which makes it very effective for boxes that require a water vapor seal.

Water vapor is a gas and therefore obeys the normal gas laws. The force driving the water vapor migration is the difference in the partial pressure of the water vapor between two different areas. Water vapor will tend to flow from the high partial pressure areas to the low partial pressure areas. If an electronic box is assembled in a dry climate, the partial pressure of the

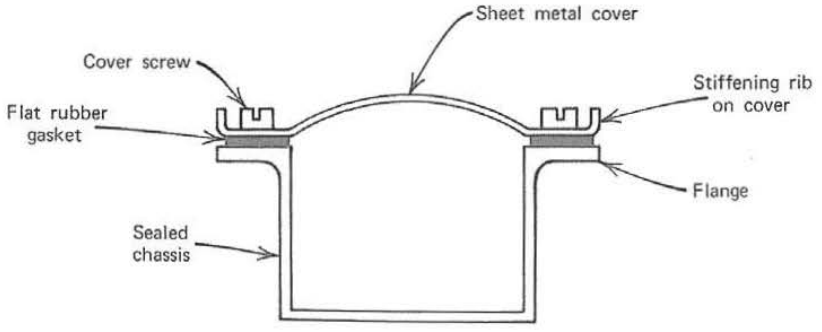


Figure 2.6 Flat rubber gasket for electronic box.

SEC et al. v. MRI
SEC Exhibit 1013.0044
IPR2023-00221

water vapor within the box will be low. If that box is now transported to a humid area, such as Florida, the partial pressure of the water vapor in the atmosphere will be quite high. The water vapor will try to equalize the partial pressure, so that it will try to enter the box. It does not matter if the absolute air pressure within the box is 50 psia and the outside ambient pressure is only 14.7 psia. Since the partial pressure of the water vapor within the box is low, the tendency will be for the water vapor to try to enter the box, to equalize the partial pressures [14].

If the type of gasket used to seal an electronic box is not effective, water vapor will pass through the seal and condense within the electronic box. It does not take very long for half a pint of water to accumulate within a small chassis due to condensation. An electronic box 15 in × 10 in wide × 8 in high, with a flat rubber gasket seal under the top cover can accumulate that amount of condensation in a period of only about 3 months when it is continually exposed to humid air.

Cover gaskets are not the only areas that will permit water vapor to enter an electronic box. Other sources may be the connector interfaces, areas around fasteners, and the basic structure itself.

Connector interfaces on an electronic box must be adequately sealed if the box is to prevent the entry of water vapor. Solder seals are very effective for small connectors, but, again, repairs are inconvenient. O-ring gaskets are convenient, relatively easy to use, and can provide an effective water vapor seal.

A sealed electronic box should not have bolts or other fasteners passing through the outer structural wall into the interior of the chassis. Every hole in the chassis wall is the possible source of a leak. If bolts are required to pass through an external wall, gaskets or some other provision must be made to ensure an effective seal at that interface.

The basic structure of the electronic box itself may not be capable of providing an effective water vapor seal. Sheet metal structures are generally satisfactory. The sheet metal is made from rolled stock, so that it is not porous. However, the box corners may be welded and welds may leak.

Cast structures are often very porous. Some magnesium and aluminum castings 1 in thick will not hold air for more than 5 minutes. Structures of this type can often be impregnated with an epoxy resin. The most effective technique for applying the resin is by means of a vacuum. The porous structure is placed in a vacuum chamber, the air is evacuated, and the resin is applied to the surfaces to be impregnated. The vacuum is then released and the increased air pressure drives the resin into the porous surface, which seals the structure.

Dry nitrogen is very effective for removing water vapor from a sealed box. An inlet and outlet valve are placed at opposite ends of the chassis, which is purged with dry nitrogen to remove all traces of water vapor. The nitrogen valves are then closed so that the box is sealed with dry nitrogen. Since most of our normal atmosphere is nitrogen, the characteristics of the

SEC et al. v. MRI
SEC Exhibit 1013.0045
IPR2023-00221

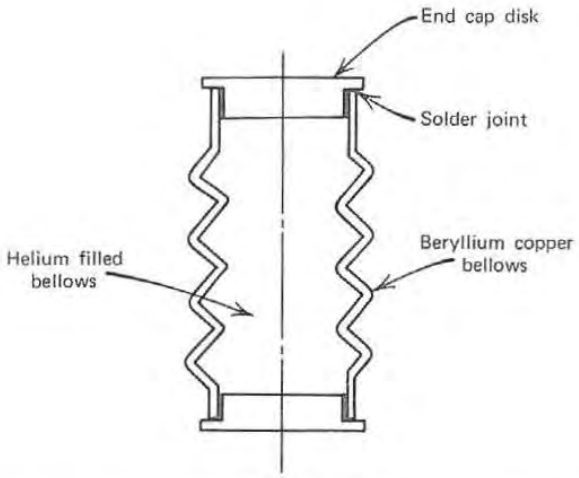


Figure 2.7 Cross section through helium filled sealed bellows with end caps.

gas within the box is very similar to the atmosphere, except that it contains no water vapor.

Desiccators may be placed within the electronic enclosure to absorb any water vapor that may seep into the sealed box over a long period of time. Materials such as silica jel or molecular sieve are quite effective for this purpose.

Care should be exercised in the selection of materials that are to be used for sealed packages. Extrusions should be avoided, if possible, because

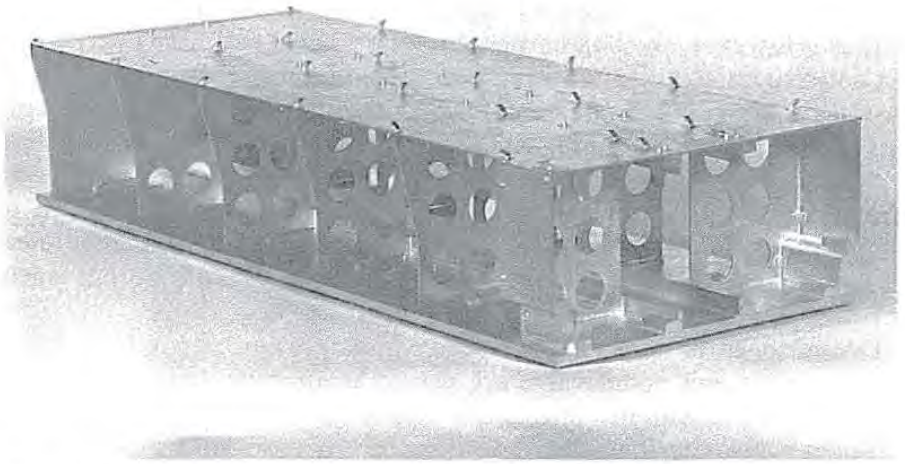


Figure 2.8 Dip brazed electronic box with self jigging tabs. (Courtesy Honeywell Co.)

SEC et al. v. MRI
SEC Exhibit 1013.0046
IPR2023-00221

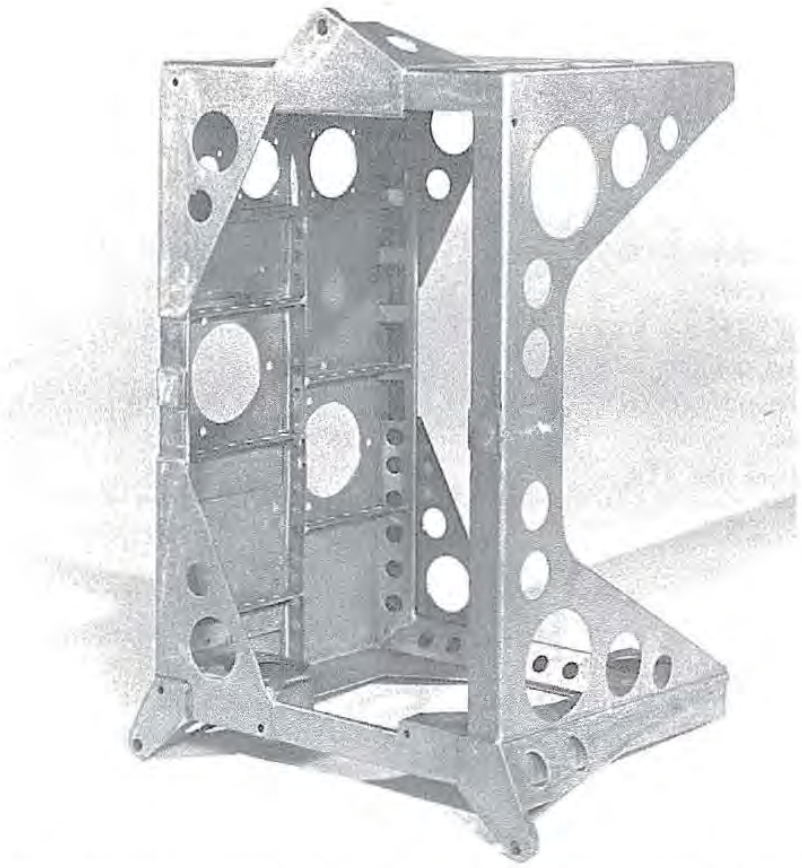


Figure 2.9 Dip brazed electronic chassis with aluminum screws and tack welds for positioning piece parts. (Courtesy Kearfott Division, The Singer Co.)

extruded sections can be very porous. For example, end caps are often used on sealed helium filled bellows, as shown in Figure 2.7. These bellows are used to take up the expansion of the liquid used in gyros, which must operate over a wide temperature range.

If the end caps are made from disk sections that have been sliced from an extruded rod, the end caps can be porous and the helium will leak out. To avoid this difficulty, the end caps should be stamped or machined from flat rolled sheet stock. This would provide end caps that have the grain running perpendicular to the axis of the bellows, so that the axial pores are sharply reduced.

2.10 STANDARD ELECTRONIC BOX SIZES

Electronic boxes can come in a wide variety of sizes and shapes, depending upon the application and environment. One organization, however, has

SEC et al. v. MRI
SEC Exhibit 1013.0047
IPR2023-00221

Table 2.1 ARINC Standard Rectangular Box Sizes

Description	Width (in)	Length (in)	Height (in)
Short one quarter ATR	2.250	12.5625	7.625
Long one quarter ATR	2.250	19.5625	7.625
Short three eights ATR	3.5625	12.5625	7.625
Long three eights ATR	3.5625	19.5625	7.625
Short half ATR	4.875	12.5625	7.625
Long half ATR	4.875	19.5625	7.625
Short three quarters ATR	7.50	12.5625	7.625
Long three quarters ATR	7.50	19.5625	7.625
One ATR	10.125	19.5625	7.625
One and one half ATR	15.375	19.5625	7.625

made an attempt to standardize the size, shape, and mounting for electronic boxes used in air transport equipment. This has become the standard known as ARINC, which is an acronym for Aeronautical Radio Inc. This specification, ARINC 404, Air Transport Equipment Cases and Racking (ATR), December 31, 1970, defines a group of rectangular plug-in types of electronic equipment cases, which have the outer dimensions listed in Table 2.1.

Figure 2.8 shows a dip brazed aluminum chassis with tabs that can be twisted or bent to lock piece parts together before brazing. In Figure 2.9, the piece parts are held by aluminum screws and tack welds prior to brazing.

SEC et al. v. MRI
SEC Exhibit 1013.0048
IPR2023-00221

3

Conduction Cooling for Chassis and Circuit Boards

3.1 CONCENTRATED HEAT SOURCES, STEADY STATE CONDUCTION

Electronic components are not 100% efficient, so that heat is generated in active components whenever the electronic system is in operation. As the heat is generated, the temperature of the component increases and heat attempts to flow through any path it can find. If the heat source is constant, the temperature within the component continues to rise until the rate of the heat being generated is equal to the rate of the heat flowing away from the component. Steady state heat transfer then exists—the heat flowing into the component is equal to the heat flowing away from the component.

Heat always flows from high temperature areas to low temperature areas. When the electronic system is operating, the electronic components are its hottest parts. To control the hot spot component temperatures, the heat flow path must be controlled. If this is not done properly, the component temperatures are forced to rise in an attempt to balance the heat flow. Eventually, the temperatures may become so high that the component is destroyed if a protective circuit or a thermal cutoff switch is not used.

Conduction heat transfer in an electronic system is generally a slow process. Heat flow will not occur until a temperature difference has been established. This means that each member along the heat flow path must experience a temperature rise before the heat will flow to the next point along the path. This process continues until the final heat sink is reached, which is often the outside ambient air. This process may take many hours, perhaps even many days for a very large system, depending upon its size.

The materials along the heat flow path sharply influence the temperature gradients developed along the path. Heat can be conducted through any material: solid, liquid, or gas. The ability of the heat to flow through any material is determined by the physical properties of the material and by the geometry of the structure. The basic heat flow relation for steady state conduction from a single concentrated heat load is shown in Eq. 3.1. This

SEC et al. v. MRI
SEC Exhibit 1013.0049
IPR2023-00221

Table 3.1 Units Used with Thermal Conductivity Equation[a]

Item	Symbol	English Units	Metric Units
Power dissipation	Q	$\frac{\text{Btu}}{\text{hr}}$	$\frac{\text{cal}}{\text{sec}}$
Thermal conductivity	K	$\frac{\text{Btu}}{\text{hr ft °F}}$	$\frac{\text{cal}}{\text{sec cm °C}}$
Cross section area	A	ft^2	cm^2
Temperature difference	Δt	°F	°C
Length	L	ft	cm

[a] *Power in watts must always be converted into Btu/hr or cal/sec before it can be used in the equation.*

relation can be used for both English and metric units (Table 3.1). Consistent sets of units must always be used, however, as defined below [15–17].

$$Q = KA\frac{\Delta t}{L} \tag{3.1}$$

3.2 MOUNTING ELECTRONIC COMPONENTS ON BRACKETS

Electronic components are often mounted on brackets, because wall space on a typical chassis is very limited. The bracket must provide a good heat flow path from the component to the ultimate heat sink, which may be a cold plate on the chassis wall. The temperature rise that develops from the component to the chassis wall is determined by the size of the bracket, how the bracket is fastened to the wall, and how the component is fastened to the bracket. The actual temperature that is developed at the mounting surface of the component is simply the temperature of the wall plus the temperature rise from the wall to the surface of the component.

Structural epoxies are convenient for mounting brackets to chassis walls. Epoxies are strong, easy to apply, and cure rapidly. One note of caution: Avoid silicone greases in areas that will be cemented. Silicones tend to migrate. Even if the grease is not in contact with the cemented joint, the migration will carry it to the joint before the epoxy is applied. This prevents the epoxy from bonding properly, resulting in a weakened joint.

If the chassis will experience any vibration or shock, it is a good idea to use mechanical fasteners such as rivets or screws with the epoxy to improve the bond.

Cantilevered brackets should be avoided in shock and vibration environments because they have low resonant frequencies and high transmissibilities. This results in high dynamic stresses and a low fatigue life [1].

SEC et al. v. MRI
SEC Exhibit 1013.0050
IPR2023-00221

Table 3.2 Thermal Conductivity of Various Materials [18–23]

Material (room temperature)	English Units $\left(\frac{\text{Btu}}{\text{hr ft °F}}\right)$	Metric Units $\left(\frac{\text{cal}}{\text{sec cm °C}}\right)$
	Metal	
Aluminum		
Pure	125	0.52
5052	83	0.34
6061 T6	90	0.37
2024 T4	70	0.29
7075 T6	70	0.29
356 T6	87	0.36
Beryllium	95	0.39
Beryllium copper	50	0.21
Copper		
Pure	230	0.95
Drawn wire	166	0.68
Bronze	130	0.54
Red brass	64	0.26
Yellow brass	54	0.22
5% phosphor bronze	30	0.12
Gold	170	0.70
Iron		
Wrought	34	0.14
Cast	32	0.13
Kovar	9	0.037
Lead	19	0.078
Magnesium		
Pure	92	0.38
Cast	41	0.17
Molybdenum	75	0.31
Nickel		
Pure	46	0.19
Inconel	10	0.041
Monel	15	0.062
Silver	242	1.00
Steel		
SAE 1010	34	0.14
1020	32	0.13
1045	26	0.11
4130	24	0.10
Tin	36	0.15
Titanium	9	0.037
Zinc	59	0.24

SEC et al. v. MRI
SEC Exhibit 1013.0051
IPR2023-00221

Table 3.2 (Continued)

Material (room temperature)	English Units $\left(\frac{\text{Btu}}{\text{hr ft °F}}\right)$	Metric Units $\left(\frac{\text{cal}}{\text{sec cm °C}}\right)$
	Nonmetal	
Air	0.0153	0.000063
Alumina	17.0	0.070
Bakelite	0.11	0.00045
Beryllia (99.5%)	114.	0.47
Carbon	4.0	0.016
Epoxy		
No fill	0.12	0.00049
High fill	1.25	0.0051
Epoxy fiberglass	0.15	0.00062
Multilayer 0.005 in epoxy lamina with 0.0028 copper runs parallel to plane	20.	0.083
Perpendicular to plane (with plated throughholes)	2.	0.0082
Glass	0.50	0.0021
Glass wool	0.023	0.00009
Ice	1.23	0.0051
Mica	0.41	0.0017
Mylar	0.11	0.00045
Nylon	0.14	0.00058
Phenolic		
Plain	0.30	0.0012
Paper base	0.16	0.00066
Plexiglass	0.11	0.00045
Polystyrene	0.061	0.00025
Polyvinyl chloride	0.09	0.00037
Pyrex	0.73	0.0030
Rubber		
Butyl	0.15	0.00062
Hard	0.11	0.00045
Soft	0.08	0.00033
Silicone grease	0.12	0.00049
Silicone rubber	0.11	0.00045
Styrofoam	0.02	0.00008
Teflon	0.11	0.00045
Water	0.38	0.0016
Wood		
Maple	0.096	0.00039
Pine	0.067	0.00027

SEC et al. v. MRI
SEC Exhibit 1013.0052
IPR2023-00221

3.3 SAMPLE PROBLEM—TRANSISTOR MOUNTED ON A BRACKET

Determine the mounting surface temperature on the case of a transistor that is bolted to an aluminum bracket (5052 aluminum) as shown in Figure 3.1. The bracket is cemented to a heat sink wall, which is maintained at a temperature of 131°F (55°C). The transistor has a mica insulator at the mounting interface and the power dissipation is 7.5 watts. Radiation and convection heat transfer from the assembly are small and can be ignored.

SOLUTION

The heat transfer path from the transistor to the heat sink wall is broken into three parts, and each part is examined separately, using Eq. 3.1 written in a slightly different form, as shown in Eq. 3.2. The solution is shown in English units and metric units.

$$\Delta t = \frac{QL}{KA} \tag{3.2}$$

Part 1. Temperature Rise across Cemented Interface

Given Q = concentrated heat load of 7.5 watts
Q = 25.6 Btu/hr = 1.79 cal/sec (ref. Table 1.1)
L = thickness of cemented interface
L = 0.010 in = 0.000833 ft = 0.0254 cm
K = thermal conductivity epoxy cement (ref. Table 3.2)
K = 0.167 Btu/hr ft °F = 0.000689 cal/sec cm °C (ref. Table 1.2)
A = cross section area along heat flow path
A = (1.3)(0.75) = 0.975 in² = 0.00677 ft² = 6.29 cm²

Substitute into Eq. 3.2 using English units.

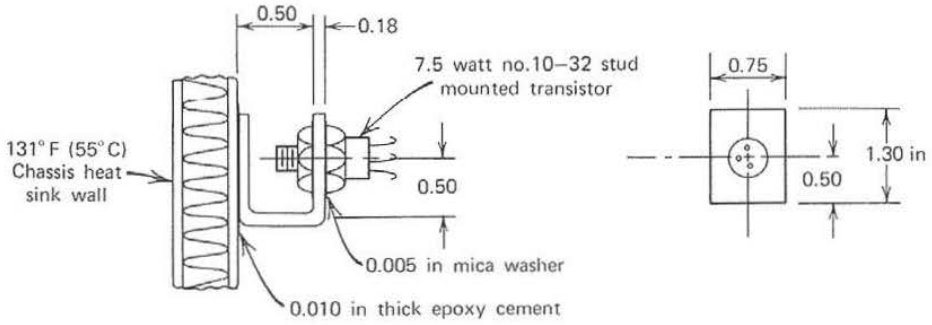


Figure 3.1 Transistor mounted on a bracket.

SEC et al. v. MRI
SEC Exhibit 1013.0053
IPR2023-00221

$$\Delta t_1 = \frac{(25.6 \text{ Btu/hr})(0.000833 \text{ ft})}{(0.167 \text{ Btu/hr ft °F})(0.00677 \text{ ft}^2)} = 18.9°\text{F} \tag{3.3}$$

Substitute into Eq. 3.2 using metric units.

$$\Delta t_1 = \frac{(1.79 \text{ cal/sec})(0.0254 \text{ cm})}{(0.000689 \text{ cal/sec cm °C})(6.29 \text{ cm}^2)} = 10.5°\text{C} \tag{3.3a}$$

Part 2. Temperature Rise along the Aluminum Bracket

Given Q = 7.5 watts = 25.6 Btu/hr = 1.79 cal/sec
L = length of bracket from center of cemented face to center of transistor mount
L = 0.50 + 0.50 + 0.50 = 1.5 in = 0.125 ft = 3.81 cm
K = thermal conductivity 5052 aluminum bracket (ref. Table 3.2)
K = 83 Btu/hr ft °F = 0.343 cal/sec cm °C
A = cross section area of bracket
A = (0.75)(0.18) = 0.135 in² = 0.000937 ft² = 0.871 cm²

Substitute into Eq. 3.2 using English units.

$$\Delta t_2 = \frac{(25.6)(0.125)}{(83)(0.000937)} = 41.1°\text{F} \tag{3.4}$$

Substitute into Eq. 3.2 using metric units.

$$\Delta t_2 = \frac{(1.79)(3.81)}{(0.343)(0.871)} = 22.8°\text{C} \tag{3.4a}$$

Part 3. Temperature Rise across Mica Insulator (Ref. Table 3.3)

Given Q = 7.5 watts (3.5)
R = 2.4 °C/watt (10–32 stud; ref. Table 3.3)
Δt_3 = (7.5 watts) (2.4 °C/watt)
Δt_3 = 18°C = 32.4°F

The mounting surface temperature of the transistor is determined from the chassis wall temperature plus the temperature rises along the heat flow path.

$$t_{\text{trans}} = t_{\text{surf}} + \Delta t_1 + \Delta t_2 + \Delta t_3$$

In English units, the transistor temperature is

$$t_{\text{trans}} = 131 + 18.9 + 41.1 + 32.4 = 223.4°\text{F} \tag{3.6}$$

SEC et al. v. MRI
SEC Exhibit 1013.0054
IPR2023-00221

Table 3.3 Thermal Resistance R across Interface from Case to Sink for Stud Mounted Transistors and Diodes

Stud Size	Hex Size across Flats	Dry	With Thermal Grease	Across Mica Washer 0.005 in (0.0127 cm)
10–32	$\frac{7}{16}$ in	$0.60\ \frac{°C}{watt}$	$0.40\ \frac{°C}{watt}$	$2.4\ \frac{°C}{watt}$
$\frac{1}{4}$–28	$\frac{9}{16}$ in	$0.45\ \frac{°C}{watt}$	$0.30\ \frac{°C}{watt}$	$2.0\ \frac{°C}{watt}$

In metric units, the transistor temperature is

$$t_{trans} = 55 + 10.5 + 22.8 + 18.0 = 106.3°C \tag{3.6a}$$

3.4 UNIFORMLY DISTRIBUTED HEAT SOURCES, STEADY STATE CONDUCTION

Identical electronic components are often placed next to one another, on mounting brackets or on circuit boards, as shown in Figure 3.2. When each component dissipates approximately the same amount of power, the result will be a uniformly distributed heat load.

When any one strip of electronic components is considered, as shown in

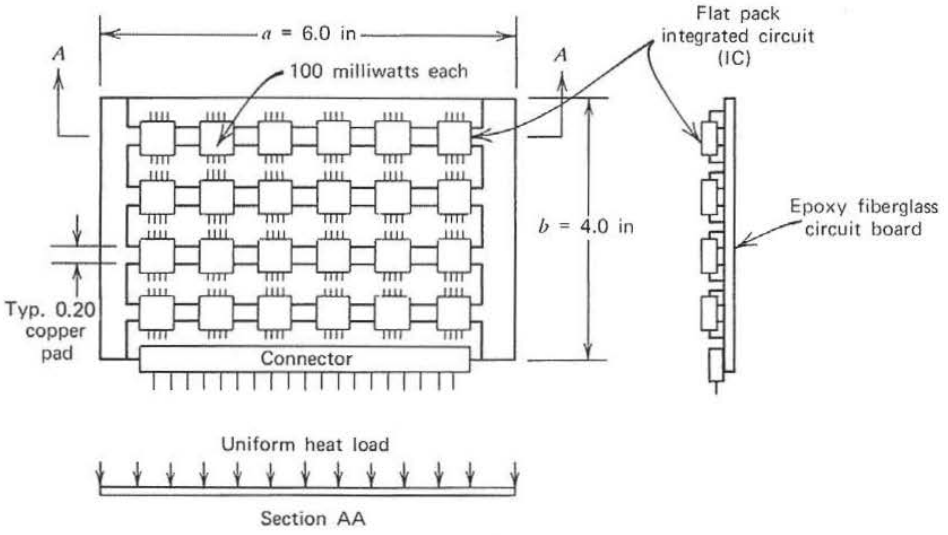


Figure 3.2 Printed circuit board with flat pack integrated circuits.

SEC et al. v. MRI
SEC Exhibit 1013.0055
IPR2023-00221

section AA, the heat input can be evaluated as a uniformly distributed heat load. On a typical printed circuit board (PCB), the heat will flow from the component to the heat sink strip under the component, and then to the outer edges of the PCB, where it is removed. The heat sink strip is usually made of copper or aluminum, which have high thermal conductivity. The maximum temperature will occur at the center of the PCB and the minimum temperature will occur at the edges. This produces a parabolic temperature distribution, as shown in Figure 3.3.

When only one side of one strip is considered, a heat balance equation can be obtained by considering a small element dx of the strip, along the span with a length of L. Then

$$dQ_1 + dQ_2 = dQ_3$$

where $dQ_1 = q\,dx$ = heat input

$$dQ_2 = -KA\frac{dt}{dx} = \text{heat flow}$$

$$dQ_3 = KA\frac{d}{dx}(t + dt) = \text{total heat}$$

Then

$$q\,dx - KA\frac{dt}{dx} = -KA\frac{dt}{dx} - KA\frac{d}{dx}(dt)$$

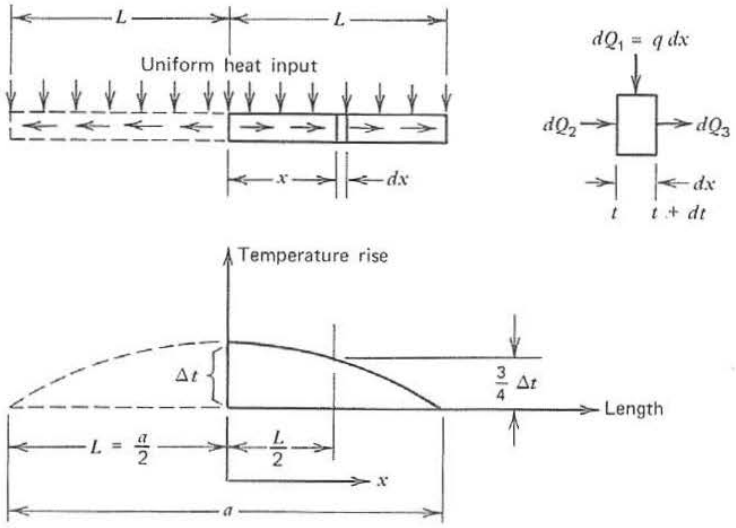


Figure 3.3 Parabolic temperature distribution for uniform heat load on a circuit board.

SEC et al. v. MRI
SEC Exhibit 1013.0056
IPR2023-00221

or

$$\frac{d^2t}{dx^2} = -\frac{q}{KA}$$

This is a second order differential equation, which can be solved by integrating twice. Integrating once yields

$$\frac{dt}{dx} = -\frac{qx}{KA} + C_1$$

Integrating twice yields

$$t = -\frac{qx^2}{2KA} + C_1x + C_2$$

The constant C_1 is zero because at $x = 0$, the plate is adiabatic and no heat is lost. The slope of the temperature gradient is therefore zero.

The constant C_2 is determined by letting the temperature at the end of the plate be t_e; then

$$C_2 = t_e + \frac{qL^2}{2KA}$$

The temperature at any point along the plate (or strip) is

$$t = -\frac{qx^2}{2KA} + t_e + \frac{qL^2}{2KA}$$

Let the temperature at the center of the plate be t_0; then

$$t_0 - t_e = \frac{q}{2KA}(L^2 - x^2)$$

When $t_0 - t_e$ is represented as Δt,

$$\Delta t = \frac{q}{2KA}(L^2 - x^2) \tag{3.7}$$

This equation produces the parabolic shape for the temperature distribution, which is shown in Figure 3.3

When the full length of the strip is considered, the value of x in Eq. 3.7 is zero. The total heat flow along the length L then becomes

$$Q = qL \tag{3.8}$$

SEC et al. v. MRI
SEC Exhibit 1013.0057
IPR2023-00221

Substitute Eq. 3.8 into Eq. 3.7 when $x = 0$. This results in the equation for the maximum temperature rise in a strip with a uniformly distributed heat load.

$$\Delta t = \frac{QL}{2KA} \tag{3.9}$$

When the temperature rise at the midpoint along the heat sink strip of length L is desired, then $x = L/2$. Substituting this value into Eq. 3.7 will result in the temperature rise at the midpoint of the strip:

$$\text{midpoint } \Delta t = \frac{3QL}{8KA} \tag{3.10}$$

Considering only the strip with a length of L, the ratio of the strip midpoint temperature rise to the maximum temperature rise is shown in the following relation:

$$\frac{\text{midpoint } \Delta t}{\text{maximum } \Delta t} = \frac{3QL/8KA}{QL/2KA} = \frac{3}{4} \tag{3.11}$$

3.5 SAMPLE PROBLEM—COOLING INTEGRATED CIRCUITS ON A PCB

A series of flat pack integrated circuits are to be mounted on a multilayer printed circuit board (PCB) as shown in Figure 3.2. Each flat pack dissipates 100 milliwatts of power. Heat from the components is to be removed by conduction through the printed circuit copper pads, which have 2 ounces of copper [thickness is 0.0028 in (0.0071 cm)]. The heat must be conducted to the edges of the PCB, where it flows into a heat sink. Determine the temperature rise from the center of the PCB to the edge to see if the design will be satisfactory.

SOLUTION

The flat packs generate a uniformly distributed heat load, which results in the parabolic temperature distribution shown in Figure 3.4. Because of symmetry, only one half of the system is evaluated. Equation 3.9 is used to determine the temperature rise from the center of the PCB to the edge for one strip of components, using English units and metric units.

SEC et al. v. MRI
SEC Exhibit 1013.0058
IPR2023-00221

Given $Q = 3(0.10) = 0.30$ watt heat input, one half strip
$Q = 1.02$ Btu/hr $= 0.0717$ cal/sec (ref. Table 1.1)
$L = 3.0$ in $= 0.25$ ft $= 7.62$ cm (length)
$K = 166$ Btu/hr ft °F $= 0.685$ cal/sec cm °C (Thermal conductivity of copper)
$A = (0.20)(0.0028) = 0.00056$ in² (cross section area)
$A = 3.89 \times 10^{-6}$ ft² $= 0.00361$ cm²

Substitute into Eq. 3.9 to obtain the temperature rise along the strip using English units.

$$\Delta t = \frac{(1.02)(0.25\,\text{ft})}{(2)(166)(3.89 \times 10^{-6}\,\text{ft}^2)} = 197°\text{F} \tag{3.12}$$

Substitute into Eq. 3.9 using metric units.

$$\Delta t = \frac{(0.0717)(7.62\,\text{cm})}{(2)(0.685)(0.00361\,\text{cm}^2)} = 110°\text{C} \tag{3.12a}$$

The amount of heat that can be removed by radiation or convection for this type of system is very small. The temperature rise is therefore too high. By the time the sink temperature is added, assuming that it is 80°F, the case temperature on the component will be 277°F. Since the typical maximum allowable case temperature is about 212°F, the design is not acceptable.

If the copper thickness is doubled to 4 ounces, which has a thickness of 0.0056 in (0.014 cm), the temperature rise will be 98.5°F (55°C), which is still too high for any sink greater than about 115°F (46°C). For high temperature applications, the copper thickness will have to be increased to about 0.0112 in (0.0284 cm) for a good design.

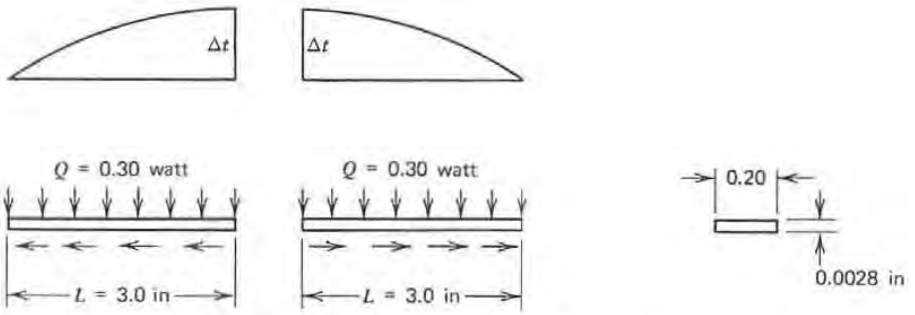


Figure 3.4 Uniformly distributed heat load on one copper strip.

SEC et al. v. MRI
SEC Exhibit 1013.0059
IPR2023-00221

3.6 CIRCUIT BOARD WITH AN ALUMINUM HEAT SINK CORE

Another method is often used for fabricating PCBs that will have a good conduction heat flow path. A thin circuit lamina is cemented to an aluminum plate, and flat pack integrated components are lap soldered to the lamina. Laminas can be cemented to both sides of the aluminum plate, so that both sides of the plate can be populated with integrated circuit components, as shown in Figure 3.5.

Twice as many integrated circuits can be mounted on this type of PCB. This means the total power dissipation will be twice that of a PCB which has components on one side only. If all the components have about the same power dissipation, a uniform heat load will be produced, so that the temperature rise distribution will still be parabolic, as shown in Figure 3.3.

3.7 SAMPLE PROBLEM—TEMPERATURE RISE ALONG A PCB HEAT SINK PLATE

Determine the temperature rise from the center of the aluminum heat sink plate to the side edges for the conductively cooled PCB shown in Figure 3.5. The heat flow path from the center of the heat sink to the side edges is shown in Figure 3.6. Components are mounted on both sides of the board, which results in a power dissipation of 2.4 watts per side, or a total power dissipation of 4.8 watts for the complete PCB.

SOLUTION

The temperature rise along the heat sink plate is determined from Eq. 3.9 considering English units and metric units.

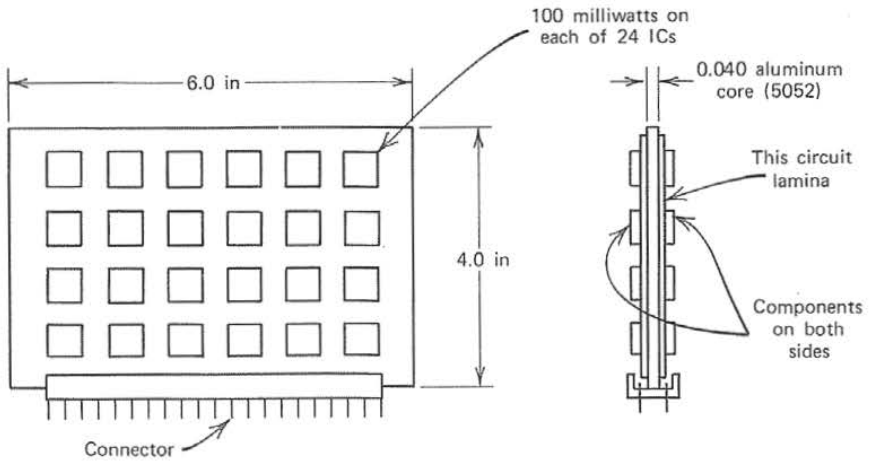


Figure 3.5 Circuit board with aluminum core for good heat conduction.

SEC et al. v. MRI
SEC Exhibit 1013.0060
IPR2023-00221

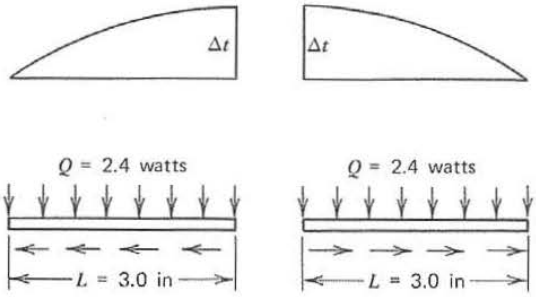


Figure 3.6 Uniformly distributed heat load on an aluminum board core.

Given $Q = 2.4$ watts $= 8.19$ Btu/hr $= 0.574$ cal/sec (ref. Table 1.1)
$L = 3.0$ in $= 0.25$ ft $= 7.62$ cm (heat sink length)
$K = 83$ Btu/hr ft °F $= 0.343$ cal/sec cm °C [aluminum (5052) conductivity]
$A = (0.04)(4.0) = 0.16$ in²
$A = 0.00111$ ft² $= 1.032$ cm² (cross section area)

Substitute the English units into Eq. 3.9.

$$\Delta t = \frac{(8.19)(0.25)}{(2)(83)(0.00111)} = 11.1°\text{F} \tag{3.13}$$

Substitute the metric units into Eq. 3.9.

$$\Delta t = \frac{(0.574)(7.62)}{(2)(0.343)(1.032)} = 6.2°\text{C} \tag{3.13a}$$

Note that these results do *not* include the temperature rise across the thin circuit lamina. The results are only for the aluminum heat sink plate.

With such a small temperature rise, this type of design is well suited for high temperature applications.

3.8 HOW TO AVOID WARPING ON PCBS WITH METAL HEAT SINKS

Aluminum and copper heat sink cores on circuit boards are capable of conducting away large amounts of heat if the proper construction techniques are used. Extreme care must be used if only one thin circuit lamina is to be bonded to a metal core. Improper bonding can result in warped boards, because of the big difference in the coefficient of thermal expansion between the metal and the epoxy fiberglass lamina.

SEC et al. v. MRI
SEC Exhibit 1013.0061
IPR2023-00221

Thin sheets of prepreg are often used to bond the epoxy fiberglass printed circuit laminas to the metal heat sink plate. This requires a high temperature and a high pressure to cure the prepreg adhesive. Since aluminum and copper both have a coefficient of thermal expansion that is about double that of the epoxy fiberglass, in the plane of the PCB the metal will expand more than the epoxy at the high curing temperature. When the cured assembly is brought down to room temperature, the metal will shrink more than the epoxy, causing the bonded assembly to warp.

The warping caused by nonuniform shrinking between the metal core and the epoxy fiberglass lamina can usually be reduced or eliminated by bonding a similar part on the opposite side of the plate.

When only one side of a metal plate is to be laminated with a printed circuit, a room temperature (or slightly higher) cementing process should be used instead of the high temperature bonding process. There are many epoxy cements that have a relatively low temperature curing cycle [22, 23].

3.9 CHASSIS WITH NONUNIFORM WALL SECTIONS

Electronic chassis always seem to require cutouts, notches, and clearance holes for assembly access, wire harnesses, or maintenance. These openings will generally cut through a bulkhead or other structural member, which is required to carry heat away from some critical high power electronic component. The cutouts result in nonuniform wall sections, which must be analyzed to determine their heat flow capability.

One convenient method for analyzing nonuniform wall sections is to subdivide them into smaller units that have relatively uniform sections. The heat flow path through each of the smaller, relatively uniform sections can then be defined in terms of a thermal resistance. A specific resistance value for each section can then be determined by its geometry and physical properties. This will result in a thermal analog resistor network, or mathematical model, which describes the thermal characteristics of that structural section of the electronic system.

The basic conduction heat flow relation shown by Eq. 3.1 can be modified slightly to utilize the thermal resistance concept. The thermal resistance for conduction heat flow is then defined by Eq. 3.14 [24].

$$R = \frac{L}{KA} \tag{3.14}$$

Substituting this value into Eq. 3.1 results in the temperature rise relation when the thermal resistance concept is used, as shown in Eq. 3.15.

$$\Delta t = QR \tag{3.15}$$

SEC et al. v. MRI
SEC Exhibit 1013.0062
IPR2023-00221



Figure 3.7 Series flow resistor network.

The thermal resistance concept is very convenient for developing mathematical models of simple or complex electronic boxes. Analog resistor networks can be established for heat flow in one, two, and three dimensions with little effort. Complex shapes can often be modeled with many simple thermal resistors to provide an effective means for determining the temperature profile for almost any type of system.

Two basic resistance patterns, series and parallel, are used to generate analog resistor networks. A simple series pattern network is shown in Figure 3.7.

The total effective resistance (R_t) for the series flow network is determined by adding all the individual resistances, as shown in Eq. 3.16 [15, 17, 24].

$$R_t = R_1 + R_2 + R_3 + \cdots \tag{3.16}$$

A simple parallel flow resistor network is shown in Figure 3.8.

The total effective resistance for the parallel flow network is determined by combining the individual resistances as shown in Eq. 3.17 [15, 17, 24].

$$\frac{1}{R_t} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \cdots \tag{3.17}$$

A typical electronic system will normally consist of many different combinations of series and parallel flow resistor networks.

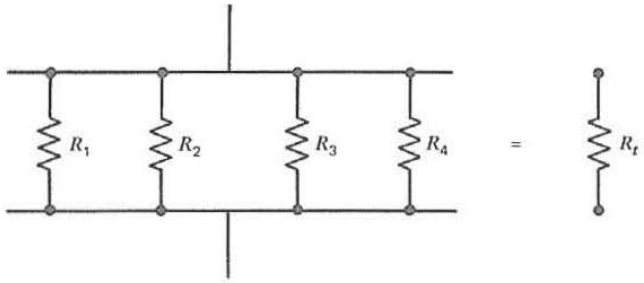


Figure 3.8 Parallel flow resistor network.

SEC et al. v. MRI
SEC Exhibit 1013.0063
IPR2023-00221

3.10 SAMPLE PROBLEM—HEAT FLOW ALONG NONUNIFORM BULKHEAD

An aluminum bulkhead is used to support a row of six power resistors. Each resistor dissipates 1.5 watts, for a total power dissipation of 9 watts. The bulkhead conducts the heat to the opposite wall of the chassis, which is cooled by a multiple fin heat exchanger. The bulkhead has two cutouts for connectors to pass through, as shown in Figure 3.9. Determine the temperature rise across the length of the bulkhead.

SOLUTION

A mathematical model with series and parallel thermal resistor networks can be established to represent the heat flow path, as shown in Figure 3.10*a*.

The thermal analog resistor network is simplified by first finding the equivalent effective single resistance for the parallel section represented by resistors R_2, R_3, and R_4 with the use of Eq. 3.17 and the geometry shown in Figure 3.9.

Determine resistor R_2:

Given $L_2 = 1.5$ in $= 0.125$ ft $= 3.81$ cm (length)
$K_2 = 83$ Btu/hr ft °F $= 0.343$ cal/sec cm °C [aluminum (5052) conductivity]
$A_2 = (0.50)(0.060) = 0.030$ in²
$A_2 = 0.000208$ ft² $= 0.194$ cm² (cross section area)

$$R_2 = \frac{0.125}{(83)(0.000208)} = 7.24 \frac{\text{hr °F}}{\text{Btu}} \text{ (English units)} \tag{3.18}$$

$$R_2 = \frac{3.81}{(0.343)(0.194)} = 57.3 \frac{\text{sec °C}}{\text{cal}} \text{ (metric units)} \tag{3.18a}$$

Determine resistor R_3:

Given $L_3 = 1.5$ in $= 0.125$ ft $= 3.81$ cm (length)
$K_3 = 83$ Btu/hr ft °F $= 0.343$ cal/sec cm °C (aluminum)
$A_3 = (1.0)(0.060) = 0.060$ in² (area)
$A_3 = 0.000417$ ft² $= 0.387$ cm²

$$R_3 = \frac{0.125}{(83(0.000417)} = 3.61 \frac{\text{hr °F}}{\text{Btu}} \text{ (English units)} \tag{3.19}$$

$$R_3 = \frac{3.81}{(0.343)(0.387)} = 28.7 \frac{\text{sec °C}}{\text{cal}} \text{ (metric units)} \tag{3.19a}$$

SEC et al. v. MRI
SEC Exhibit 1013.0064
IPR2023-00221

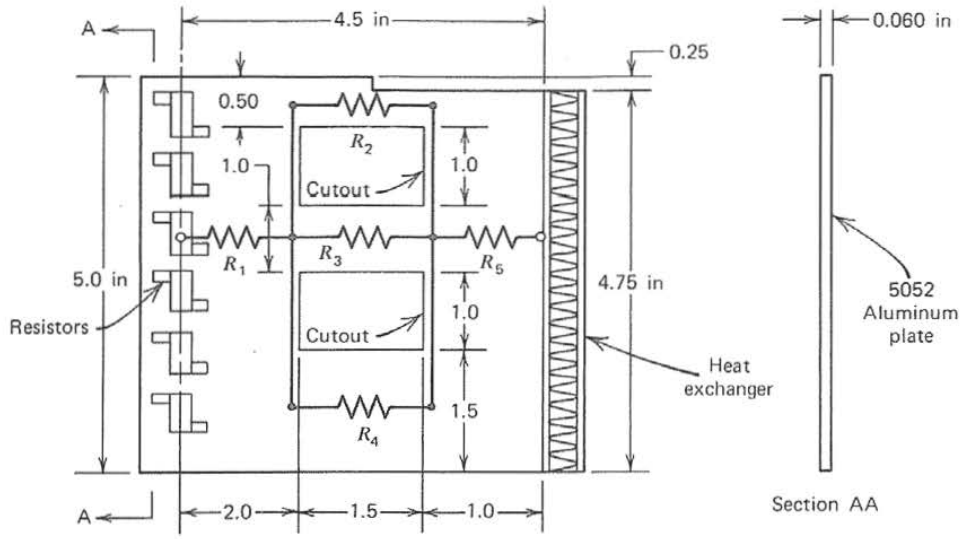


Figure 3.9 Bulkhead with two cutouts for connectors.

Determine resistor R_4:

Given $L_4 = 1.5$ in $= 0.125$ ft $= 3.81$ cm (length)

$K_4 = 83$ Btu/hr ft °F $= 0.343$ cal/sec cm °C (aluminum)

$A_4 = (1.5)(0.060) = 0.090$ in^2

$A_4 = 0.000625$ ft$^2 = 0.581$ cm^2 (cross section area)

$$R_4 = \frac{0.125}{(83)(0.000625)} = 2.41 \frac{\text{hr °F}}{\text{Btu}} \text{(English units)} \qquad (3.20)$$

$$R_4 = \frac{3.81}{(0.341)(0.581)} = 19.1 \frac{\text{sec °C}}{\text{cal}} \text{(metric units)} \qquad (3.20a)$$

Resistors R_2, R_3, and R_4 are combined in parallel using Eq. 3.17, which results in resistor R_6.

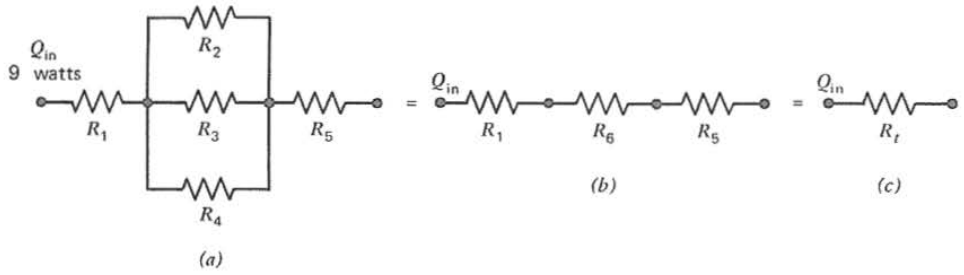


Figure 3.10 Bulkhead thermal model using a series and a parallel resistor network.

SEC et al. v. MRI
SEC Exhibit 1013.0065
IPR2023-00221

$$\frac{1}{R_6} = \frac{1}{7.24} + \frac{1}{3.61} + \frac{1}{2.41}$$

$$R_6 = 1.20 \frac{\text{hr °F}}{\text{Btu}} \text{(English units)} \tag{3.21}$$

$$\frac{1}{R_6} = \frac{1}{57.3} + \frac{1}{28.7} + \frac{1}{19.1}$$

$$R_6 = 9.56 \frac{\text{sec °C}}{\text{cal}} \text{(metric units)} \tag{3.21a}$$

The mathematical model now appears as shown in Figure 3.10*b*. This can be further reduced by considering thermal resistance R_6 to be in series with resistors R_1 and R_5. Each resistance value is determined from Eq. 3.14. The geometry for these resistors is obtained from Figure 3.9.

Determine resistor R_1:

Given $L_1 = 2.0$ in $= 0.167$ ft $= 5.08$ cm (length)
$K_1 = 83$ Btu/hr ft °F $= 0.343$ cal/sec cm °C [aluminum (5052) conductivity]
$A = (5.0)(0.060) = 0.30$ in²
$A = 0.00208$ ft² $= 1.935$ cm² (cross section area)

$$R_1 = \frac{0.167}{(83)(0.00208)} = 0.97 \frac{\text{hr °F}}{\text{Btu}} \text{(English units)} \tag{3.22}$$

$$R_1 = \frac{5.08}{(0.343)(1.935)} = 7.65 \frac{\text{sec °C}}{\text{cal}} \text{(metric units)} \tag{3.22a}$$

Determine resistor R_5:

Given $L_5 = 1.0$ in $= 0.0833$ ft $= 2.54$ cm (length)
$K_5 = 83$ Btu/hr ft °F $= 0.343$ cal/sec cm °C (aluminum)
$A_5 = (4.75)(0.060) = 0.285$ in²
$A_5 = 0.00198$ ft² $= 1.839$ cm² (cross section area)

$$R_5 = \frac{0.0833}{(83)(0.00198)} = 0.51 \frac{\text{hr °F}}{\text{Btu}} \text{(English units)} \tag{3.23}$$

$$R_5 = \frac{2.54}{(0.343)(1.839)} = 4.03 \frac{\text{sec °C}}{\text{cal}} \text{(metric units)} \tag{3.23a}$$

The values of the three resistors in Figure 3.10*b* are now known, so that Eq. 3.16 can be used to determine the total resistance of the three resistors in series.

SEC et al. v. MRI
SEC Exhibit 1013.0066
IPR2023-00221

$$R_t = 0.97 + 1.20 + 0.51 = 2.68 \frac{\text{hr °F}}{\text{Btu}} \text{ (English units)} \quad (3.24)$$

$$R_t = 7.65 + 9.56 + 4.03 = 21.24 \frac{\text{sec °C}}{\text{cal}} \text{ (metric units)} \quad (3.24a)$$

The temperature rise across the length of the bulkhead is then determined from Eq. 3.15, in English units and metric units.

Given $Q = 9$ watts $= 30.72$ Btu/hr $= 2.15$ cal/sec (heat dissipated)
$R_t = 2.68$ hr °F/Btu $= 21.24$ sec °C/cal (thermal resistance)

$$\Delta t = (30.72)(2.68) = 82.3°\text{F (English units)} \quad (3.25)$$

$$\Delta t = (2.15)(21.24) = 45.7°\text{C (metric units)} \quad (3.25a)$$

3.11 TWO DIMENSIONAL ANALOG RESISTOR NETWORKS

The conduction heat flow path through a typical electronic chassis usually occurs in two or more directions at the same time. If there are several concentrated heat input sources along the path, the temperature distribution along the heat flow path may be difficult to evaluate. The problem may be simplified by breaking up the structure into a group of smaller elements, which are interconnected by thermal resistors. The mass of each small element would be concentrated at the geometric center of the element. This point is called the node point of the element and is shown in Figure 3.11. The network of resistors and nodes is called an analog resistor network, and it is based upon the electrical networks that are used so extensively by electrical and electronic engineers [15, 17, 24].

Very large two and three dimensional analog resistor networks are often used with high speed digital computers to determine the thermal profile of

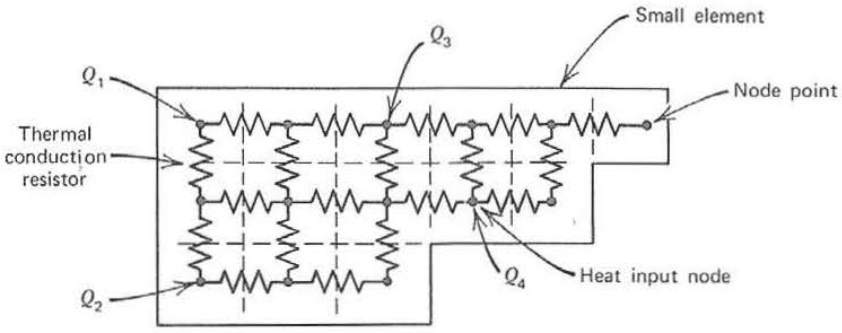


Figure 3.11 Analog resistor network for two dimensional heat flow in a plate.

SEC et al. v. MRI
SEC Exhibit 1013.0067
IPR2023-00221

complex structures. Temperatures in smaller models can often be determined with the use of the basic conduction heat flow relation shown in Eq. 3.1. A heat balance equation is written for each node point in the mathematical model. The simultaneous solution of these equations then results in the temperatures at each node point.

The heat balance equations are a little easier to use when the concept of thermal conductance is used instead of thermal resistance. Thermal conductance (k') is defined as the inverse of the thermal resistance (R), as shown in Eq. 3.26.

$$k' = \frac{1}{R} = \frac{KA}{L} \tag{3.26}$$

Substitute Eq. 3.26 into Eq. 3.1. This results in the steady state heat flow relation using the concept of conductance, which is shown in Eq. 3.27.

$$Q = k' \, \Delta t = k'(t_2 - t_1) \tag{3.27}$$

3.12 SAMPLE PROBLEM—TWO DIMENSIONAL CONDUCTION ON A POWER SUPPLY HEAT SINK

A cantilevered copper bracket is used to support part of a power supply, which consists of two power transistors and two wire wound resistors, with power dissipations as shown in Figure 3.12. The power supply heat sink is bolted to a cold plate, which is maintained at a temperature of 160°F (71°C). Determine the temperatures at the four component mounting points on the power supply heat sink.

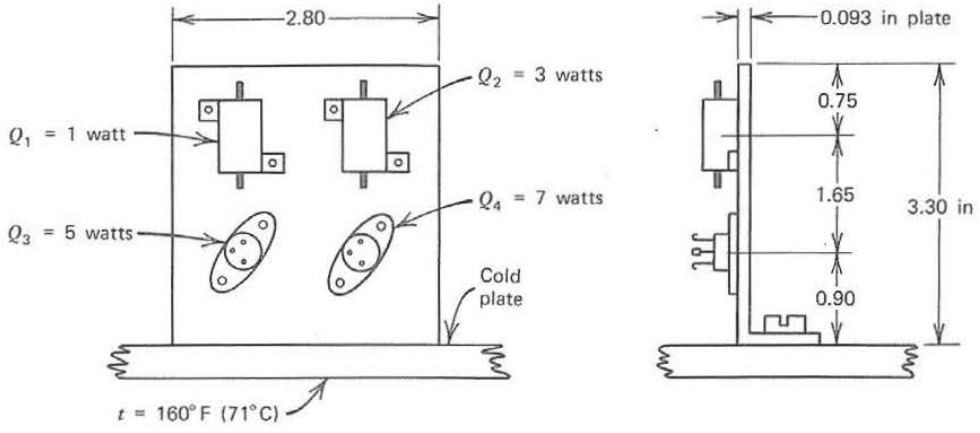


Figure 3.12 Components mounted on a heat sink bracket.

SEC et al. v. MRI
SEC Exhibit 1013.0068
IPR2023-00221

SOLUTION

The bracket is divided into small rectangular elements, with a node point at the geometric center of each element. The node points in each element are interconnected to form a rectangular grid pattern. This results in a mathematical model of the heat sink bracket, which is in the form of an analog conductance network representing a system with heat flow in two directions, as shown in Figure 3.13.

A heat balance equation must be written for each of the node points in the mathematical model to determine the temperature at each node point. To avoid the possibility of making an error in establishing a positive (+) or negative (−) heat flow direction within the heat sink bracket, a simple rule can be followed. Assume that all of the heat is flowing into each node as it is being examined. This will require changing the assumed heat flow direction in the various conductors, as the heat balance equations are written for different nodes. Starting at node 1, assume that the heat from node 2 flows into node 1, the heat from node 3 flows into node 1, and the heat from Q_1 flows into node 1. This is equivalent to assuming that the temperatures at node points 2 and 3 are higher than the temperature at node point 1. The heat flow relation shown by Eq. 3.27 is then used at each node.

The heat flow balance at node 1 is as follows:

$$k'_A(t_2 - t_1) + k'_B(t_3 - t_1) + Q_1 = 0$$

This leads to

$$(k'_A + k'_B)t_1 - k'_A t_2 - k'_B t_3 = Q_1 \tag{3.28}$$

The heat flow balance at node 2 is as follows:

$$k'_A(t_1 - t_2) + k'_C(t_4 - t_2) + Q_2 = 0$$

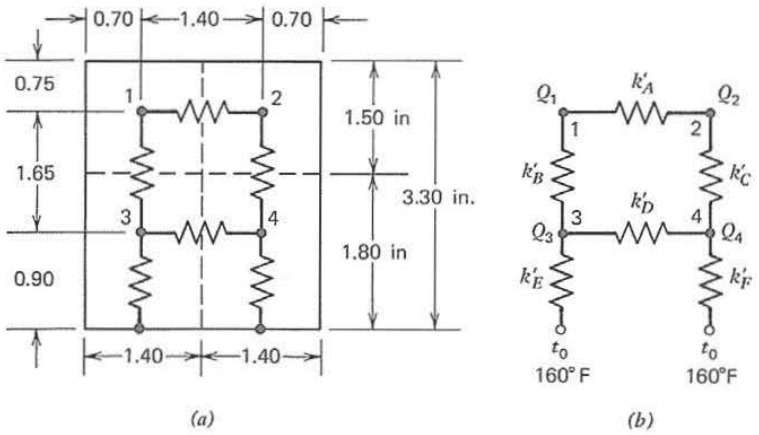


Figure 3.13 Mathematical model of the component bracket.

SEC et al. v. MRI
SEC Exhibit 1013.0069
IPR2023-00221

This leads to

$$-k'_A t_1 + (k'_A + k'_C)t_2 - k'_C t_4 = Q_2 \tag{3.29}$$

The heat flow balance at node 3 is as follows:

$$\begin{aligned} k'_B(t_1 - t_3) + k'_D(t_4 - t_3) + k'_E(t_0 - t_3) + Q_3 &= 0 \\ -k'_B t_1 + (k'_B + k'_D + k'_E)t_3 - k'_D t_4 &= Q_3 + k'_E t_0 \end{aligned} \tag{3.30}$$

The heat flow balance at node 4 is as follows:

$$\begin{aligned} k'_C(t_2 - t_4) + k'_D(t_3 - t_4) + k'_F(t_0 - t_4) + Q_4 &= 0 \\ -k'_C t_2 - k'_D t_3 + (k'_C + k'_D + k'_F)t_4 &= Q_4 + k'_F t_0 \end{aligned} \tag{3.31}$$

There are now four equations, 3.28 through 3.31, with four unknowns, t_1 through t_4, which can be solved simultaneously once all the conductances k'_A through k'_F have been determined. Each conductance can be determined using Figure 3.12 with Eq. 3.26 for English units and metric units. Start with conductance k'_A.

Given $K_A = 166$ Btu/hr ft °F $= 0.686$ cal/sec cm°C (copper conductivity)
$A_A = (1.5)(0.093) = 0.139$ in² (cross section area)
$A_A = 0.000965$ ft² $= 0.897$ cm²
$L_A = 1.40$ in $= 0.117$ ft $= 3.56$ cm (length)

In English units:

$$k'_A = \frac{(166 \text{ Btu/hr ft °F})(0.000965 \text{ ft}^2)}{0.117 \text{ ft}} = 1.37 \frac{\text{Btu}}{\text{hr °F}} \tag{3.32}$$

In metric units:

$$k'_A = \frac{(0.686 \text{ cal/sec cm °C})(0.897 \text{ cm}^2)}{3.56 \text{ cm}} = 0.173 \frac{\text{cal}}{\text{sec °C}} \tag{3.32a}$$

Calculate conductance k'_B.

Given $K_B = 166$ Btu/hr ft °F $= 0.686$ cal/sec cm°C (copper conductivity)
$A_B = (1.40)(0.093) = 0.130$ in² (cross section area)
$A_B = 0.000903$ ft² $= 0.839$ cm²
$L_B = 1.65$ in $= 0.137$ ft $= 4.19$ cm (length)

SEC et al. v. MRI
SEC Exhibit 1013.0070
IPR2023-00221

In English units:

$$k'_B = \frac{(166)(0.000903)}{0.137} = 1.09 \frac{\text{Btu}}{\text{hr °F}} \tag{3.33}$$

In metric units:

$$k'_B = \frac{(0.686)(0.839)}{4.19} = 0.137 \frac{\text{cal}}{\text{sec °C}} \tag{3.33a}$$

Calculate conductance k'_E.

Given $K_E = 166$ Btu/hr ft °F $= 0.686$ cal/sec cm °C (copper conductivity)
$A_E = (1.40)(0.093) = 0.130$ in² (cross section area)
$A_E = 0.000903$ ft² $= 0.839$ cm²
$L_E = 0.90$ in $= 0.075$ ft $= 2.29$ cm (length)

In English units:

$$k'_E = \frac{(166)(0.000903)}{0.075} = 2.00 \frac{\text{Btu}}{\text{hr °F}} \tag{3.34}$$

In metric units:

$$k'_E = \frac{(0.686)(0.839)}{2.29} = 0.251 \frac{\text{cal}}{\text{sec °C}} \tag{3.34a}$$

Calculate conductance k'_D on the copper plate.

Given $A_D = (1.80)(0.093) = 0.167$ in² (cross section area)
$A_D = 0.00116$ ft² $= 1.077$ cm²
$L_D = 1.40$ in $= 0.117$ ft $= 3.56$ cm (length)

In English units:

$$k'_D = \frac{(166)(0.00116)}{0.117} = 1.64 \frac{\text{Btu}}{\text{hr °F}} \tag{3.35}$$

In metric units:

$$k'_D = \frac{(0.686)(1.077)}{3.56} = 0.207 \frac{\text{cal}}{\text{sec °C}} \tag{3.35a}$$

Because of symmetry

$$k'_C = k'_B \quad \text{and} \quad k'_F = k'_E \tag{3.36}$$

SEC et al. v. MRI
SEC Exhibit 1013.0071
IPR2023-00221

The power dissipations at each of the four nodes are as follows (see Table 1.1 for power conversions):

$$\left.\begin{aligned} Q_1 &= 1 \text{ watt} = 3.41\,\frac{\text{Btu}}{\text{hr}} = 0.239\,\frac{\text{cal}}{\text{sec}} \\ Q_2 &= 3 \text{ watts} = 10.24\,\frac{\text{Btu}}{\text{hr}} = 0.717\,\frac{\text{cal}}{\text{sec}} \\ Q_3 &= 5 \text{ watts} = 17.06\,\frac{\text{Btu}}{\text{hr}} = 1.19\,\frac{\text{cal}}{\text{sec}} \\ Q_4 &= 7 \text{ watts} = 23.89\,\frac{\text{Btu}}{\text{hr}} = 1.67\,\frac{\text{cal}}{\text{sec}} \end{aligned}\right\} \tag{3.37}$$

Substitute Eqs. 3.32, 3.33, and 3.37 into Eq. 3.28 using English units.

$$2.46t_1 - 1.37t_2 - 1.09t_3 = 3.41\,\frac{\text{Btu}}{\text{hr}} \tag{3.38}$$

Substitute Eqs. 3.32, 3.36, and 3.37 into Eq. 3.29 using English units.

$$-1.37t_1 + 2.46t_2 - 1.09t_4 = 10.24\,\frac{\text{Btu}}{\text{hr}} \tag{3.39}$$

Substitute Eqs. 3.33, 3.35, 3.34, and 3.37 into Eq. 3.30 using English units.

$$\begin{aligned} -1.09t_1 + 4.73t_3 - 1.64t_4 &= 17.06 + (2.0)(160) \\ -1.09t_1 + 4.73t_3 - 1.64t_4 &= 337.06\,\frac{\text{Btu}}{\text{hr}} \end{aligned} \tag{3.40}$$

Substitute Eqs. 3.36, 3.35, and 3.37 into Eq. 3.31 using English units.

$$\begin{aligned} -1.09t_2 - 1.64t_3 + 4.73t_4 &= 23.89 + (2.0)(160) \\ -1.09t_2 - 1.64t_3 + 4.73t_4 &= 343.89\,\frac{\text{Btu}}{\text{hr}} \end{aligned} \tag{3.41}$$

Equations 3.38 through 3.41 must be solved simultaneously to obtain the temperatures at node points 1 through 4. Several different methods can be used if a calculator is not available. For example, using the substitution method, Eqs. 3.38 and 3.39 can be combined to eliminate t_1.

$$\begin{aligned} 2.46t_1 - 1.37t_2 - 1.09t_3 &= 3.41 && \text{(ref. Eq. 3.38)} \\ -2.46t_1 + 4.42t_2 - 1.96t_4 &= 18.39 && \left(\text{ref. Eq. 3.39} \times \frac{2.46}{1.37}\right) \\ \hline + 3.05t_2 - 1.09t_3 - 1.96t_4 &= 21.80 \end{aligned} \tag{3.42}$$

SEC et al. v. MRI
SEC Exhibit 1013.0072
IPR2023-00221

Next, combine Eqs. 3.39 and 3.40, eliminating t_4.

$$\begin{aligned} 2.06t_1 - 3.70t_2 + 1.64t_4 &= -15.41 \\ -1.09t_1 + 4.73t_3 - 1.64t_4 &= 337.06 \\ \hline 0.97t_1 - 3.70t_2 + 4.73t_3 &= 321.65 \end{aligned} \tag{3.43}$$

Combine Eqs. 3.40 and 3.41, eliminating t_3.

$$\begin{aligned} -1.09t_1 + 4.73t_3 - 1.64t_4 &= 337.06 \\ -3.14t_2 - 4.73t_3 + 13.64t_4 &= 991.83 \\ \hline -1.09t_1 - 3.14t_2 + 12.00t_4 &= 1328.89 \end{aligned} \tag{3.44}$$

Combine Eqs. 3.42 and 3.43, eliminating t_2.

$$\begin{aligned} 3.70t_2 - 1.32t_3 - 2.38t_4 &= 26.44 \\ 0.97t_1 - 3.70t_2 + 4.73t_3 &= 321.65 \\ \hline 0.97t_1 + 3.41t_3 - 2.38t_4 &= 348.09 \end{aligned} \tag{3.45}$$

Combine Eqs. 3.43 and 3.44, eliminating t_2.

$$\begin{aligned} 0.97t_1 - 3.70t_2 + 4.73t_3 &= 321.65 \\ 1.28t_1 + 3.70t_2 - 14.14t_4 &= -1565.89 \\ \hline 2.25t_1 + 4.73t_3 - 14.14t_4 &= -1244.24 \end{aligned} \tag{3.46}$$

Combine Eqs. 3.41 and 3.42, eliminating t_2.

$$\begin{aligned} -3.05t_2 - 4.59t_3 + 13.23t_4 &= 962.26 \\ 3.05t_2 - 1.09t_3 - 1.96t_4 &= 21.80 \\ \hline -5.68t_3 + 11.27t_4 &= 984.06 \end{aligned} \tag{3.47}$$

Combine Eqs. 3.45 and 3.46, eliminating t_1.

$$\begin{aligned} -2.25t_1 - 7.91t_3 + 5.52t_4 &= -807.42 \\ 2.25t_1 + 4.73t_3 - 14.14t_4 &= -1244.24 \\ \hline -3.18t_3 - 8.62t_4 &= -2051.66 \end{aligned} \tag{3.48}$$

Combine Eqs. 3.47 and 3.48, eliminating t_3. This will give the temperature at node point 4.

$$\begin{aligned} -5.68t_3 + 11.27t_4 &= 984.06 \\ +5.68t_3 + 15.40t_4 &= 3664.60 \\ \hline 26.67t_4 &= 4648.66 \\ t_4 &= 174.3°\text{F} \end{aligned} \tag{3.49}$$

SEC et al. v. MRI
SEC Exhibit 1013.0073
IPR2023-00221

Substitute Eq. 3.49 into Eq. 3.48 for the temperature at mounting point 3, which is also node. 3.

$$-3.18t_3 - 8.62(174.3) = -2051.66$$

$$t_3 = 172.9°F \tag{3.50}$$

Substitute Eqs. 3.49 and 3.50 into Eq. 3.46 for the temperature at mounting point 1.

$$2.25t_1 + 4.73(172.9) - 14.14(174.3) = -1244.24$$

$$t_1 = 178.8°F \tag{3.51}$$

Substitute Eqs. 3.49 and 3.50 into Eq. 3.42 for the temperature at mounting point 2.

$$3.05t_2 - 1.09(172.9) - 1.96(174.3) = 21.80$$

$$t_2 = 181.0°F \tag{3.52}$$

The same process can be repeated with the metric units by substituting Eqs. 3.32a through 3.36a and Eq. 3.37 into Eqs. 3.28 through 3.31. This will result in four equations with four unknowns, for temperatures in degrees Celsius.

3.13 HEAT CONDUCTION ACROSS INTERFACES IN AIR

The electronics industry makes extensive use of brackets, heat sinks, and circuit boards for mounting electronic components that are cooled by conduction. This often requires the transfer of heat across interfaces that may be bolted, riveted, or just clamped together. Extreme caution must be used if large amounts of heat will be conducted across these interfaces, because a poor thermal contact will produce a large thermal resistance, which means a high temperature rise. High temperatures will then result in more rapid electrical failures.

An enlarged view of two surfaces in contact will reveal that a very small portion of the total surface is actually in contact, as shown in Figure 3.14.

Most of the heat flow across the interface will take place between the various high points that contact each other. The number of high points per square inch of surface area that contact each other will generally depend upon the flatness, smoothness, hardness, stiffness, and interface pressure between the two mating surfaces [25–28].

The amount of heat that can be conducted across the interface will also depend upon the presence of another material at the interface, such as air, oil, or grease. Many manufacturers use special silicone greases, or pastes,

SEC et al. v. MRI
SEC Exhibit 1013.0074
IPR2023-00221

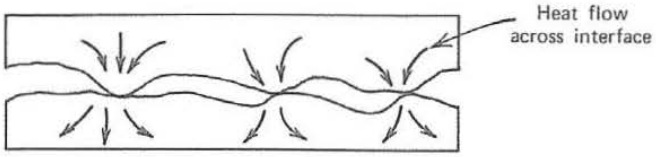


Figure 3.14 Enlarged view of two surfaces in contact.

which are filled with thermally conductive metal oxides to improve the overall heat transfer. These materials will not dry out or harden at temperatures well above the maximum operating temperatures of most electronic components. Water, which has a high thermal conductivity, is excellent as an interface conductor, but it evaporates too easily.

Sometimes large surfaces, which are not flat, must be mated using a dry interface. A soft, thin foil such as aluminum, indium, or copper about 0.002 in (0.0051 cm) thick will often improve the heat transfer across the interface.

Large surfaces should use stiff interface sheets to reduce the amount of warping. Large surfaces may also be improved with the use of an interface material made of a soft wire screen impregnated with a high carbon rubber. A thin beryllium copper sheet stamped with thousands of little spring fingers has worked well in many applications.

It is convenient to express the thermal interface characteristics in terms of a conductance coefficient h_i so that the heat transfer relation shown in Eq. 3.53 can be used [28].

$$Q = h_i A\ \Delta t \tag{3.53}$$

Given

Item	*Symbol*	*English Units*	*Metric units*
Power dissipation	Q	$\frac{\text{Btu}}{\text{hr}}$	$\frac{\text{cal}}{\text{sec}}$
Interface conductance	h_i	$\frac{\text{Btu}}{\text{hr ft}^2\ ^\circ\text{F}}$	$\frac{\text{cal}}{\text{sec cm}^2\ ^\circ\text{C}}$
Area	A	ft^2	cm^2
Temperature difference	Δt	°F	°C

A considerable number of test data are available for the interface conductances of different types of joints, with different metals, different surface finishes, and different interface materials. These data are generally based upon the interface pressure between the two mating surfaces and the finish of these surfaces. The finish is measured by the rms (root mean square)

SEC et al. v. MRI
SEC Exhibit 1013.0075
IPR2023-00221

index value of the small variations in the heights and depths of the many peaks and valleys found on all surfaces. Some typical rms values for various surfaces are shown in Table 3.4 [18, 19].

In general, a smooth surface is considered to have an rms finish less than about 15, and a rough surface is considered to have an rms finish greater than about 80. Some typical test data are shown in Figures 3.15, 3.16, and 3.17.

Table 3.5 shows typical interface conductances for various materials with an interface pressure of 10 psi.

An examination of the curves and tables shows that for a given surface finish and interface pressure, hard materials such as steel will have a lower interface conductance than will softer materials such as aluminum. Also, when the surface finish and surface flatness are improved, the interface conductance will rise substantially.

The interface conductance will increase when the clamping forces are increased. Low clamping forces will have lower conductance values than will high clamping forces. Smooth surfaces have high interface conductances with relatively low clamping forces. Increasing the clamping force between smooth surfaces does not greatly improve the interface conductance. Rough surfaces, on the other hand, have low interface conductances with low interface pressures. Increasing the clamping force between rough surfaces improves the interface conductance substantially [27, 28].

Figure 3.16 shows that for a given interface pressure and surface finish, a higher heat density will result in a higher interface conductance. Test data also show that the interface conductance rises substantially when the temperature is increased.

Most of the test data between two structural members, which are riveted,

Table 3.4 Typical Surface Finishes for Different Operations

Operation	Surface Finish (μin rms)
Lap	4
Grind	10
Rolled surface	15–20
Lathe, smooth cut	80
Lathe, rough cut	120
Milling machine, rough cut	125
Investment casting	125–240
File	240
Sand casting	240–300
Shaper	1000

SEC et al. v. MRI
SEC Exhibit 1013.0076
IPR2023-00221

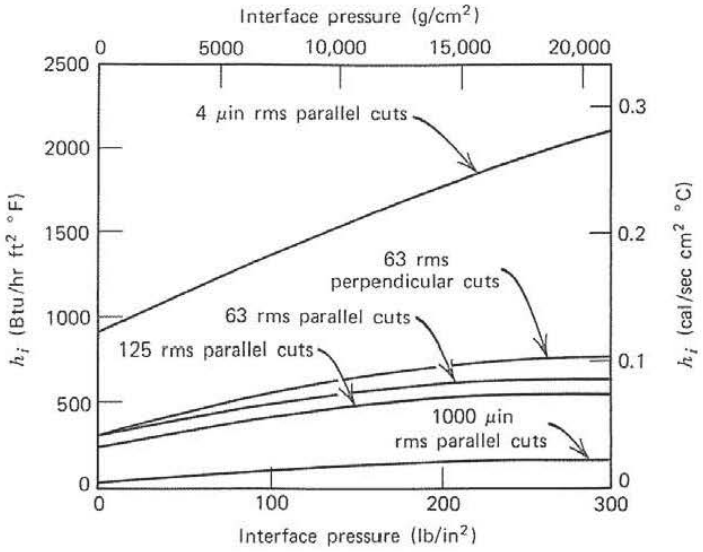


Figure 3.15 Interface conductance for parallel and perpendicular machine cuts for 18% carbon steel at 100°C (212°F) at sea level conditions.

bolted, or spot welded together, will show a substantial amount of variation in the interface conductance values. Structural members that appear to be identical in all other respects will often have large variations in their interface conductances. These values are heavily dependent upon the surface contact area and interface pressure in the sections immediately under and around the riveted, bolted, and spot welded members. These parameters are difficult to control exactly, even with automatic machines. Therefore, the test data for structural members will show a great deal of variation [27, 28].

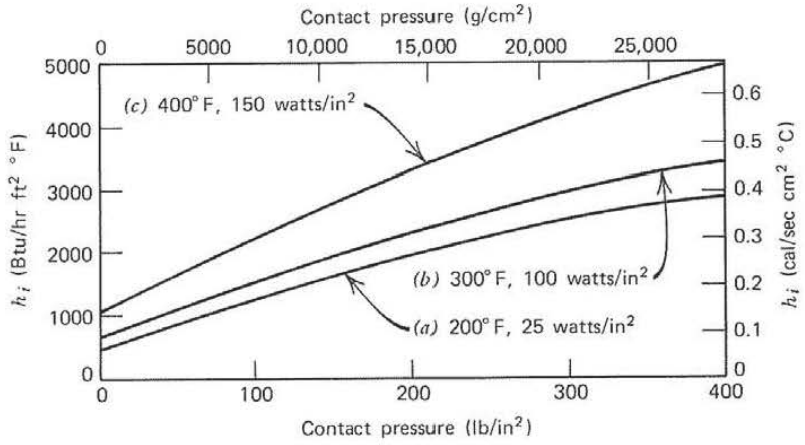


Figure 3.16 Interface conductance for 7075 T6 aluminum at sea level at various temperatures with 65 μin rms surface finish.

SEC et al. v. MRI
SEC Exhibit 1013.0077
IPR2023-00221

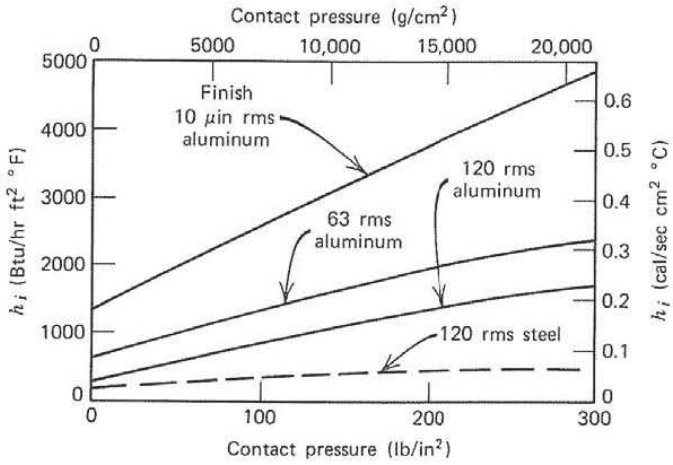


Figure 3.17 Interface conductance for various interface pressures at sea level.

3.14 SAMPLE PROBLEM—TEMPERATURE RISE ACROSS A BOLTED INTERFACE

Determine the temperature rise across the bolted interfaces of an aluminum bracket that is bolted to a chassis side wall. The bolts will produce an interface pressure of 200 lb/in² (14,074 g/cm²), as shown in Figure 3.18. The total power dissipation of the components on the bracket is 12 watts. The surface roughness at the bolted interfaces is about 65 μin rms. The bracket is in a confined area where very little heat will be lost by convection or radiation.

SOLUTION

Equation 3.53 can be used to determine the temperature rise across the bolted interfaces, together with Figure 3.16.

Given $Q = 12$ watts $= 40.9$ Btu/hr $= 2.87$ cal/sec (heat input)
$A = 2[(0.50)(0.50) - (\pi/4)(0.150)^2] = 0.464$ in²
$A = 0.00322$ ft² $= 2.99$ cm² (interface area)
$\frac{Q}{A} =$ 12 watts/(0.464 in²) $= 25.8$ watts/in² $= 4.01$ watts/cm² (power density)
$h_i = 2000$ Btu/hr ft² °F $= 0.270$ cal/sec cm² °C (interface conductance; Figure 3.16, curve A)

Substitute into Eq. 3.53 for the temperature rise in English units.

$$\Delta t = \frac{40.9 \text{ Btu/hr}}{(2000 \text{ Btu/hr ft}^2 \text{ °F})(0.00322 \text{ ft}^2)} = 6.4\text{°F} \tag{3.54}$$

SEC et al. v. MRI
SEC Exhibit 1013.0078
IPR2023-00221

Table 3.5 Interface Conductance for Various Materials with an Interface Pressure of 10 psi

Material	Rms Surface 1	Rms Surface 2	Interface Conductance Btu/hr ft² °F Dry	Btu/hr ft² °F Oil[a]	cal/sec cm² °C Dry	cal/sec cm² °C Oil[a]
SAE 4141 steel	3	3	2200		0.297	
	70	85	400	1350	0.054	0.182
5051 aluminum	16	17	1800		0.243	
	60	60	1300	2000	0.175	0.270
	15	90	800	1600	0.108	0.216
Bronze AMS 4846	70	80	800	1200	0.108	0.162

[a] *Oil thermal conductivity* K = *0.073 Btu/hr ft °F = 0.00030 cal/sec cm °C.*

In metric units:

$$\Delta t = \frac{2.87 \text{ cal/sec}}{(0.270 \text{ cal/sec cm}^2 \text{ °C})(2.99 \text{ cm}^2)} = 3.5\text{°C} \tag{3.54a}$$

Air is a relatively good conductor of heat. Often, the air at the interface of two surfaces will carry a large part of the heat if the surfaces are very rough or the interface pressure is very low. This can be demonstrated by considering a slight modification of the preceding sample problem.

3.15 SAMPLE PROBLEM—TEMPERATURE RISE ACROSS A SMALL AIR GAP

In the preceding sample problem, there is dirt on the bracket, so that there is poor physical contact between the component bracket and the heat sink.

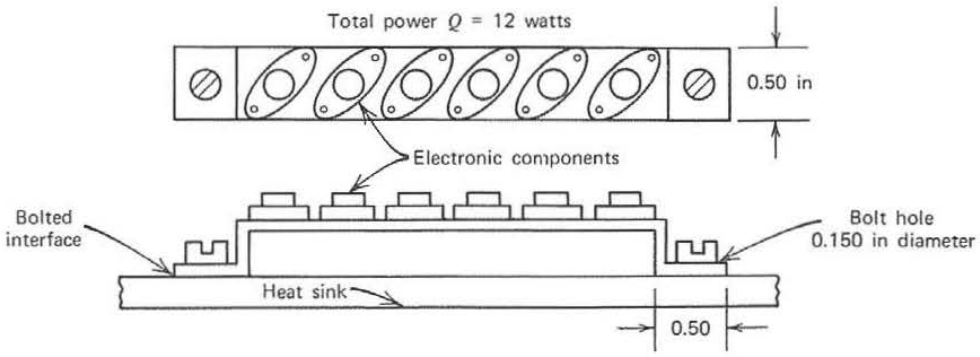


Figure 3.18 Component bracket bolted to a heat sink.

SEC et al. v. MRI
SEC Exhibit 1013.0079
IPR2023-00221

There is an air gap of 0.001 in (0.0025 cm) under the bracket. Determine the temperature rise across the air gap.

SOLUTION

The temperature rise across the air gap is determined with the use of the conduction relation shown in Eq. 3.1.

Given $Q = 12$ watts $= 40.9$ Btu/hr $= 2.87$ cal/sec (heat input)
$L = 0.001$ in $= 0.000083$ ft $= 0.0025$ cm (length of gap)
$K = 0.017$ Btu/hr ft °F $= 0.0000702$ cal/sec cm °C (air conductivity at 160°F)
$A = 0.464$ in^2 $= 0.00322$ ft^2 $= 2.99$ cm^2 (interface area)

In English units:

$$\Delta t = \frac{(40.9 \text{ Btu/hr})(0.000083 \text{ ft})}{(0.017 \text{ Btu/hr ft °F})(0.00322 \text{ ft}^2)} = 62.0°\text{F} \tag{3.55}$$

In metric units:

$$\Delta t = \frac{(2.87 \text{ cal/sec})(0.0025 \text{ cm})}{(0.0000702 \text{ cal/sec cm°C})(2.99 \text{ cm}^2)} = 34.2°\text{C} \tag{3.55a}$$

When a thermal grease is used at the interface of the mounting bracket, the temperature rise across the interface can be reduced. Thermal greases have thermal conductivities that generally range from about 0.2 to about 0.9 Btu/hr ft °F (0.00083 to 0.0037 cal/sec cm °C). For an average value of about 0.5 Btu/hr ft °F (0.0021 cal/sec cm °C), with an average thickness of 0.002 in (0.000166 ft), the temperature rise across the interface will be

$$\Delta t = \frac{(40.9)(0.000166)}{(0.5)(0.00322)} = 4.2°\text{F}\ (2.3°\text{C}) \tag{3.56}$$

To get the same temperature rise across the bracket interface without the thermal grease, a better surface finish and a higher interface pressure will be required.

3.16 HEAT CONDUCTION ACROSS INTERFACES AT HIGH ALTITUDES

The amount of heat that can be transferred between two clamped surfaces is often sharply reduced at high altitudes. No heat will be conducted between two adjacent surfaces in a hard vacuum environment if they are not

SEC et al. v. MRI
SEC Exhibit 1013.0080
IPR2023-00221

in intimate contact with each other. Small amounts of heat may be radiated from one interface to another, depending upon the temperatures and the physical properties of the surfaces. However, conduction heat transfer requires some medium—solid, liquid, or gas—through which the heat can be transferred.

The change in the interface conductance between two adjacent surfaces with a low interface pressure, for various altitude conditions from sea level to a hard vacuum, is shown in Figure 3.19. This curve shows that special care must be used at the thermal interfaces of electronic assemblies that will be cooled by conduction in outer space applications. It also shows that there is very little change in the interface conduction for altitudes up to about 80,000 ft. Above that altitude, the interface conductance values begin to drop off very rapidly [25, 26].

Rigid thermal interfaces must be provided for surfaces that are expected to transfer heat in a vacuum environment. Test data show that typical sheet metal types of structures can have interface conductances that are only 10% of what they are at sea level. Some test data for different surface finishes are shown in Figure 3.20.

Most of the interface test data found in the literature are based upon very small and very rigid structural elements. This type of test sample is easy to control. The interface pressure can be established accurately, it is uniform across the entire surface, and different finishes are easy to obtain.

In the real world of electronic structures, the interface areas are often quite large, the interface pressures are not uniform, and the surfaces are not stiff and flat. Because of these factors, the actual interface conductances for large sheet metal structures will generally be much lower than most published test data show. Tests on large bolted sheet metal plates show that typical interface conductances will probably be in the range of a few hundred

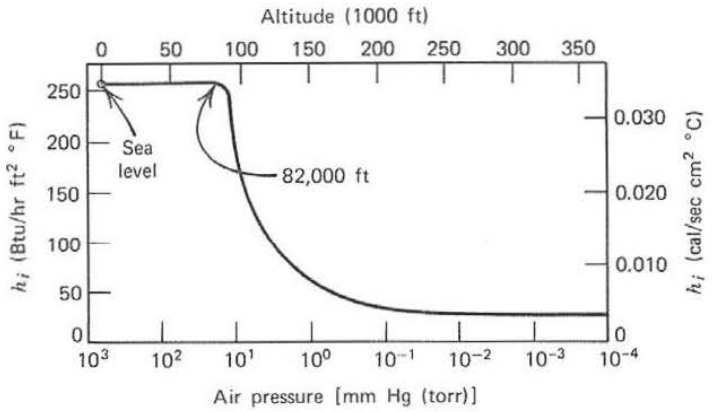


Figure 3.19 Interface conductance with contact pressure of 2 psi (140.7 g/cm²). Copper, 0.011 TIR, 10 μin rms finish in contact with brass, 0.004 TIR, 22 μin rms finish, as a function of altitude.

SEC et al. v. MRI
SEC Exhibit 1013.0081
IPR2023-00221

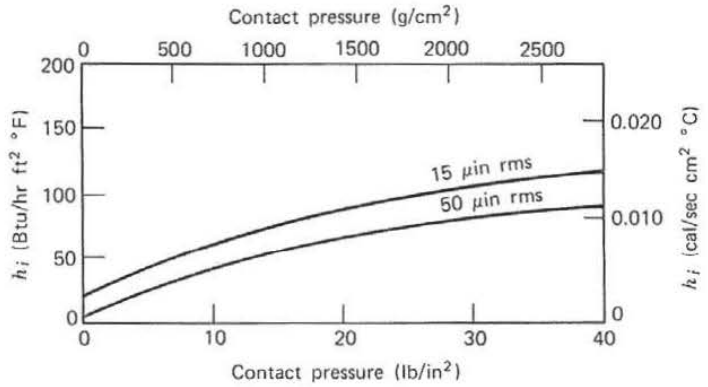


Figure 3.20 Interface conductance for 2024 T6 aluminum blocks at 10^{-4} torr.

rather than in the range of a few thousand (in English units). Some test data for large bolted sheet metal plates are shown in Table 3.6. Additional test data on various interfaces are shown in Tables 3.7 and 3.8.

When it is necessary to conduct heat across large nonrigid surfaces in a hard vacuum, some special heat transfer interfaces may have to be fabricated. One type of assembly that can provide an interface conductance of about 100 Btu/hr ft² °F (0.0135 cal/sec cm² °C) with an interface pressure of only 10 lb/in² (703.7 g/cm²) in a hard vacuum (10^{-6} torr) is shown in Figure

Table 3.6 Average Interface Conductance for Two As-Received Aluminum Plates 10 × 20 × 0.25 in Thick (25.4 × 50.8 × 0.635 cm) Bolted Only at the Four Corners and Using Various Interface Materials

	Interface Conductance (h_i)			
	In Air at Sea Level		Vacuum (10^{-6} torr)	
Interface Material	Btu / hr ft² °F	cal / sec cm² °C	Btu / hr ft² °F	cal / sec cm² °C
0.002 in (0.0051 cm) copper or aluminum foil	300	0.0405	100	0.0135
0.020 in (0.0508 cm) rubber-impregnated aluminum screen	200	0.027	40	0.0054
0.005 in (0.0127 cm) beryllium copper foil with 1000 small fingers	200	0.027	40	0.0054
No interface material	200	0.027	20	0.0027

SEC et al. v. MRI
SEC Exhibit 1013.0082
IPR2023-00221

Table 3.7 Average Interface Conductance for 5052 (1/4 Hard) Aluminum Strip 3.0 × 0.25 × 0.06 in Thick (7.62 × 0.635 × 0.152 cm) Clamped to a 0.25 in (0.635 cm) Thick Aluminum Plate[a]

	Interface Conductance (h_i)			
	Sea Level		10^{-5} Torr (mm hg)	
Interface Pressure	Btu / hr ft² °F	cal / sec cm² °C	Btu / hr ft² °F	cal / sec cm² °C
500 psi	350	0.0472	170	0.0230
1000 psi	520	0.0702	340	0.0459

[a] *Surface finish on both members about 15 μin rms.*

3.21. The interface assembly consists of a soft thin copper foil that is wrapped around a silicone rubber tube and soldered to a flat copper sheet.

3.17 OUTGASSING AT HIGH ALTITUDES

Many materials tend to release trapped gasses when the surrounding pressure is reduced. This applies to many common materials, such as rubbers, epoxies, greases, plastics, and even metals. More trapped gasses are released when the outside pressures are lower.

Outgassing may not be a problem unless there is optical equipment in the area or a hard vacuum condition is required for thermal isolation or electrical isolation.

Camera lenses and mirrors, which require a smooth clear surface, can often become coated if outgassing materials are present. The coating process may take a few days or a few weeks to develop, depending upon the

Table 3.8 Average Interface Conductance for a Small Chassis with a 5052 Aluminum Base Plate 4 × 6 × 0.125 in Thick (10.16 × 15.24 × 0.317 cm) Bolted to a 1.0 in (2.54 cm) Thick Cold Plate with Four No. 10–32 Screws at the Corners[a]

	Interface Conductance (h_i)			
	Sea Level		10^{-5} torr (mm hg)	
Interface Condition	English Units	Metric Units	English Units	Metric Units
Bolted at the four corners	220	0.0297	50	0.00675

[a] *Surface finish on both members about 32 μin rms.*

SEC et al. v. MRI
SEC Exhibit 1013.0083
IPR2023-00221

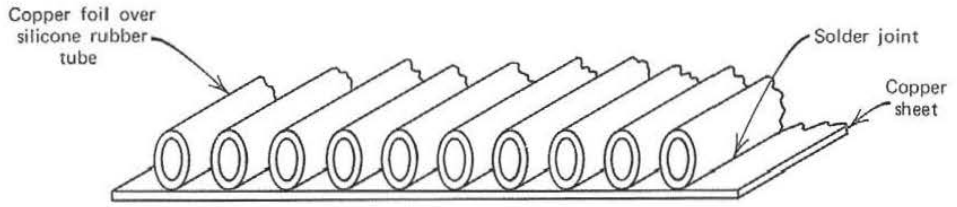


Figure 3.21 Resilient assembly for conducting heat across large nonrigid interfaces in a hard vacuum.

conditions. Once the optical surfaces are coated, however, the performance of the equipment will degrade substantially.

Outgassing can have the effect of increasing the pressure in a local area. Some electronic applications require thermal isolation to permit high temperatures to be developed with very low heating element power. If outgassing is present, it will substantially increase the heat transfer by gaseous conduction in small confined areas, so that the required high temperatures may never be reached.

In electrical applications, it is possible for arcing to develop if outgassing occurs in a vacuum environment. Arcing may not occur at sea level conditions or in a hard vacuum. However, with outgassing present, the effective altitude in the immediate area is changed so that ionization in the gas can occur and arcing will develop.

3.18 CIRCUIT BOARD EDGE GUIDES

Plug-in PCBs are often used with guides, which help to align the PCB connector with the chassis connector. These guides, which are usually fastened to the side walls of a chassis, grip the edges of the PCB as it is inserted and removed. If there is enough contact pressure and surface area at the interface between the edge guide and the PCB, the edge guide can be used to conduct heat away from the PCB. A typical installation is shown in Figure 3.22.

Plug-in PCBs must engage a blind connector, which is usually fastened to the chassis. If the chassis connector is rigid, with no floating provisions, the edge guide must provide that float, or many connector pins will be bent and broken. Rigid chassis connectors are generally used for electrical connections because wire wrap harnesses, flex tape harnesses, and multilayer master interconnecting mother boards are being used for production systems. These interconnections cannot withstand the excessive motion required for a floating connector system. Therefore, the floating mechanism is often built into the circuit board edge guides.

High pressures with dry, smooth interfaces must be provided in the edge connector if it is to be a good heat conductor. Thermal greases are unde-

SEC et al. v. MRI
SEC Exhibit 1013.0084
IPR2023-00221

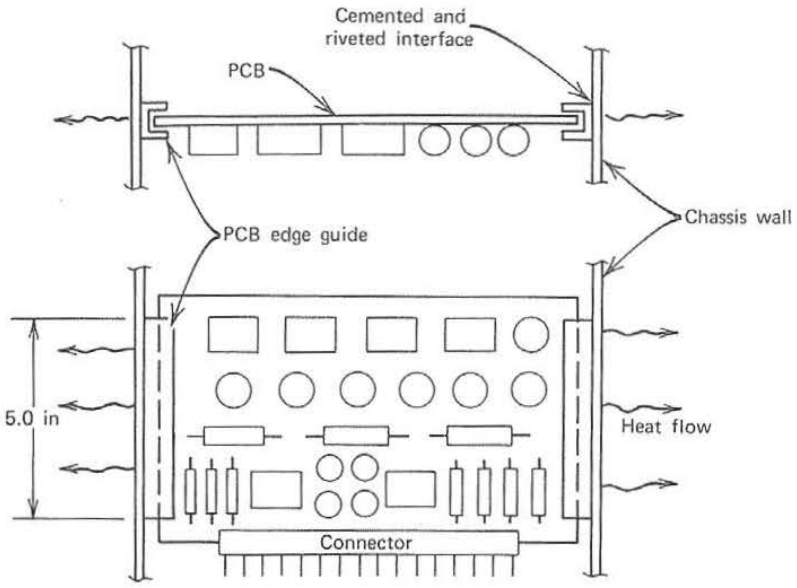


Figure 3.22 Plug-in PCB assembly with board edge guides.

sirable at these interfaces because they are very messy to handle and must be replaced every time a PCB is pulled for inspection or repair.

Many different types of circuit board edge guides are used by the electronics industry. The thermal resistance across a typical guide is rather difficult to calculate accurately, so that tests are used to establish these values. The units used to express the thermal resistance across an edge guide are °C in/watt. When the length of the guide is expressed in inches and the heat flow is in watts, the temperature rise is in degrees Celsius. Figure 3.23 shows four different types of edge guides, with the thermal resistance across each guide.

These board edge guides are generally satisfactory for applications at sea level or medium altitudes up to about 50,000 ft. At altitudes of 100,000 ft, test data show that the resistance values for the guides in Figure 3.23*a–c* will increase about 30%. The wedge clamp shown in Figure 3.23*d* will have about a 5% increase.

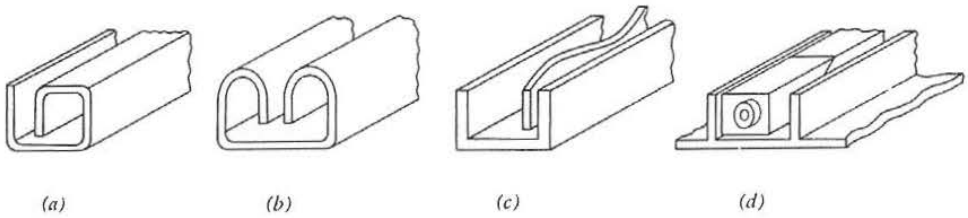


Figure 3.23 Board edge guides with typical thermal resistance. (*a*) G guide, 12°C in/watt; (*b*) B guide, 8°C in/watt; (*c*) U guide, 6°C in/watt; (*d*) wedge clamp, 2°C in/watt.

SEC et al. v. MRI
SEC Exhibit 1013.0085
IPR2023-00221

For hard vacuum conditions, where the pressure is less than 10^{-6} torr (mm hg), as normally experienced in outer space work, some of these edge guides may not be satisfactory, except for very low power dissipations. Outer space environments generally require very high pressure interfaces, with flat and smooth surfaces, to conduct heat effectively. Wedge clamps are very effective here because they can produce high interface pressures. Bolted interfaces are also used extensively because even small bolts can produce high forces. However, bolted interfaces lack the plug-in feature that is considered very desirable for maintenance.

3.19 SAMPLE PROBLEM—TEMPERATURE RISE ACROSS A PCB EDGE GUIDE

Determine the temperature rise across the PCB edge guide (from the edge of the PCB to the chassis wall) for the assembly shown in Figure 3.22. The edge guide is 5.0 in long, type *c*, as shown in Figure 3.23. The total power dissipation of the PCB is 10 watts, uniformly distributed, and the equipment must operate at 100,000 ft.

SOLUTION

Since there are two edge guides, half of the total power will be conducted through each guide. The temperature rise at sea level conditions can be determined from Eq. 3.57.

$$\Delta t = \frac{RQ}{L} \tag{3.57}$$

Given $R = 6$ °C in/watt
$Q = 10/2 = 5$ watts (on half of the PCB)
$L = 5$ in (length of one guide)

$$\Delta t = \frac{(6\ ^\circ\text{C in/watt})(5\ \text{watts})}{5\ \text{in}} = 6^\circ\text{C (rise at sea level)}$$

At an altitude of 100,000 ft, test data and Figure 3.24 show that the resistance across the edge guide will increase about 30%. The temperature rise at this altitude will then be

$$\Delta t = 1.30(6^\circ\text{C}) = 7.8^\circ\text{C (rise)} \tag{3.58}$$

3.20 HEAT CONDUCTION THROUGH SHEET METAL COVERS

Lightweight electronic boxes often use sheet metal covers to enclose the chassis. Sometimes these covers are expected to conduct the heat away

SEC et al. v. MRI
SEC Exhibit 1013.0086
IPR2023-00221

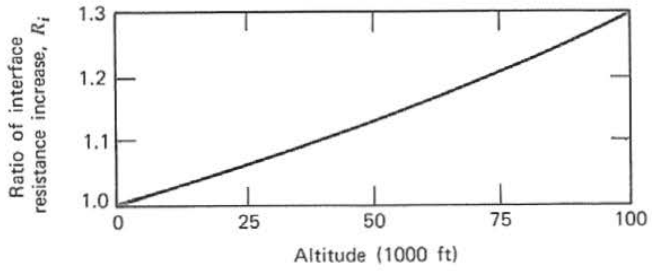


Figure 3.24 Change of board edge guide thermal interface resistance as a function of altitude (types *a–c*, Figure 3.23).

from the chassis for additional cooling. To effectively conduct heat from the chassis to the cover, a flat, smooth, high pressure interface must be provided. The normal surface contact obtained at the interfaces of sheet metal members will not provide an effective heat conduction path. However, if a lip can be formed in the cover or on the side walls of the chassis, the heat conduction path can be substantially improved. Figure 3.25 shows how lips may be formed in the chassis and cover to improve the heat transfer across the interface.

3.21 RADIAL HEAT FLOW

Electronic support structures are not always rectangular in shape. If an electronic system is enclosed within a cylindrical structure, such as a small missile, it would be a waste of space to use a rectangular box for the electronics. The packaging form factor would probably conform to the natural cylindrical shape. Electronic components mounted on the inside surface of a cylindrical chassis would require a radial heat flow path to remove the heat when the heat sink is on the outer surface of the structure. The temperature rise from the inside surface to the outside surface of the

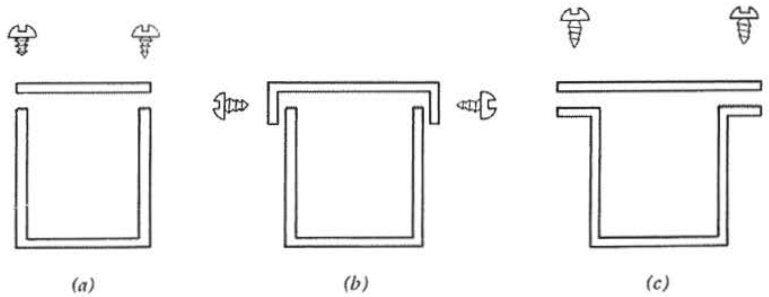


Figure 3.25 Different types of sheet metal covers on electronic boxes. (*a*) Poor; (*b*) good; (*c*) good.

SEC et al. v. MRI
SEC Exhibit 1013.0087
IPR2023-00221

cylindrical structure can be determined from the standard heat flow relation shown by Eq. 3.2 for a concentrated heat load. However, in a radial heat flow system, the cross section area is constantly changing as the radius changes, as shown in Figure 3.26.

An average cross section area can be used to obtain a rough approximation of the temperature rise through the cylindrical wall. If a more accurate evaluation of the temperature rise is desired, the logarithmic mean area must be used, as shown by Eq. 3.59 [15, 17, 24].

$$A_{lm} = \frac{A_{out} - A_{in}}{\log_e(A_{out}/A_{in})} \tag{3.59}$$

When the ratio of A_{out}/A_{in} is less than 2, the average cross section area is within 4% of the logarithmic area.

3.22 SAMPLE PROBLEM—TEMPERATURE RISE THROUGH A CYLINDRICAL SHELL

A hollow steel cylindrical sheel has a group of resistors mounted on the inside surface, as shown in Figure 3.27. Heat is removed from the outside surface of the cylinder, so that the heat flow path is radial through the wall of the cylinder. Determine the temperature rise through the steel cylinder when the power dissipation is 10 watts.

SOLUTION

The logarithmic mean area is determined from Eq. 3.59.

Given

$$A_{out} = \pi D_{out} a = (3.14)(4.2)(1.5) = 19.79 \text{ in}^2 = 0.1374 \text{ ft}^2$$
$$A_{in} = \pi D_{in} a = (3.14)(2.0)(1.5) = 9.42 \text{ in}^2 = 0.0654 \text{ ft}^2$$
$$Q = 10 \text{ watts} = 34.13 \text{ Btu/hr (heat input)}$$
$$K = 24 \text{ Btu/hr ft °F (steel conductivity)}$$
$$A_{lm} = \frac{0.1374 - 0.0654}{\log_e(0.1374/0.0654)} = 0.0970 \text{ ft}^2 \text{ (logarithmic mean area)}$$

Substitute into Eq. 3.2 for the temperature rise through the steel cylin-



Figure 3.26 Hollow cylinder.

SEC et al. v. MRI
SEC Exhibit 1013.0088
IPR2023-00221

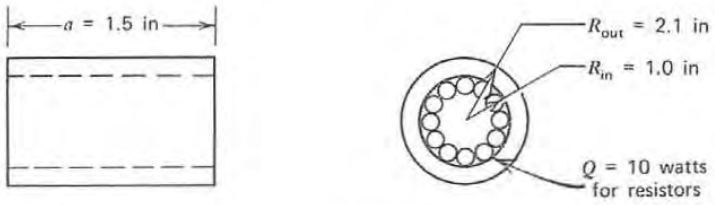


Figure 3.27 Resistors mounted on the inside wall of a hollow cylinder.

drical wall. The length of the heat flow path is the thickness of the steel wall, 1.1 in = 0.0916 ft.

$$\Delta t = \frac{(34.13)(0.0916)}{(24)(0.0970)} = 1.34°\text{F} \tag{3.60}$$

If the average cross section area for radial heat flow through the cylinder is used:

$$A_{av} = \frac{A_{out} + A_{in}}{2} = \frac{0.1374 + 0.0654}{2} = 0.1014\ \text{ft}^2$$

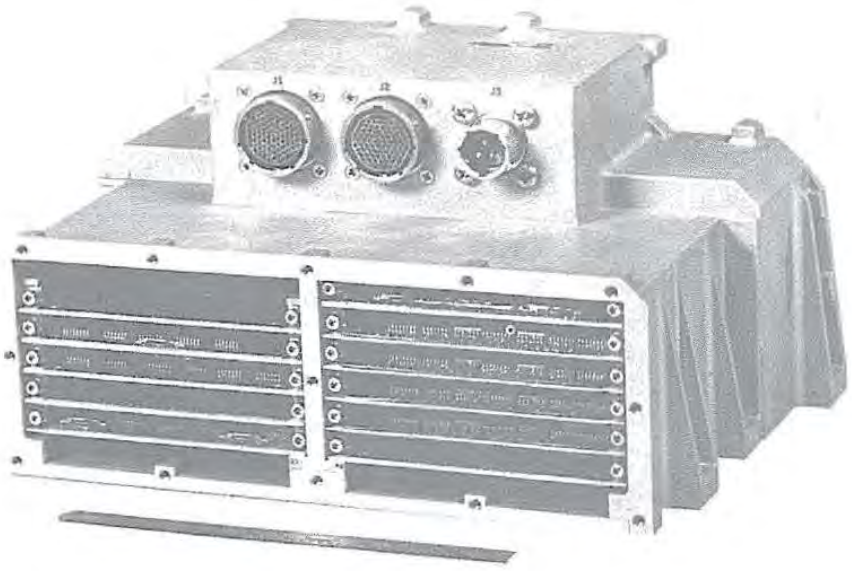


Figure 3.28 Cast chassis with PCBs held by wedge clamps for conduction cooling. (Courtesy Litton Systems, Inc.)

SEC et al. v. MRI
SEC Exhibit 1013.0089
IPR2023-00221

Substitute into Eq. 3.2 for the temperature rise.

$$\Delta t = \frac{(34.13)(0.0916)}{(24)(0.1014)} = 1.28°\text{F} \tag{3.61}$$

In this case there is very little difference in the temperature rise using the logarithmic mean cross section area or the average cross section area for the radial heat flow through the cylinder.

Figure 3.28 shows a rugged cast aluminum chassis with conduction-cooled circuit boards held by wedge clamps.

SEC et al. v. MRI
SEC Exhibit 1013.0090
IPR2023-00221

4

Mounting and Cooling Techniques for Electronic Components

4.1 VARIOUS TYPES OF COMPONENTS

Plug-in types of printed circuit boards (PCBs) have become the standard in the electronics industry. This type of equipment is easy to troubleshoot and maintain. A considerable amount of time and effort has therefore been expended in finding ways to improve the performance and to reduce the cost of PCBs. Many different construction techniques and many different cooling schemes have been devised by the electronic companies involved in the manufacturing of this hardware.

One area that has received a considerable amount of attention is that of mounting electronic components on PCBs. Many different types of components are involved with various shapes and electrical lead wire arrangements. Dual inline packages (DIPs), large scale integrated circuits (LSIs), hybrids, and most recently the microprocessors have been replacing the discrete resistors, capacitors, transistors, and diodes. However, many discrete components are still being used because of their low costs [29].

No matter which groups of components are used for an electronic system, the mounting techniques must provide sufficient cooling to permit the device to operate effectively in its environment. The best method for determining the effectiveness of any cooling system appears to be in the measurement of the case and the junction temperatures of the individual electronic components. Experience has shown that excessive component case temperatures will result in rapid electronic failures. Reliability studies have shown that it is not a good practice to exceed case temperatures of 212°F (100°C) for long periods on electrically operating equipment with any of the components mentioned above, or the failure rates will show a large increase.

SEC et al. v. MRI
SEC Exhibit 1013.0091
IPR2023-00221

When the electronic components are to be cooled by conduction, a good heat flow path must be provided from each component to the ultimate heat sink. The ultimate heat sink may simply be the outside ambient air, or it may be a sophisticated liquid cooled heat exchanger. Each segment along the heat flow path must be examined in detail to be sure that the thermal resistance is low enough to ensure the proper cooling [30].

4.2 MOUNTING COMPONENTS ON PCBS

The electronic components are usually the major heat source in electronic systems. For effective cooling of these components, it is necessary to plan in advance the heat flow mechanism and heat flow path from the component to the sink. The method used for mounting components to PCBs cooled by conduction should provide a low thermal resistance from the component to the PCB. Metal heat sinks are often laminated to the PCB because metals have a high thermal conductivity. The two most popular metals are aluminum and copper, which can be laminated in several different ways, as shown in Figure 4.1.

Some types of components, such as DIPs, LSIs, hybrids, and microprocessors, have a large number of electrical lead wires extending from their cases. About half of the heat dissipated by these components may be conducted through the wires, and about half may be conducted through the bodies of each component. Therefore, when provisions are made for mount-

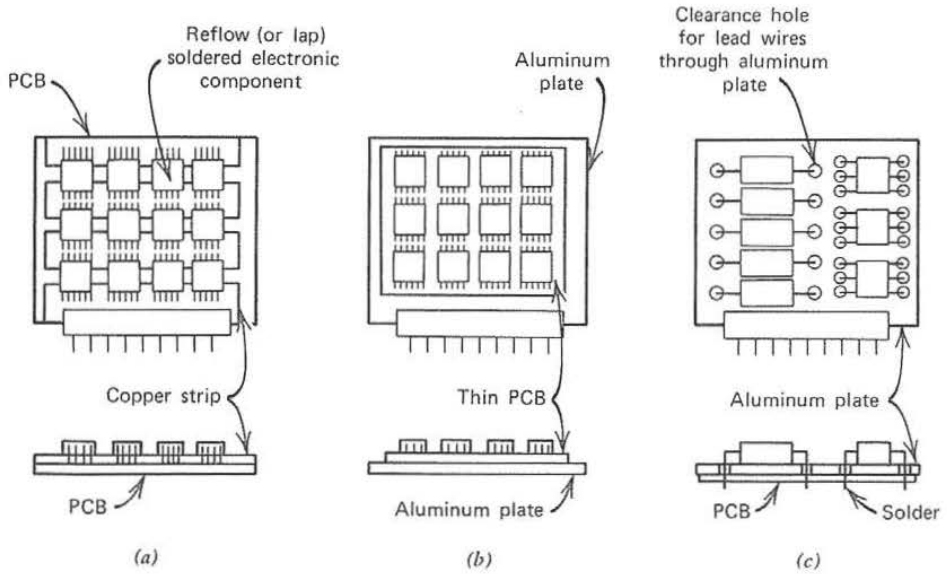


Figure 4.1 Various methods for mounting components on PCBs for conduction cooling. (*a*) Copper strips over PCB and under components; (*b*) aluminum plate bonded to a thin PCB; (*c*) clearance holes in aluminum plate with PCB on back side.

SEC et al. v. MRI
SEC Exhibit 1013.0092
IPR2023-00221

ing these components on PCBs, adequate attention should be given to the heat flow through the wires. For example, if the mounting method shown in Figure 4.1*b* is used, plated throughholes should be located in every pad that has a soldered lead connection. Plated throughholes will reduce the thermal resistance through the PCB, which will reduce the temperature rise from the component to the aluminum heat sink plate. Plated throughholes will also prevent the pads from lifting off the PCB, when the components are soldered to the pads on a lap-soldered joint.

The body of the component with a high heat dissipation (greater than about 4 watts/in^2) should be fastened to the PCB so that there is a low thermal resistance path from the body of the component to the heat sink plate. Many different adhesives can be used, such as RTV cements, double-back tape (Mylar or Kapton), epoxies, or just the adhesive quality of a conformal coating if it is used to protect the PCB. The heat flow from the components to the ultimate heat sink should then follow a predetermined path similar to the one shown in Figure 4.2 [9, 29–31].

Some type of adhesive should be used under high power components mounted on the PCB. If no adhesive is used and only an air gap exists, it is very difficult to control the size of the gap. A small air gap may grow into a large air gap after the PCB has been handled by several people. As the uncontrolled air gap grows, the thermal resistance also grows. This increases the temperature rise across the gap, which increases the component case temperatures.

Tests were run on several different types of carbon composition resistors, which were mounted on heat sinks using different mounting methods. The thermal resistance from the top surface of the resistors to the aluminum heat sink immediately under the resistors is shown in Table 4.1.

Tests were also run on $\frac{1}{4} \times \frac{1}{4}$ in flat pack integrated circuits mounted on a 0.005 in thick PCB which was laminated to an aluminum heat sink using a 0.003 in thick prepreg cement, making a total PCB thickness of 0.008 in. The results are shown in Table 4.2.

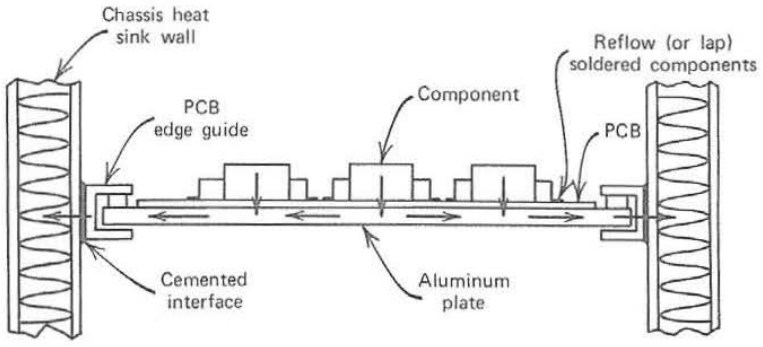


Figure 4.2 Conduction heat flow path from component to heat sink.

SEC et al. v. MRI
SEC Exhibit 1013.0093
IPR2023-00221

Table 4.1 Thermal Resistance for Resistors Mounted on an Aluminum Heat Sink

Component	Interface Cemented with Humiseal Conformal Coating	Resistance with No Cement
¼ watt Resistor RC 07	46.2°C/watt	75°C/watt
½ watt Resistor RC 20	34.2°C/watt	58°C/watt
1 watt Resistor RC 32	19.1°C/watt	26°C/watt

Epoxy adhesives under components should be avoided if extensive repairs are anticipated on the PCB. Components cemented with epoxy adhesives are extremely difficult to remove without damaging the PCB. Other adhesives, such as RTVs, should be used if repairs are required.

Double-sided Mylar tapes work well for holding components to the PCB. However, cleaning solvents normally used to clean the solder flux off of the lamina will dissolve some of the Mylar adhesive and wash it away. At least half of the adhesive can be lost due to cleaning solvents. Pull tests should be run on components that use double-back Mylar tape, to make sure most of the solvent has not been washed away. A minimum pull force of at least 15 psi should be required in the vertical direction (perpendicular to the PCB) for acceptance. This means that a component with a body size of 1 by 1 in should be capable of withstanding a straight vertical pull force of 15 pounds perpendicular to the plane of the PCB.

Table 4.2 Thermal Resistance from Integrated Circuit Case, through 0.008 in Thick Epoxy Fiberglass Lamina, to Aluminum Heat Sink Plate

Component	Resistance with Cement under Component	Resistance with Air Gap under Component
¼ × ¼ in flat pack	20°C/watt with 0.001 in Humiseal conformal coat	—
¼ × ¼ in flat pack	29°C/watt with 0.003 in thick double-sided Mylar tape	60°C/watt with 0.005 in air gap
¼ × ¼ in flat pack	30°C/watt with 0.002 in thick RTV adhesive over 0.004 in thick Stycast epoxy, which insulates copper runs on PCB	35°C/watt with 0.001 in air gap over 0.004 in thick Stycast epoxy, which insulates copper runs on PCB

SEC et al. v. MRI
SEC Exhibit 1013.0094
IPR2023-00221

Extra plated throughholes should be used under the bodies of high power dissipating components mounted on PCBs that are cooled by conduction. The plated throughholes have a small cylindrical ring of copper, which is a good heat conductor. This can decrease the thermal resistance through the PCB lamina as much as 25% in many cases. A lower thermal resistance means a lower component case temperature.

Low power dissipating components may not require any type of adhesive for fastening the components to the PCB. It may be possible to use an air gap under the components if the height of the gap can be controlled to a value less than about 0.005 in. Test data in Table 4.2 show the thermal resistance across an air gap of 0.005 in (including an 0.008 in thick epoxy fiberglass lamina) is 60°C/watt.

When the power dissipation of a $\frac{1}{4} \times \frac{1}{4}$ inch integrated circuit is only 0.10 watt, the temperature rise across the 0.005 in air gap (and the 0.008 in lamina) will only be about 6°C. If this is acceptable, a substantial sum of money may be saved in the fabrication of these PCBs [32].

PCBs with air gaps under the components must not be used for very high altitude or for space applications. There is no heat conduction across an air gap in a vacuum.

4.3 SAMPLE PROBLEM—HOT SPOT TEMPERATURE OF AN INTEGRATED CIRCUIT ON A PLUG-IN PCB

An electronic chassis, with many plug-in types of PCBs, is to be bolted to a liquid cooled cold plate because the system must be capable of operating at an altitude of 100,000 ft (30,480 m). Conduction cooling is the only reliable method of cooling, because natural convection is sharply reduced at high altitudes. In addition, there are other warm electronic boxes in the same equipment bay, which will prevent effective cooling by radiation. The PCBs will be populated with flat pack integrated circuits. These will be uniformly distributed, with a total power dissipation of 8 watts for each board, as shown in Figure 4.3. Components are mounted on both sides of a PCB which has an aluminum heat sink core at the center. The maximum allowable component case temperature is 212°F (100°C). Determine if the proposed design is satisfactory.

SOLUTION

The hot spot case temperature on the flat pack IC can be determined by calculating the temperature rise along individual segments of the heat flow path from the components to the liquid cooled cold plate heat sink. The heat flow path is broken up into five individual segments as follows:

Δt_1 = temperature rise from integrated circuit (IC) case through an 0.008

SEC et al. v. MRI
SEC Exhibit 1013.0095
IPR2023-00221

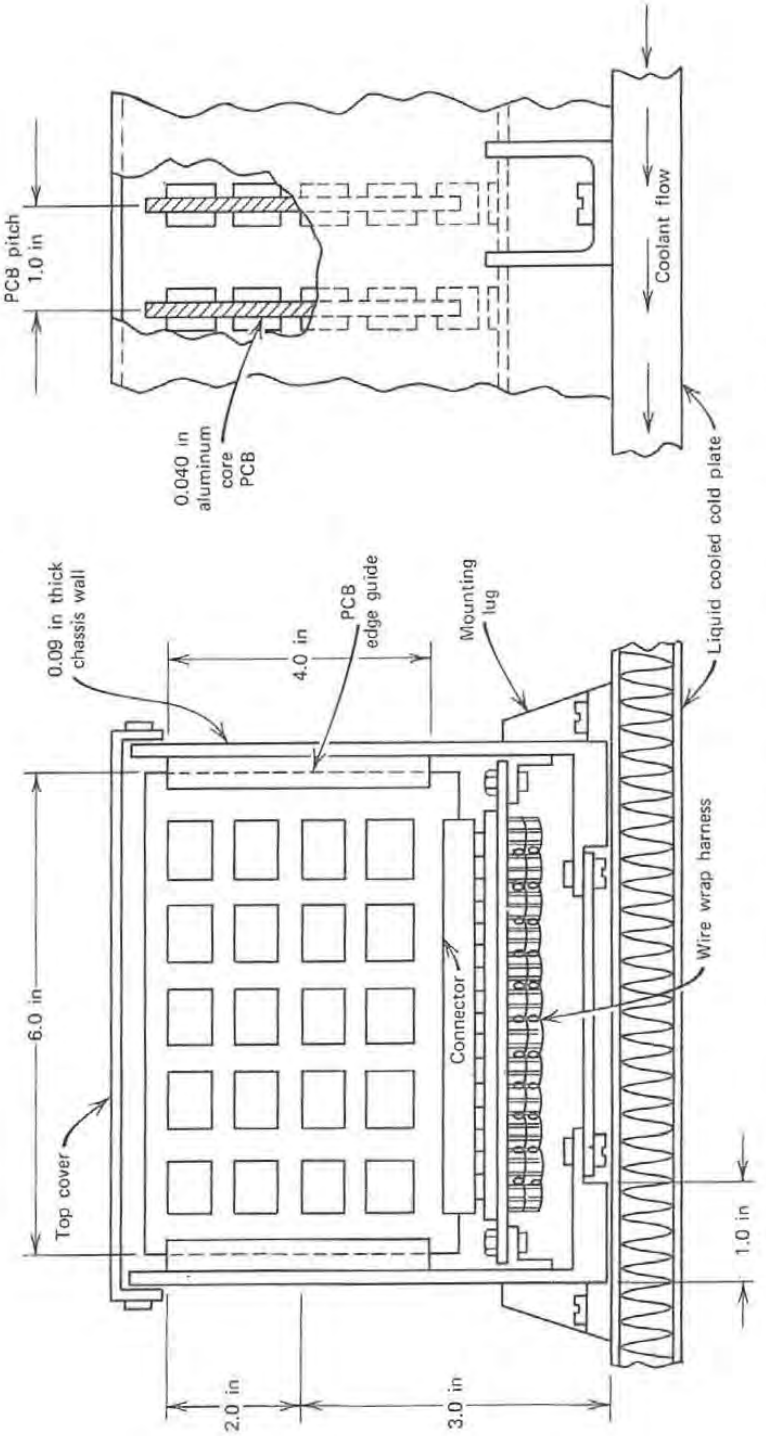


Figure 4.3 Cross section through a chassis with flat pack integrated circuits on PCBs.

SEC et al. v. MRI
SEC Exhibit 1013.0096
IPR2023-00221

in (0.020 cm) thick PCB lamina, plus an air gap of 0.005 in (0.013 cm) under IC case

Δt_2 = temperature rise from center of aluminum heat sink core on circuit board to sides of heat sink by edge guide

Δt_3 = temperature rise across board edge guide from heat sink core to chassis side wall, including cemented interface from guide to wall at 100,000 ft

Δt_4 = temperature rise down along the chassis side wall from the board edge guide to the base of the chassis

Δt_5 = temperature rise across the bolted interface from the base of the chassis to the liquid cooled cold plate at 100,000 ft

The various temperature rises are determined for each of the five segments, using English units and metric units.

All the PCBs in the chassis are very similar and all are on a 1 in pitch. For convenience, therefore, at 1.0 in (2.54 cm) chassis width is analyzed together with one typical PCB.

Δt_1: This represents the temperature rise across the 0.008 in thick PCB lamina to the aluminum core for a single integrated circuit (IC) component that dissipates 0.20 watt. To reduce manufacturing costs, the ICs will *not* be cemented to the PCB. Instead, a 0.005 in air gap will be allowed under the component. This will be controlled by forming the electrical lead wires in a forming die. The ICs will be lap soldered to the printed circuits on the laminated board. The thermal resistance from the component to the aluminum center core, which includes the 0.008 in lamina and the 0.005 in air gap, is obtained from Table 4.2.

$$\Delta t_1 = 60 \frac{^\circ\text{C}}{\text{watt}} \times 0.20 \text{ watt} = 12^\circ\text{C} = 21.6^\circ\text{F} \tag{4.1}$$

Δt_2: This represents the temperature rise along the aluminum core from the center of the board to the sides, for a uniformly distributed heat load of 8.0 watt, on both sides of the PCB. Since the heat load is symmetrical, each half of the heat sink can be considered separately, as shown in Figure 4.4. The following information is required.

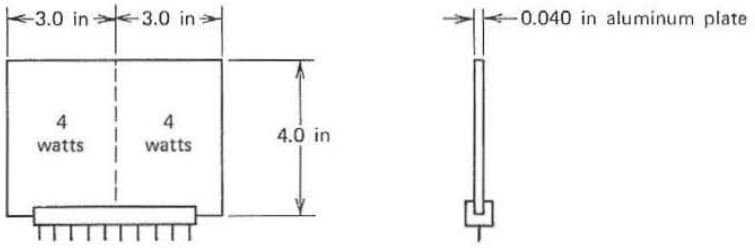


Figure 4.4 Aluminum PCB heat sink core.

SEC et al. v. MRI
SEC Exhibit 1013.0097
IPR2023-00221

Given Q = 4.0 watts = 13.65 Btu/hr = 0.956 cal/sec (heat input on half a board)
L = 3.0 in = 0.25 ft = 7.62 cm (length)
K = 83 Btu/hr ft °F = 0.343 cal/sec cm °C (aluminum conductivity)
A = (4)(0.04) = 0.16 in² = 0.00111 ft² = 1.03 cm² (area)

Substitute into Eq. 3.9 for the Δt from the center of the board heat sink to the side edge, in English units.

$$\Delta t_2 = \frac{(13.65 \text{ Btu/hr})(0.25 \text{ ft})}{(2)(83 \text{ Btu/hr ft °F})(0.00111 \text{ ft}^2)} = 18.5°\text{F} \tag{4.2}$$

In metric units:

$$\Delta t_2 = \frac{(0.956 \text{ cal/sec})(7.62 \text{ cm})}{(2)(0.343 \text{ cal/sec cm °C})(1.03 \text{ cm}^2)} = 10.3°\text{C} \tag{4.2a}$$

Δt_3: This represents the temperature rise across the board edge guide from the aluminum PCB core to the side wall of the chassis, using the Birtcher-type edge guide shown in Figure 3.23*a*.

$$\Delta t_3 = \left(12 \frac{\text{°C in}}{\text{watt}}\right)\left(\frac{4 \text{ watts}}{4 \text{ in}}\right) = 12°\text{C (at sea level)}$$

At an altitude of 100,000 ft this resistance across the edge guide increases about 30%, as shown in Figure 3.24.

$$\Delta t_3 = 1.3(12°\text{C}) = 15.6°\text{C} = 28.1°\text{F} \tag{4.3}$$

Δt_4: This represents the temperature rise along the side of the chassis wall down to the bottom surface of the chassis. The temperature rise can be calculated as (a) a concentrated heat load down the side of the chassis or as (b) a combination of a uniformly distributed heat load and a concentrated heat load down the side of the chassis. Both methods will give the same results. The PCBs are on a 1.0 in (2.54 cm) pitch, so that a 1.0 in section of the chassis is analyzed.

(a) *Concentrated Heat Load.* For the concentrated heat load down the side of the chassis, the length of the heat flow path is taken from the center of the PCB to the center of the bottom flange, as shown in Figure 4.5.

Q = 4 watts = 13.65 Btu/hr = 0.956 cal/sec (heat; ref. Table 1.1)
L = 3.5 in = 0.292 ft = 8.89 cm (length)
K = 83 Btu/hr ft °F = 0.343 cal/sec cm °C (thermal conductivity; ref. Table 3.2)
A = (1.0)(0.09) = 0.090 in² = 0.000625 ft² = 0.581 cm² (area)

SEC et al. v. MRI
SEC Exhibit 1013.0098
IPR2023-00221

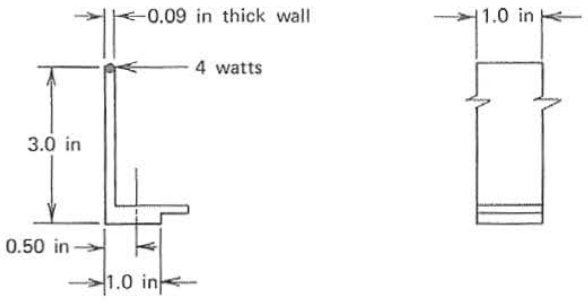


Figure 4.5 Concentrated heat load on side wall of the chassis.

Substitute into Eq. 3.2 for the temperature rise in English units.

$$\Delta t_{4a} = \frac{(13.65)(0.292)}{(83)(0.000625)} = 76.8°\text{F} \tag{4.4}$$

In metric units:

$$\Delta t_{4a} = \frac{(0.956)(8.89)}{(0.343)(0.581)} = 42.6°\text{C} \tag{4.4a}$$

(b) *Uniform Heat Load with Concentrated Heat Load.* For the combination of a uniformly distributed heat load and a concentrated heat load down the side of the chassis, the uniform load acts only on the upper section of the chassis, as shown in Figure 4.6.

Using Eqs. 3.9 for the uniform heat load, and 3.2 for the concentrated

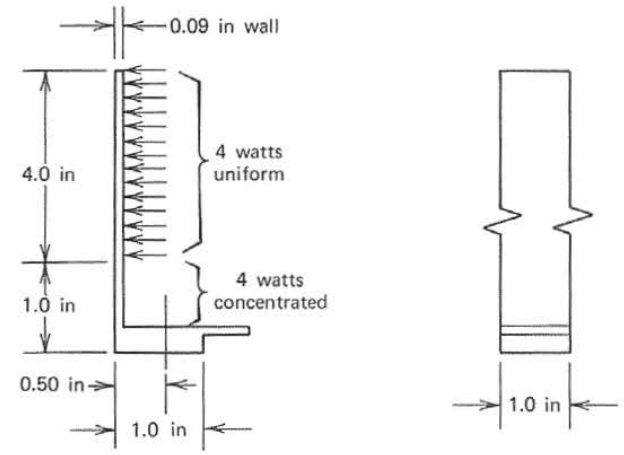


Figure 4.6 Uniform heat load combined with a concentrated heat load.

SEC et al. v. MRI
SEC Exhibit 1013.0099
IPR2023-00221

heat load:

$$\Delta t_{4b} = \frac{(13.65)(4.0/12)}{(2)(83)(0.000625)} + \frac{(13.65)(1.5/12)}{(83)(0.000625)}$$

$$\Delta t_{4b} = 43.9 + 32.9 = 76.8°\text{F} \tag{4.5}$$

The results agree with Eq. 4.4

Δt_5: This represents the temperature rise across the interface from the chassis to the liquid cooled cold plate, for a section of the chassis that is 1 in wide. If the mounting bolts are spaced far apart on sheet metal structures, the average interface pressure will be quite low, so that the average interface conductance will also be quite low. For an air interface at sea level conditions, Table 3.6 shows a value of 200 Btu/hr ft² °F. At an altitude of 100,000 ft, the temperature rise will increase about 30%.

Q = 4 watts = 13.65 Btu/hr = 0.956 cal/sec (heat)
h_i = 200 Btu/hr ft² °F = 0.027 cal/sec cm² °C (interface conductance)
A = (1.0)(1.0) = 1.0 in² = 0.00694 ft² = 6.45 cm² (area)

Substitute into Eq. 3.53 for an altitude of 100,000 ft, in English units.

$$\Delta t_5 = \frac{(1.30)(13.65)}{(200)(0.00694)} = 12.8°\text{F} \tag{4.6}$$

In metric units at the high altitude:

$$\Delta t_5 = \frac{(1.30)(0.956)}{(0.027)(6.45)} = 7.1°\text{C} \tag{4.6a}$$

The hot spot temperature at the mounting interface of the integrated circuits (ICs) at the center of the PCB can be determined from the 80°F (26.6°C) cold plate temperature and the sum of all the individual temperature rises in the heat flow path from the ICs to the cold plate. In English units:

$$t_{IC} = 80 + \Delta t_1 + \Delta t_2 + \Delta t_3 + \Delta t_4 + \Delta t_5 \tag{4.7}$$

$$t_{IC} = 80 + 21.6 + 18.5 + 28.1 + 76.8 + 12.8 = 237.8°\text{F} \tag{4.8}$$

In metric units:

$$t_{IC} = 26.6 + 12 + 10.3 + 15.6 + 42.6 + 7.1 = 114.2°\text{C} \tag{4.8a}$$

For a reliable electronic system, the maximum component surface temperature should not exceed 212°F (100°C). The design is therefore unsatis-

SEC et al. v. MRI
SEC Exhibit 1013.0100
IPR2023-00221

factory, so that some design changes must be made. If the power dissipation cannot be reduced, the mechanical design of the structure must be changed.

An examination of the individual temperature rises shows that Δt_4 has the highest value of 76.8°F (42.6°C), which is shown in Eq. 4.4. The easiest way to reduce the temperature rise is to simply increase the wall thickness. If weight is not too critical, the wall thickness could be increased from 0.090 in to 0.120 in (0.230 to 0.305 cm). This increase would result in a new temperature rise, which is obtained with a direct ratio of the thickness.

$$\Delta t_4' = 76.8\left(\frac{0.090}{0.120}\right) = 57.6°\text{F}\ (32.0°\text{C}) \tag{4.9}$$

This will result in a component surface temperature of 218.6°F (103.6°C), which is still too high, so that another change must be made. If the ICs are cemented to the PCB, Table 4.2 shows that the temperature rise will be reduced but the cost of the assembly will be increased. If a different PCB edge guide is used, the temperature rise can be reduced. For example, if the edge guide shown in Figure 3.23*b* is used, the thermal resistance across the guide will be 8°C in/watt. The new temperature rise across the guide can be obtained with a direct ratio of the resistance, using Eq. 4.3 for an altitude of 100,000 ft.

$$\Delta t_3' = 28.1\left(\frac{8}{12}\right) = 18.7°\text{F}\ (10.4°\text{C}) \tag{4.10}$$

This will now result in a component surface temperature of 209.2°F (98.4°C), which is slightly less than the maximum allowable temperature for the 100,000 ft condition, so that the design is now satisfactory.

4.4 HOW TO MOUNT HIGH POWER COMPONENTS

High power components require special care for mounting, because temperatures can rise very rapidly and failures can occur quickly if the mounting interface conditions change [9]. Stud mounted diodes and transistors, which use insulating washers, must be mounted with care. Some types of insulating washers will deform and cold-flow after temperature cycling tests. This can result in a reduced mounting interface pressure between the component and the heat sink. This loose condition may not be noticed during assembly or inspection if a conformal coating is used, because the coating will tend to hold the parts in place. However, in any vibration or shock environments, the conformal coating may crack and the component will become loose. Without a good heat sink mount, these high power components will fail rapidly.

The insulating washers must be made of materials that will not crack or

SEC et al. v. MRI
SEC Exhibit 1013.0101
IPR2023-00221

cold flow under high interface pressures. They must be capable of withstanding punctures from small sharp foreign particles and must have a high dielectric strength to prevent shorting. The insulating washers should also be easy to manufacture and to install.

Flat and smooth interfaces must be provided for the component and the heat sink mounting surfaces. All parts must be free from burrs and foreign or dirt particles, which would interfere with good heat transfer. The nut on stud mounted devices should be torqued to some predetermined value to ensure a consistent high interface pressure. Some manufacturers have been using Belleville-type spring washers under the nuts to compensate for minor cold flowing in the insulators. These spring washers tend to maintain a high component interface pressure under all conditions. A typical installation is shown in Figure 4.7.

Thermal greases, such as Dow Corning No. 340 and G.E. No. 2850, can increase the interface conductance values as much as 30%. When repairs are made, this thermal grease must be cleaned off and replaced carefully to ensure proper thermal operation. Caution must be exercised with thermal greases when cements or conformal coatings are used. Thermal greases that have a silicone base tend to migrate. In addition, these greases are very difficult to clean off. As a result, they often tend to contaminate clean surfaces, which interferes with the adhesion of cements and conformal coatings.

Junction temperatures must be controlled in high power components to prevent rapid failures. Most component manufacturers will specify the in-

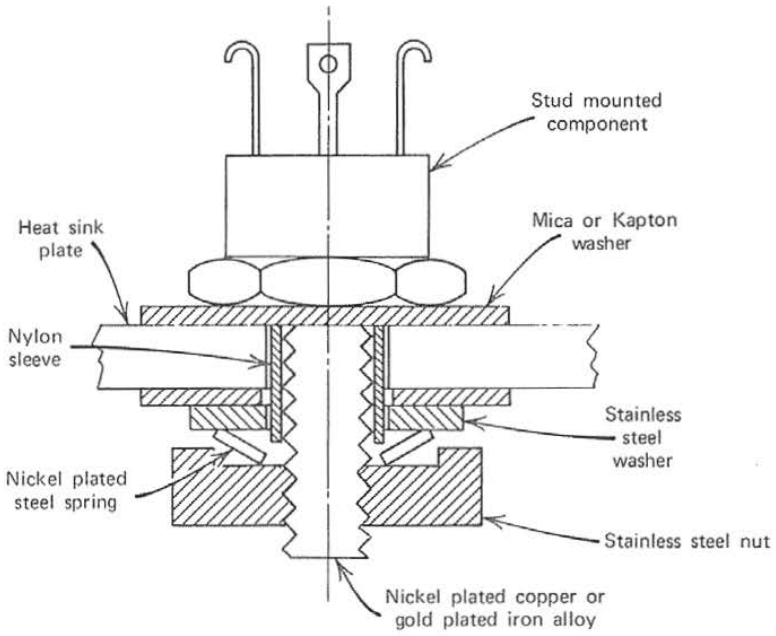


Figure 4.7 Cross section through a stud mounted component.

SEC et al. v. MRI
SEC Exhibit 1013.0102
IPR2023-00221

ternal thermal resistance from the junction of the device to the case using terms such as R_{jc} or Θ_{jc}. For high power components, this value is generally less than 1.0 °C/watt. These manufacturers often rate their components as 175°C or 200°C devices, which means that the junction temperatures are permitted to operate continuously at these values. For high reliable applications, however, the maximum allowable junction temperatures for power devices are usually specified as 125°C. When values this low are hard to achieve for a specific application, it may be possible for the manufacturer to reduce the internal resistance of the device so that the junction temperature will also be reduced for the same power dissipation. If this cannot be done, look for a similar device that has a lower internal resistance before making changes in the component mounting.

It is not always easy to control the junction or chip temperature within a power component. Many interfaces must be controlled to accomplish this. A typical cross section through such a device, showing these internal interfaces, is shown in Figure 4.8 [32].

The mounting position of the high power device within the chassis can often control the hot spot temperature. Sometimes it is relatively easy to reduce the hot spot component temperature just by changing the position of the component. For example, when a high power transistor is mounted at the center of the plug-in PCB, which is cooled by conduction, it will be much hotter than if the same component is mounted at the edge of the same PCB. This is demonstrated in the following sample problem.

4.5 SAMPLE PROBLEM—MOUNTING HIGH POWER TRANSISTORS ON A HEAT SINK PLATE

Several power transistors, which dissipate 5 watts each, are mounted on a power supply circuit board that has a 0.093 in (0.236 cm) thick 5052 alu-

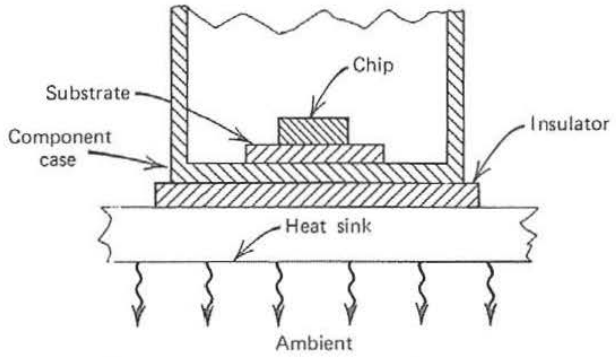


Figure 4.8 Cross section through a power device showing various heat transfer interfaces. The interfaces shown are: chip to substrate; substrate to case; case to insulator; insulator to heat sink; heat sink to ambient.

SEC et al. v. MRI
SEC Exhibit 1013.0103
IPR2023-00221

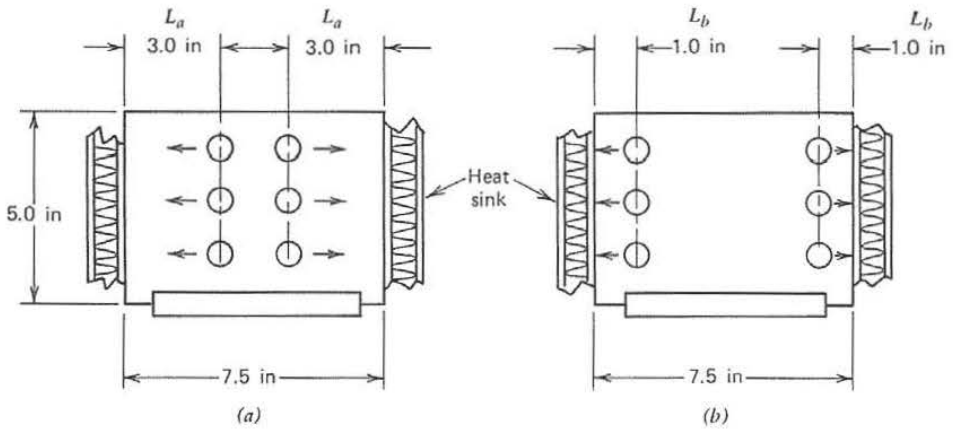


Figure 4.9 Power transistors mounted on an aluminum heat sink plate. (*a*) Old design; (*b*) new design.

minum heat sink plate, as shown in Figure 4.9. Determine how much lower the case temperatures will be when these components are mounted close to the edge of the PCB, as shown in Figure 4.9*b*, instead of the center, as shown in Figure 4.9*a*.

SOLUTION

Since both plug-in PCBs are symmetrical about the center, consider each half of the board for the analysis.

Given $Q = 3 \times 5 = 15$ watts $= 51.2$ Btu/hr $= 3.58$ cal/sec (heat)
$L_a = 3.0$ in $= 0.25$ ft $= 7.62$ cm (length of old design)
$L_b = 1.0$ in $= 0.0833$ ft $= 2.54$ cm (length of new design)
$K = 83$ Btu/hr ft °F $= 0.343$ cal/sec cm °C (conductivity 5052 aluminum)
$A = (5.0)(0.993) = 0.465$ in^2 $= 0.00323$ ft^2 $= 3.0$ cm^2 (area)

Substitute into Eq. 3.2 for the temperature rise at the two different mounting locations, in English units.

$$\Delta t_a = \frac{(51.2 \text{ Btu/hr})(0.25 \text{ ft})}{(83 \text{ Btu/hr ft °F})(0.00323 \text{ ft}^2)} = 47.7°\text{F} \qquad (4.11)$$

In metric units:

$$\Delta t_a = \frac{(3.58 \text{ cal/sec})(7.62 \text{ cm})}{(0.343 \text{ cal/sec cm °C})(3.0 \text{ cm}^2)} = 26.5°\text{C} \qquad (4.11a)$$

SEC et al. v. MRI
SEC Exhibit 1013.0104
IPR2023-00221

For the mounting position near the edge of the PCB, in English units:

$$\Delta t_b = \frac{(51.2)(0.0833)}{(83)(0.00323)} = 15.9°\text{F} \tag{4.12}$$

In metric units:

$$\Delta t_b = \frac{(3.58)(2.54)}{(0.343)(3.00)} = 8.8°\text{C} \tag{4.12a}$$

This shows that moving the transistors closer to the edge of the PCB can reduce the component surface mounting temperature by 47.7 − 15.9 = 31.8°F (17.7°C).

4.6 ELECTRICALLY ISOLATING HIGH POWER COMPONENTS

High power components, such as transistors and diodes, are often mounted on brackets, which are then mounted on heat sinks. When several different components are mounted on the same sink and the component cases are electrically hot, it is obvious that each component must be electrically isolated from the other. This is usually accomplished with the use of electrical isolators under each component, which are relatively good heat conductors. Various type of insulators are available ranging from a soft silicone rubber filled with thermally conductive metal oxides, to rigid but brittle beryllium oxide washers.

The soft type of silicone rubber gasket will deform when the mounting bolts on stud mounted components are tightened. The soft gasket tends to cold-flow into the small voids on the mating surfaces, which eliminates the air pockets on the component and the bracket. The thermal conductivity of the soft silicone gasket is not very high, but its ability to flow into small voids and its ability to deform to fit warped surfaces help to reduce the thermal resistance across the mounting interface.

Beryllium oxide washers provide a high thermal conductivity, which is equal to that of aluminum, while maintaining good electrical insulation. These washers are very brittle and may crack when mounting bolts are tightened if the mounting interfaces are not flat. If the mounting interfaces have small voids, the washer will not deform to fill them. Therefore, for high power applications and for easy replacement of parts, the washer is usually cemented to the heat sink and the component is usually cemented to the washer with an RTV cement, which can be removed easily.

There are conditions where the power requirements are so critical that no temperature gradient can be permitted between the component and the heat sink. Under these circumstances, the component is usually soldered directly to the heat sink so that the interface temperature rise is eliminated.

SEC et al. v. MRI
SEC Exhibit 1013.0105
IPR2023-00221

Hard anodized aluminum is a good electrical insulator and a good heat conductor. Often, components that have electrically hot cases or noncommon grounds may be mounted on the same heat sink plate, with no other insulation except the anodized finish for protection. This is not a good practice because experience has shown that the anodized finish is very hard and brittle and chips easily. This has often resulted in electrical shorts, which have caused electronic systems to malfunction.

Sometimes the thermal design of an electronic system will provide a component mounting surface temperature of 100°C, but the design is still not satisfactory, because the junction temperature is too high. Many component manufacturers make similar power devices which have different internal thermal resistances from the junction to the case (R_{jc}). Because of this, it is possible to have several different transistors or diodes with case temperatures of 100°C and junction temperatures that may range from 125°C to 150°C or higher. When the maximum allowable junction temperature is specified instead of the case temperature and the analysis shows that the resulting case temperature must be well below 100°C to provide the proper junction temperature, try looking for another transistor or diode which has a lower thermal resistance. If this is not available, a mechanical design change must be made to reduce the component case temperature.

It is extremely important to provide a low thermal resistance heat flow path from the high power dissipating components to the heat sink. When several different interfaces are involved, it may be possible to shift the location of the critical interface (the one with the highest temperature rise) so that the overall temperature rise to the component is reduced.

For example, in the sample problem shown in Figure 3.1, the transistor can be mounted directly to the aluminum bracket, without a mica washer. This will sharply reduce the temperature rise across the component interface, but it will make the bracket electrically hot. The bracket must now be electrically isolated from the heat sink. A thin strip of epoxy fiberglass can be used for the insulator. This can be made much larger than the mica washer because the interface from the bracket to the heat sink is much larger, so that the overall resistance is much lower. This results in a much lower temperature rise. This idea is illustrated in the following sample problem.

4.7 SAMPLE PROBLEM—MOUNTING A TRANSISTOR ON A HEAT SINK BRACKET

Figure 3.1 shows a sample problem where a transistor is mounted on an aluminum bracket with a mica washer insulator. Determine the component case temperature of the transistor when the mica washer is removed and the transistor is mounted directly to the aluminum bracket. Since the bracket will now be electrically hot, add a 0.002 in thick epoxy fiberglass insulator

SEC et al. v. MRI
SEC Exhibit 1013.0106
IPR2023-00221

between the bracket and the heat sink. Compare the case temperatures of the transistor with and without the mica washer insulator.

SOLUTION

The heat transfer path from the transistor to the heat sink wall is broken up into three parts, and each part is examined separately, using Eq. 3.2.

Part 1. Temperature Rise across Cemented Epoxy Fiberglass

Given $L = 0.002$ in epoxy fiberglass + 0.002 in (epoxy cement)
$L = 0.004$ in $= 0.000333$ ft $= 0.0101$ cm (thickness)
$Q = 7.5$ watts $= 25.6$ Btu/hr $= 1.79$ cal/sec (heat dissipated)
$K = 0.167$ Btu/hr ft °F (thermal conductivity epoxy PCB + cement)
$A = (1.3)(0.75) = 0.975$ in^2 $= 0.00677$ ft^2 (insulator area)

Substitute into Eq. 3.2 using English units.

$$\Delta t_1 = \frac{(25.6)(0.000333)}{(0.167)(0.00677)} = 7.5°\text{F} \tag{4.13}$$

Part 2. Temperature Rise along Aluminum Bracket

This will be the same as the value shown by Eq. 3.4 because the conditions are exactly the same.

$$\Delta t_2 = 41.1°\text{F} \qquad \text{(ref. Eq. 3.4)}$$

Part 3. Temperature Rise across Transistor Interface without Mica Washer

Table 3.3 is used to determine the temperature rise across the transistor interface, considering a No. 10-32 bolt stud.

$R = 0.60$ °C/watt (dry interface; ref. Table 3.3)
Q = 7.5 watts

$$\Delta t_3 = (7.5 \text{ watts})\left(0.60 \frac{°\text{C}}{\text{watt}}\right) = 4.5°\text{C} = 8.1°\text{F} \tag{4.14}$$

The transistor case temperature is determined by adding the heat sink temperature of 131°F to the three Δt's just computed.

$$t_{\text{trans}} = 131 + 7.5 + 41.1 + 8.1 = 187.7°F \tag{4.15}$$

Comparing Eqs. 4.15 and 3.6 shows there can be a net saving of 35.7°F

SEC et al. v. MRI
SEC Exhibit 1013.0107
IPR2023-00221

by removing the mica washer and reducing the thickness of the epoxy cement while adding an insulator at the bracket interface to the heat sink.

4.8 POTTED MODULES

Electronic components with high heat densities, or components that must operate in severe vibration and shock environments, are often encapsulated (or potted) in various types of foams, epoxies, and silicone compounds. Thermally conductive additives, such as aluminum powder and metal oxides, may also be used to improve the overall thermal conductivity of the potting material.

Aluminum powder additives can increase the thermal conductivity of an epoxy by a factor of 10. However, experience with aluminum powder has shown that shorts can develop under certain circumstances. Tests on potted modules, which are filled with aluminum powder, show that when they are machined with sharp cutters, there will be no shorts between two points on the machined surfaces. However, when a dull tool, such as the tip of a screwdriver, is drawn across the machined surface, there will be electrical continuity between any two points within the groove. Apparently, a sharp machine cutting tool generates a clean cut on the small aluminum particles. A dull tool, on the other hand, smears the soft particles so that they overlap and join one another to form an electrically conductive path, which can cause short circuits.

High power transformers and chokes are often potted in thermally conductive epoxies to improve their heat transfer. These devices have a large amount of iron and copper, which are good heat conductors themselves. However, it is still necessary to provide a low resistance thermal path from the copper and iron to the ultimate heat sink, to prevent overheating in severe operating environments.

Discrete electronic components such as resistors, capacitors, diodes, and transistors are often potted in small modules to improve their vibration and shock resistance or to provide a single compact package that can be handled and mounted like a single component. Heat removal can be a problem, even for packages with relatively low power dissipations, unless proper attention is given to the packaging of these components.

The orientation of the component electrical lead wires within the potted module can sharply influence the heat transfer characteristics. This is because as much as 50% of the heat generated within some components may be conducted through the metal lead wires. In addition, the thermal conductivities of the various cylindrical component bodies on the resistors, capacitors, and diodes are much higher than the thermal conductivity of the plain potting material. For the lowest temperature rise along a given length, the heat flow path should therefore be parallel to the cylindrical axes of the component bodies.

SEC et al. v. MRI
SEC Exhibit 1013.0108
IPR2023-00221

Internal heat sinks can be used effectively in potted modules. When several components have high power dissipations, they can be mounted on insulated aluminum or copper strips before the potting compound is added. The aluminum or copper strips should then be brought close to the heat transfer surface for efficient heat removal. These strips will reduce the internal thermal resistance and reduce local hot spot temperatures.

When transistor heat sinks are required within a potted module, a good heat flow path can often be provided by mounting the transistors upside down, with their hat sections on metal heat sink strips, as shown in Figure 4.10.

When the module is to be mounted to a cold plate, it may be necessary to machine the mounting surface flat to ensure a good thermal interface for efficient heat removal.

Many potting compounds have a high coefficient of thermal expansion, which can cause problems in electronic systems that must be capable of operating over a wide temperature range. Most potting compounds will expand and contract through large displacements over a temperature range from −65°F (−54°C) to +185°F (+85°C). To avoid cracked components and broken solder joints, a thin resilient conformal coating should be applied to the entire module before it is potted.

The effective thermal conductivity of the potted module can be increased by spreading a number of thin copper wires throughout the module. This will increase the heat flow in the areas adjacent to the wires, which can help to reduce hot spot temperatures. This technique is demonstrated in the following sample problem.

4.9 SAMPLE PROBLEM—TEMPERATURE RISE IN A POTTED MODULE

Determine the temperature rise from the mounting surface to the top edge of the components in the potted module, which is shown in Figure 4.11, for

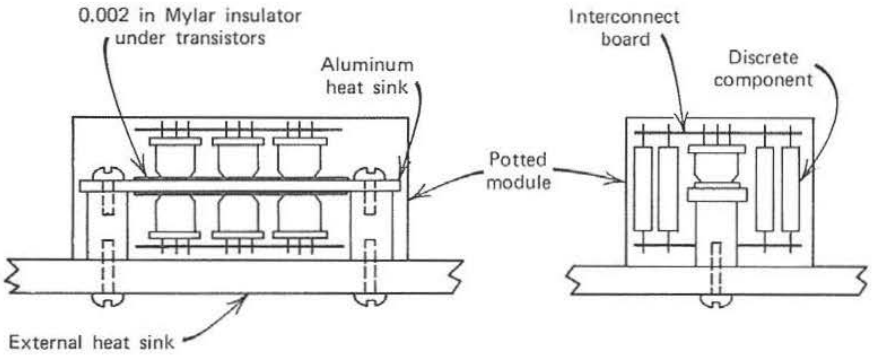


Figure 4.10 Cross section through a potted module showing an internal heat sink.

SEC et al. v. MRI
SEC Exhibit 1013.0109
IPR2023-00221

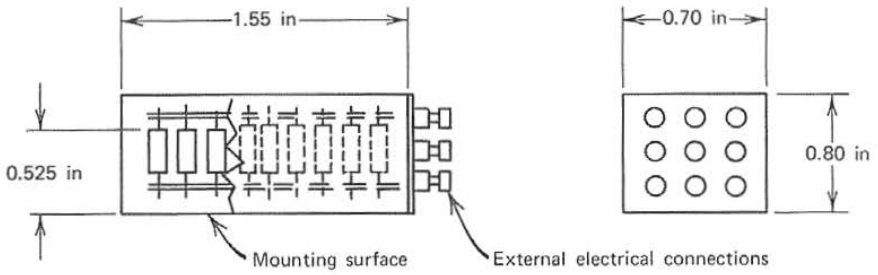


Figure 4.11 Potted module with a uniform power dissipation.

two different conditions:

1 Using an epoxy compound filled with aluminum oxide.
2 Using the compound in (1), but with 12 0.031 in (0.0787 cm) diameter copper wires distributed through the module adjacent to the components.

The power dissipation of the module is 2.0 watts and it is uniformly distributed through the module.

SOLUTION

Condition 1. Epoxy Potting Filled with Aluminum Oxide

The following information is required:

Given Q = 2.0 watts = 6.83 Btu/hr = 0.478 cal/sec (heat) (see Table 1.1 for power conversions)
L_e = 0.525 in = 0.0437 ft = 1.333 cm (epoxy length)
K_e = 1.25 Btu/hr ft °F = 0.00516 cal/sec cm °C (epoxy conductivity; see Table 3.2 for conductivity and Table 1.2 for conversions to English and metric units)
A_e = (1.55)(0.70) = 1.085 in^2 = 0.00753 ft^2 = 7.0 cm^2
A_e = (cross section area along heat flow path)

Substitute into Eq. 3.2 for the temperature rise in epoxy.

$$\Delta t_1 = \frac{(6.83)(0.0437)}{(1.25)(0.00753)} = 31.7°\text{F} \tag{4.16}$$

Condition 2. Epoxy Potting with Copper Wires

A parallel heat flow path exists through the module. One path is through the epoxy, and one path is through the copper wires, as shown in Figure 4.12.

SEC et al. v. MRI
SEC Exhibit 1013.0110
IPR2023-00221

The following information is required:

Given $L_{cu} = 0.525$ in $= 0.0437$ ft $= 1.333$ cm (copper length)
$K_{cu} = 166$ Btu/hr ft °F $= 0.685$ cal/sec cm °C (copper conductivity)
$A_{cu} = (\pi/4)(0.031)^2(12) = 0.00906$ in^2 $= 0.0000629$ ft^2
$A_{cu} = 0.0584$ cm^2 (copper cross section area)

Net area of epoxy without copper wires:

$$A_n = A_e - A_{cu} = 0.00753 - 0.0000629 = 0.00747 \text{ ft}^2$$

Substitute into Eq. 3.14 for the epoxy thermal resistance.

$$R_e = \frac{0.0437 \text{ ft}}{(1.25 \text{ Btu/hr ft °F})(0.00747 \text{ ft}^2)} = 4.68 \frac{\text{hr °F}}{\text{Btu}} \tag{4.17}$$

Substitute into Eq. 3.14 for the copper thermal resistance.

$$R_{cu} = \frac{0.0437}{(166)(0.0000629)} = 4.18 \frac{\text{hr °F}}{\text{Btu}} \tag{4.18}$$

Substitute Eqs. 4.17 and 4.18 into Eq. 3.17 for the combined thermal resistance of the epoxy and copper in condition 2 for the parallel flow path.

$$\frac{1}{R_2} = \frac{1}{4.68} + \frac{1}{4.18}$$

$$R_2 = 2.21 \frac{\text{hr °F}}{\text{Btu}} \tag{4.19}$$

Substitute into Eq. 3.15 to obtain the temperature rise through the module with the potting and the copper wires.

$$\Delta t_2 = QR_2 = \left(6.83 \frac{\text{Btu}}{\text{hr}}\right)\left(2.21 \frac{\text{hr °F}}{\text{Btu}}\right) = 15.1\text{°F} \tag{4.20}$$

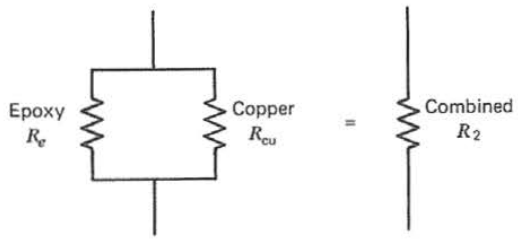


Figure 4.12 Parallel heat flow path through a potted module.

SEC et al. v. MRI
SEC Exhibit 1013.0111
IPR2023-00221

Comparing Eq. 4.16 with Eq. 4.20 shows that adding 12 thin copper wires to the potted module, parallel to the heat flow path, can reduce the average temperature rise about 50%.

4.10 COMPONENT LEAD WIRE STRAIN RELIEF

Electronic components have a wide variety of lead wire sizes, shapes, materials, and arrangements. One thing they all have in common is the potential for failure if the components are mounted without the proper strain relief in the lead wire. The lack of strain relief, or improper strain relief, can cause broken solder joints, lifted printed circuit pads, cracked components, cracked glass headers, short circuits, and many other strain related problems in systems that must operate over a wide temperature range. Special attention must be given to the component mounting techniques for electronic systems that will be exposed to temperatures that range from −65°F(−54°C) to + 185°F (+85°C). This type of environment will generate high strains in elements that have different coefficients of expansion. High strains lead to high creep rates in solder joints, which can produce failures in a very short period of time. Solder joint stresses should be limited to a maximum value of 400 lb/in² (28,148 g/cm²) for the foregoing temperature range to prevent temperature induced strain failures [20, 21, 33–35].

The trend in electronic packages is toward increased component density, so that the size of the lead wire strain relief has been constantly shrinking. A large strain relief means more space per component, which reduces the number of components that can be placed on a circuit board. However, when the strain relief gets too small, the problems begin to multiply rapidly.

Strains and stresses that are developed in the electronics are often the result of expansions and contractions in the structural elements of the system due to temperature changes. Expansion coefficients, which vary substantially for different materials, should be evaluated for their effects on the strain relief, to prevent excessive stresses from developing.

Cylindrical components such as resistors, capacitors, and diodes with axial leads should be provided with a minimum strain relief of about 0.100 in (0.254 cm), as shown in Figure 4.13*a* for lap and dip solder joints. Experience has shown that this type of strain relief will provide sufficient safety over the temperature range from −65°F (−54°C) to +185°F (+85°C) even with aluminum heat sinks.

Large rectangular components, such as transformers and chokes, generally have large wires, often with diameters of about 0.032 in (0.0813 cm). A large strain relief, such as shown in Figure 4.13*b* and *c*, may be required to prevent printed circuit pads from lifting on lap soldered joints.

Components are often placed closer together in high density packages, so that there is less room for a good strain relief in the lead wires. A longer

SEC et al. v. MRI
SEC Exhibit 1013.0112
IPR2023-00221

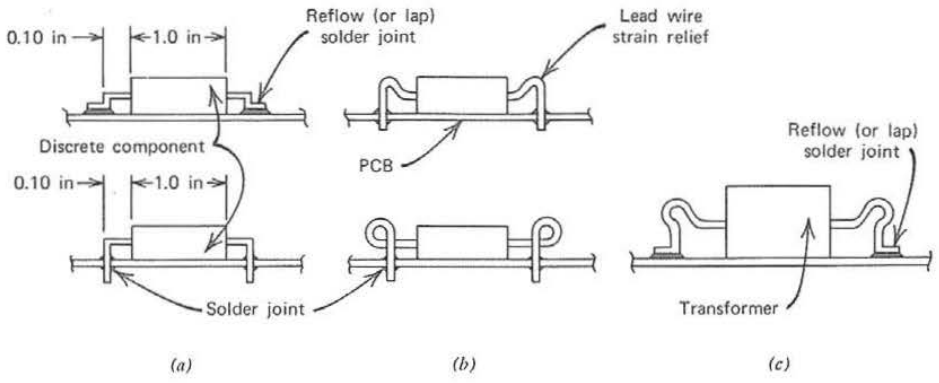


Figure 4.13 Typical lead wire strain relief for electronic components.

strain relief can be obtained by looping the lead wire, as shown in Figure 4.13*b*, to reduce the overall length of the component.

Transistors in TO-5 and TO-18 cans are often mounted on PCBs with the base of the transistor flush with the board, as shown in Figure 4.14*a*. This is a poor practice, because the lack of a strain relief in the lead wires can result in fractured solder joints as the circuit board thickness expands and contracts with temperature changes.

Flush mounted components may also produce trapped gas pockets in the electrical lead wire areas when the component is flow soldered to the PCB.

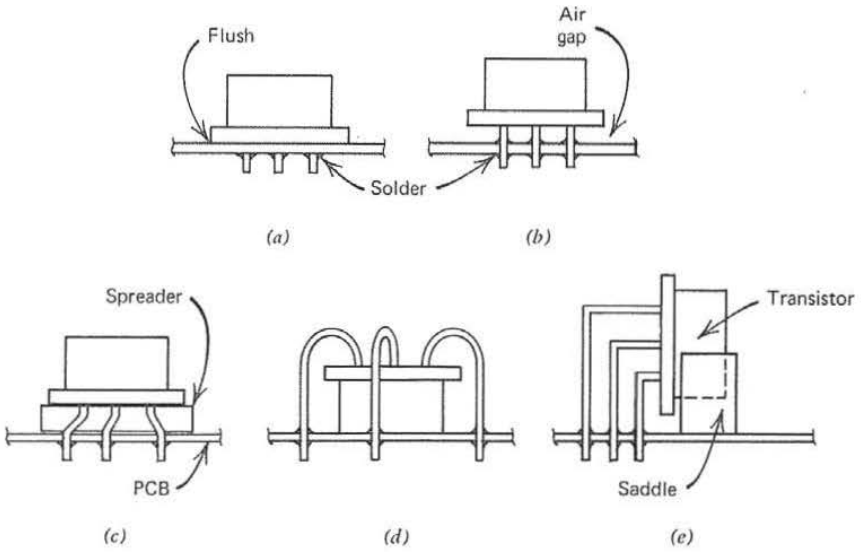


Figure 4.14 Various methods for mounting transistors. (*a*) Straight through (poor); (*b*) air gap (good low power); (*c*) spreader (good); (*d*) waterfall (good); (*e*) side saddle (good).

SEC et al. v. MRI
SEC Exhibit 1013.0113
IPR2023-00221

If the heated air is not permitted to escape from the small area around the lead wire, the increased air pressure will prevent the solder from wicking up the lead wire in the plated throughhole. This will result in a small solder fillet only at the bottom of the PCB. The outer appearance of the solder joint will seem normal, but the joint may have only 25% of the required amount of solder, which will result in a very weak solder joint that can fail very rapidly. Several techniques for mounting transistors are shown in Figure 4.14.

DIPs must also be carefully mounted, because they have very stiff lead wires with almost no strain relief. DIPs are made to be plugged into circuit boards, using the lead wires as electrical connectors, so that the lead wires must be rigid. When the dip leads are soldered to the PCB, this rigidity may produce solder joint failures when the body of the dip is mounted flush against the PCB. During temperature cycling, the expansion of the DIP body and the expansion of the epoxy fiberglass circuit board may be sufficient to produce an excessive strain in the solder joint, producing failures after several dozen cycles over a wide temperature range.

For conduction cooled systems, the body of the DIP should be cemented to the heat sink with a thermally conductive material that is resilient and will yield under the loads developed by the temperature cycle. RTV cements work well for applications of this type if the cement can be controlled to a thickness of about 0.005 to 0.010 in (0.0127 to 0.0254 cm). When the thickness of the cement exceeds 0.010 in, the thermal resistance from the component to the PCB may be too high, resulting in a high junction temperature. When the cement thickness is less than about 0.001 in (0.00254 cm), there will not be enough room for expansion at high temperatures, and the solder joints may again experience high stresses, resulting in joint failures after several dozen cycles.

DIPs have shoulders in their lead wires, which normally keeps the body of the DIP raised above the PCB. When metal strips are used under the component body for conduction cooled systems, the metal strip should be thick enough to bridge the resulting gap under the component body on high power dissipating DIPs. This should include any cements or tapes for holding the DIP to the heat sink.

When the power dissipation on a DIP is relatively low, below about 0.200 watt, it may not be necessary to cement the DIP to the heat sink in conductively cooled systems. Instead, a controlled air gap may be used under the DIP body. Air has a relatively low resistance for a small air gap. If the gap can be controlled to about 0.010 in (0.0254 cm), it may not result in extremely high junction temperatures. Some typical mounting installations for DIPs are shown in Figure 4.15.

The approximate temperature rise across the 0.010 in air gap for a power dissipation of 0.200 watt can be determined by considering only the heat conducted from the body of the DIP and ignoring the heat that is conducted

SEC et al. v. MRI
SEC Exhibit 1013.0114
IPR2023-00221

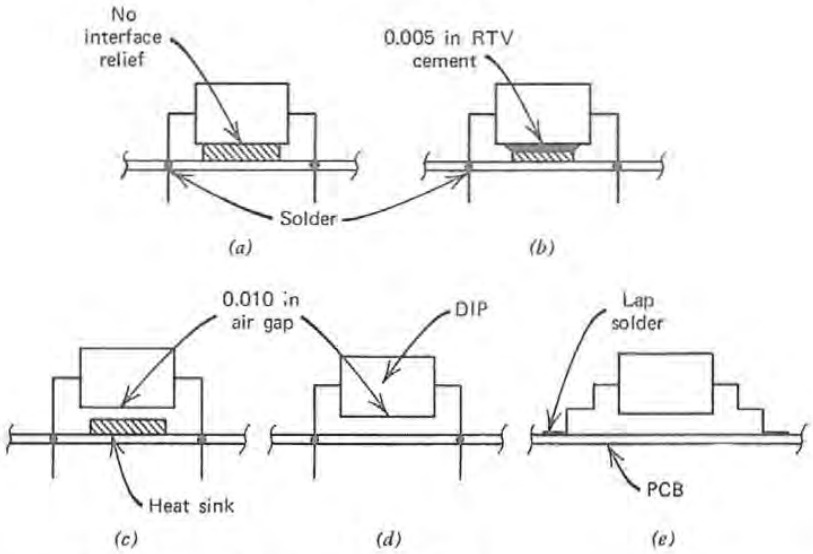


Figure 4.15 Various types of strain reliefs for DIPs mounted on PCBs. (*a*) No strain relief (poor); (*b*) RTV strain relief (good); (*c*) air gap strain relief (good low power); (*d*) air gap strain relief no sink (good low power); (*e*) air gap strain relief no sink (good low power).

through the lead wires. The following information is required:

$Q = 0.200$ watt $= 0.683$ Btu/hr (power dissipation)
$L = 0.010$ in $= 0.000833$ ft (length of air gap)
$K = 0.018$ Btu/hr ft °F (air conductivity at 200 °F; ref. Figure 6.28)
$A = (0.30)(0.70) = 0.21$ in^2 $= 0.00146$ ft^2 (area of DIP)

Substitute into Eq. 3.2 for the temperature rise across the DIP mounting interface.

$$\Delta t = \frac{(0.683)(0.000833)}{(0.018)(0.00146)} = 21.6°\text{F} \tag{4.21}$$

If the air gap is doubled to 0.020 in (0.0508 cm), the temperature rise across the mounting interface will also double, to 43.2°F (24°C).

One of the problems with an air gap is that it is difficult to control. The air gap under a DIP can change when the PCB warps as the system heats. Also, rough handling may sharply change the size of the air gap, which would affect the heat transfer characteristics.

The thermal resistance of a small air gap will not change much, even at altitudes as high as 75,000 ft. Therefore, an air gap can still be used to cool low power dissipating DIPs at these altitudes. Air gaps will not provide

SEC et al. v. MRI
SEC Exhibit 1013.0115
IPR2023-00221

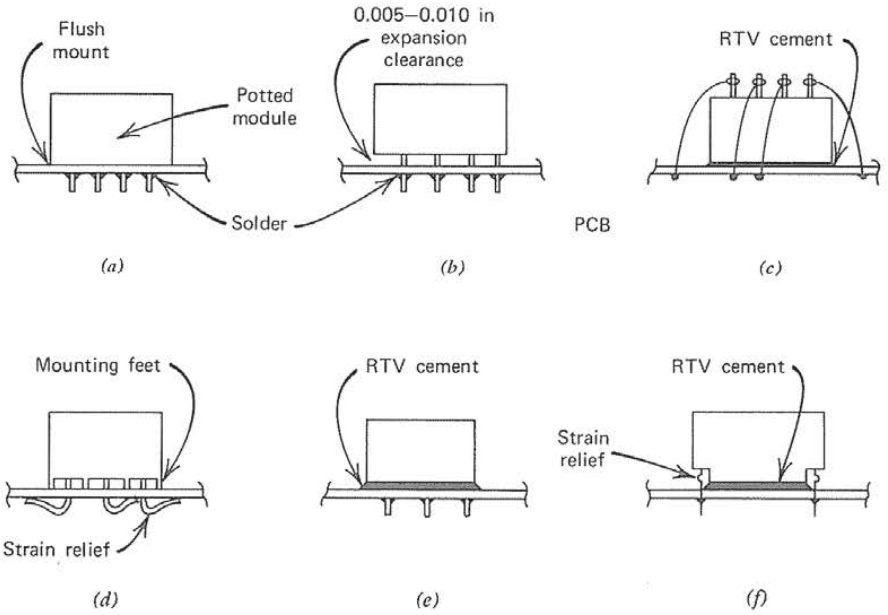


Figure 4.16 Various methods for mounting potted modules to PCBs. (*a*) No strain relief, no outgassing vents (poor); (*b*) air gap strain relief (good); (*c*) waterfall strain relief (good); (*d*) strain relief with outgassing provisions (good); (*e*) strain relief but no outgassing provisions (poor); (*f*) strain relief with outgassing provisions (good).

cooling in the hard vacuum environment of outer space, where there are no gases present to conduct away the heat.

Potted modules are often made with the electrical lead wires extending from one face of the unit. This face may then be mounted flush to a PCB and the lead wires will extend through plated through holes where they are soldered. If no strain relief is provided in the wires, the relatively high coefficients of expansion in the PCB and the potting may produce solder joint failures after a few dozen temperature cycles.

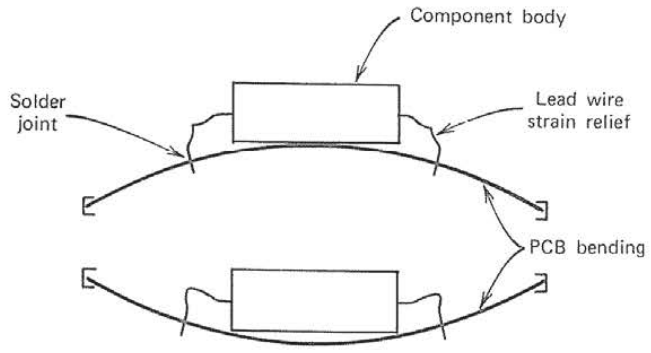


Figure 4.17 Circuit board flexing in a vibration environment.

SEC et al. v. MRI
SEC Exhibit 1013.0116
IPR2023-00221

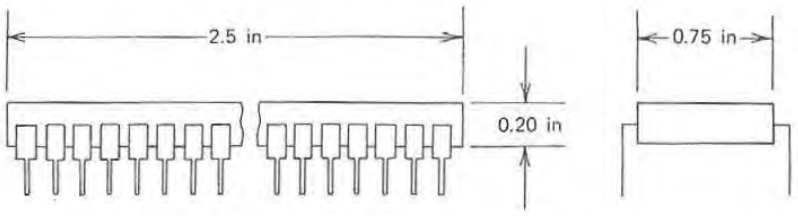


Figure 4.18 Typical microprocessor package.

Some type of strain relief should always be provided in the mounting geometry, to allow a small amount of relative motion to occur, to reduce the loads and strains developed in the solder joints. The maximum allowable thermal stresses in these solder joints should be limited to a value of about 400 psi under the most extreme temperature condition, to ensure a safe design. Some typical mounting configurations for potted modules are shown in Figure 4.16.

Conformal coatings are often used to protect the circuits and components from dirt and moisture. These coatings must not be permitted to fill in or to bridge the strain relief in the electrical lead wires or to fill in the strain

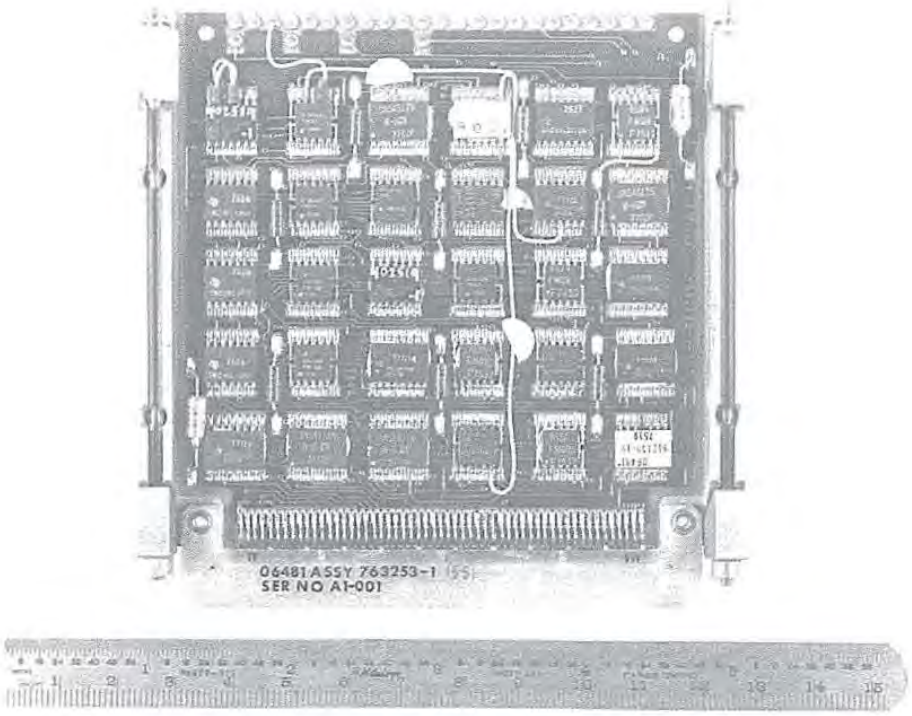


Figure 4.19 Composite aluminum PCB with side wedge clamps. (Courtesy Litton Systems, Inc.)

SEC et al. v. MRI
SEC Exhibit 1013.0117
IPR2023-00221

relief gap under large flat components that have leads that extend from the bottom surface. The bridging will increase the solder joint stresses because of the high coefficient of expansion of most conformal coatings, especially at the high tempertatures.

Circuit boards that must operate in severe vibration and shock environments should have electronic components mounted with an adequate lead wire strain relief. The type of lead wire and solder joint stresses developed in a vibration and shock environment are substantially different from the stresses developed in a thermal cycling environment. The number of stress reversals developed in a vibration environment can run into several hundred million. Therefore, stress concentrations such as nicks, scratches, and small bend radii become important and must be avoided, if possible [1].

Dynamic stresses in the lead wires are developed as the circuit board is forced to bend back and forth in a vibration environment. The condition is

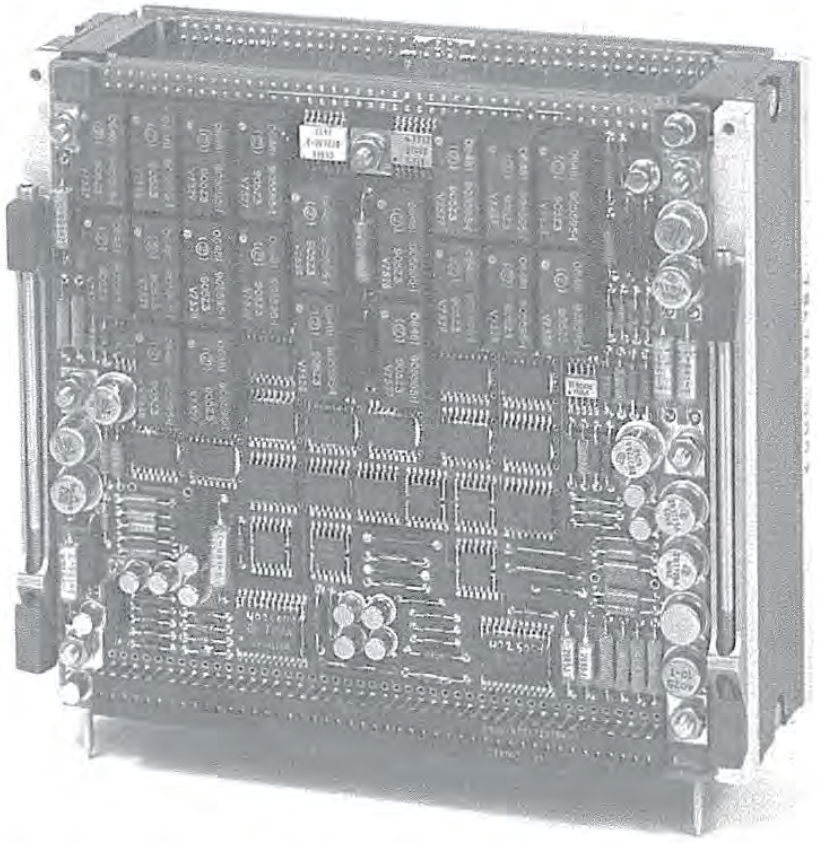


Figure 4.20 Plug-in memory module designed for conduction cooling, uses side wedge clamps that provide high pressures at the thermal mounting interfaces. (Courtesy Litton Systems, Inc.)

SEC et al. v. MRI
SEC Exhibit 1013.0118
IPR2023-00221

most severe during the PCB resonance, when the displacement amplitudes can become quite large. As the PCB bends back and forth, it forces the electrical lead wires on the components to bend back and forth, as shown in Figure 4.17. This can result in lead wire and solder joint failures when the stress levels and the number of fatigue cycles are high enough [1].

Microprocessors are particularly susceptible to solder joint and lead wire vibration faiiures. Microprocessors are manufactured in the form of a large DIP package. Typical dimensions are generally about 2.5 × 0.75 × 0.20 in (6.35 × 1.905 × 0.508 cm), as shown in Figure 4.18. The long component body length will result in an increase in the solder joint and lead wire stresses during vibration when the component is mounted at the center of the PCB. A high resonant frequency is required to keep the dynamic displacements low during resonant conditions, to provide a long fatigue life for the lead wires and solder joints. See reference [1] for more information on mounting electronic components that must operate in severe vibration environments.

Figure 4.19 shows an aluminum core composite circuit board with lap soldered electronic components.

Figure 4.20 shows a memory module made up of several circuit boards bolted together.

SEC et al. v. MRI
SEC Exhibit 1013.0119
IPR2023-00221

5

Practical Guides for Natural Convection and Radiation Cooling

5.1 HOW NATURAL CONVECTION IS DEVELOPED

Heat transfer by natural convection will occur when there is a change in the density of the fluid, which can be a gas or a liquid. Fluids tend to expand as they are heated, which results in a reduced density. In a gravity field, the fluid, which has a lower density, is lighter and therefore rises, creating a movement in the fluid which is called convection. This movement or convection permits the fluid to pick up heat and carry it away. When the fluid movement is produced only by a difference in the fluid density, the process is called natural or free convection. When the fluid movement is mechanically induced by means of pumps or fans, the process is called forced convection or forced circulation.

The driving force for natural convection is not very great because it depends upon the density change in the fluid. Therefore, any small obstacle or resistance in the flow path will sharply reduce the fluid flow rate, and therefore the cooling rate. Heat transfer by natural convection is more efficient if a well-defined, clear, unobstructed flow path is made available. Otherwise, the amount of heat carried away may be much less than anticipated and the resulting temperatures may be much higher than expected.

Natural convection heat transfer is extremely easy to use because no auxiliary equipment is required. It is also very reliable, because there are no auxiliary pumps or fans to malfunction or wear out.

The basic relation for free convection in a fluid (gas or liquid) is shown by three dimensionless ratios involving the Nusselt number, the Grashof

SEC et al. v. MRI
SEC Exhibit 1013.0120
IPR2023-00221

number, and the Prandtl number, as shown in Eq. 5.1 [15, 17, 24]:

$$\underset{\text{Nusselt number}}{\frac{hL}{K}} = C\underbrace{\left(\frac{L^3\rho^2 g\beta\,\Delta t}{\mu^2}\right)^m}_{\text{Grashof number}} \underbrace{\left(\frac{C_p\mu}{K}\right)^n}_{\text{Prandtl number}} \tag{5.1}$$

where C = constant based upon geometry of surface
L = length along heat flow path (ft)
ρ = density of fluid (gas or liquid) (lb/ft^3)
g = acceleration of gravity = 4.17×10^8 (ft/hr^2)
β = volumetric expansion = 1/°R (or bulk modulus)
Δt = temperature difference (°F)
μ = viscosity (lb/ft hr)
C_p = specific heat of fluid (Btu/lb °F)
K = thermal conductivity (Btu/hr ft °F)
h = convection coefficient (Btu/hr ft^2 °F)

Experiments have shown that the exponents m and n are very nearly equal. Free convection calculations for electronic equipment can be made using an exponent value of 0.25.

Several items in Eq. 5.1 can be combined to simplify the convection relation as follows. Let

$$a = g\beta\rho^2\frac{C_p\mu}{K}$$

Then

$$h = C\frac{K}{L}(aL^3\,\Delta t)^{0.25} \tag{5.2}$$

The relation $Ka^{0.25}$ is fairly constant over a wide temperature range from 0°F to 200°F, with a value of about 0.52. Substituting this value into Eq. 5.2 results in the simplified natural convection equation for the laminar flow range [7, 8].

$$h = 0.52C\left(\frac{\Delta t}{L}\right)^{0.25} \tag{5.3}$$

Some typical values for C, which have been determined by experiments in free air, are shown in Table 5.1 [7, 8].

The characteristic length (L) is determined by the flow path the cooling air takes as it passes over the heated surface. Some typical values are shown in Table 5.2 [3, 7, 8].

The characteristic dimension for a flat horizontal rectangular plate is

SEC et al. v. MRI
SEC Exhibit 1013.0121
IPR2023-00221

Table 5.1 Values for Constants Based upon the Surface Geometry

Shape and Position	Value of C
Vertical plates	0.56
Long vertical cylinders and pipes	0.55
Horizontal cylinders and pipes	0.52
Horizontal plates facing upward	0.52
Horizontal plates facing downward	0.26
Spheres (where L = radius)	0.63
Small parts, also wires, (L = diameter)	1.45
Components on circuit boards	0.96
Small components in free air	1.39
Components in cordwood modules	0.48

obtained by using the geometric mean of the length and width. Considering a rectangular plate 12 × 6 in, Table 5.2 shows that the characteristic length would be as follows:

$$L = \frac{2(12)\,(6)}{12 + 6} = \frac{144}{18} = 8.0 \text{ in} = 0.606 \text{ ft}$$

For square plates, the characteristic dimension is the length of one side.

Table 5.2 Characteristic Length for Various Surfaces

Surface	Position	Characteristic Length (ft)
Flat plane	Vertical	Height, limited to 2 ft[a]
Flat plane (nonrectangular)	Vertical	$\frac{\text{Area}}{\text{Horizontal width}}$
Flat plane (circular)	Vertical	$\frac{\pi}{4} \times$ diameter
Cylinder	Vertical	Height, limited to 2 ft
Flat plane	Horizontal	$\frac{2(\text{Length} \times \text{width})}{\text{Length} + \text{width}}$
Cylinder	Horizontal	Diameter
Sphere	Any	Radius

[a] *When the actual plate height* L *exceeds 2 ft, use a maximum value of 2 ft in Eq. 5.3.*

SEC et al. v. MRI
SEC Exhibit 1013.0122
IPR2023-00221

For a 9.0 in square plate, Table 5.2 gives the correct value as follows:

$$L = \frac{2(9)\ (9)}{9 + 9} = \frac{162}{18} = 9.0 \text{ in} = 0.75 \text{ ft}$$

5.2 NATURAL CONVECTION FOR FLAT VERTICAL PLATES

The simplified laminar flow natural convection equation for a heated flat vertical plate at ordinary temperatures and sea level conditions is obtained with Eq. 5.3 and Table 5.1, as shown in Eq. 5.4. The temperature difference Δt must be in °F and the vertical height L must be in feet.

$$h_c = 0.29 \left(\frac{\Delta t}{L} \right)^{0.25} \tag{5.4}$$

The average natural convection coefficient h_c defines the thermal characteristics of the air film, which clings to the plate and restricts the flow of heat from the plate to the surrounding ambient air. A cross section through the plate and air film is shown in Figure 5.1.

Laminar flow occurs along the lower edges of a tall plate. The transition from laminar to turbulent flow generally starts at a distance of about 2 ft from the bottom edge of the plate. Turbulent flow occurs at distances greater than approximately 2 ft, depending upon the temperature difference from the plate to the ambient.

5.3 NATURAL CONVECTION FOR FLAT HORIZONTAL PLATES

The simplified laminar flow natural convection equation for a heated horizontal flat plate facing upward or a cooled plate facing downward is obtained

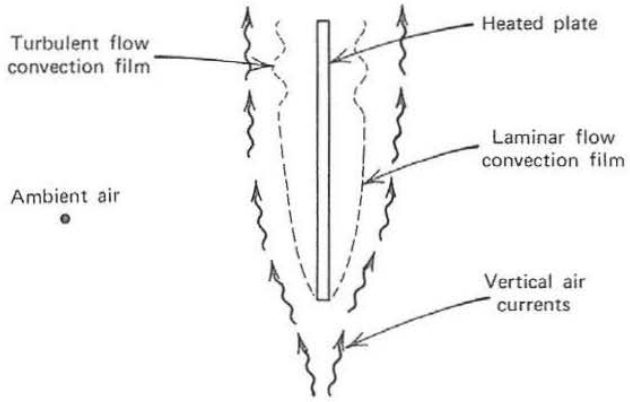


Figure 5.1 Cross section through a vertical heated plate.

SEC et al. v. MRI
SEC Exhibit 1013.0123
IPR2023-00221

from Eq. 5.3 and Table 5.1, as shown in Eq. 5.5. The temperature difference Δt must be in °F and the L dimension must be in feet.

$$h_c = 0.27 \left(\frac{\Delta t}{L} \right)^{0.25} \tag{5.5}$$

Cooling air flows in along the sides around the perimeter of the horizontal plate to replace the heated air, which rises as shown in Figure 5.2.

When the heated horizontal plate faces downward, the cooling air must flow along the bottom surface to the outer edges of the plate before it can rise, as shown in Figure 5.3. This increases the flow resistance, which reduces the cooling effectiveness.

The convection coefficient for a heated plate facing downward or a cooled plate facing upward is shown in Eq. 5.6. Its value is about half that of a heated plate facing upward.

$$h_c = 0.13 \left(\frac{\Delta t}{L} \right)^{0.25} \tag{5.6}$$

5.4 HEAT TRANSFERRED BY NATURAL CONVECTION

Once the convection coefficient h_c has been established, the amount of heat that can be carried away is determined from Eq. 5.7 [3, 7, 8].

$$Q = h_c A \, \Delta t \tag{5.7}$$

The units are the same as the relations shown in Eq. 3.53 except that the A represents the area of the heated surface, Δt represents the temperature difference between the heated surface and the surrounding ambient air, and h_c represents the convection coefficient.

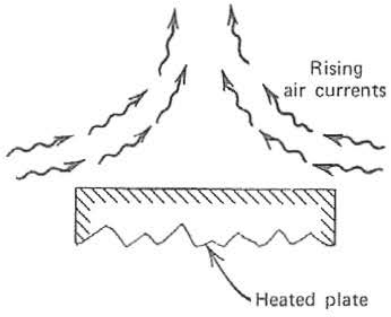


Figure 5.2 Heated horizontal surface facing upward.

SEC et al. v. MRI
SEC Exhibit 1013.0124
IPR2023-00221

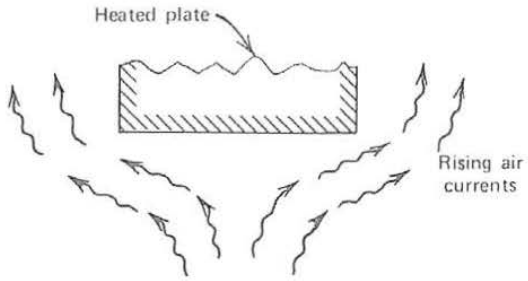


Figure 5.3 Heated horizontal surface facing downward.

5.5 SAMPLE PROBLEM—VERTICAL PLATE NATURAL CONVECTION

Determine the amount of heat that can be carried away from the vertical plate shown in Figure 5.4, considering only natural convection.

SOLUTION

Assuming laminar flow conditions (it is not known at this point if the flow is laminar or turbulent), the natural convection coefficient is determined from Eq. 5.4.

$$h_c = 0.29\left[\frac{200 - 80\ ^\circ\text{F}}{4\ \text{in}/(12\ \text{in/ft})}\right]^{0.25} = 1.26\ \frac{\text{Btu}}{\text{hr ft}^2\ ^\circ\text{F}}$$

$$A = \frac{(8.0)(4.0)\ \text{in}^2}{144\ \text{in}^2/\text{ft}^2}\ (\text{two surfaces}) = 0.444\ \text{ft}^2$$

$$\Delta t = 200 - 80 = 120^\circ\text{F}$$

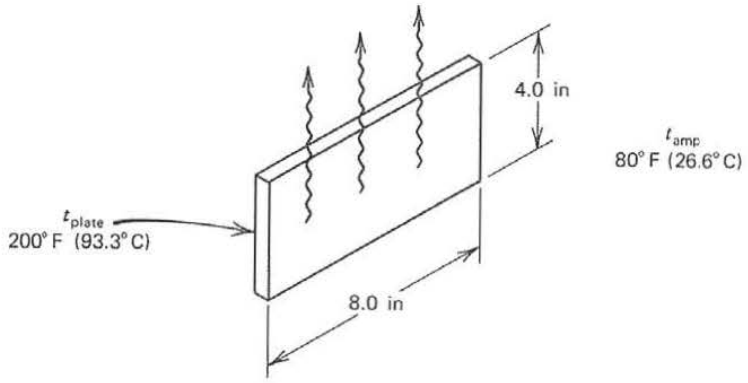


Figure 5.4 Vertical heated aluminum plate.

SEC et al. v. MRI
SEC Exhibit 1013.0125
IPR2023-00221

Substitute into Eq. 5.7 for the amount of heat convected away, using English units.

$$Q = \left(1.26 \frac{\text{Btu}}{\text{hr ft}^2\,^\circ\text{F}}\right)(0.444\ \text{ft}^2)\,(120^\circ\text{F}) = 67.1 \frac{\text{Btu}}{\text{hr}} \tag{5.8}$$

Since 1 watt = 3.413 Btu/hr,

$$Q = \frac{67.1}{3.413} = 19.6 \text{ watts}$$

Tables 1.3 and 1.10 can be used to convert from English units to metric units.

$$h_c = \left(1.26 \frac{\text{Btu}}{\text{hr ft}^2\,^\circ\text{F}}\right)(0.000135) = 0.000170 \frac{\text{cal}}{\text{sec cm}^2\,^\circ\text{C}}$$

$$A = (0.444\ \text{ft}^2)\,(929) = 412.5\ \text{cm}^2$$

$$Q = \left(0.000170 \frac{\text{cal}}{\text{sec cm}^2\,^\circ\text{C}}\right)(412.5\ \text{cm}^2)\,(66.7^\circ\text{C}) = 4.68 \frac{\text{cal}}{\text{sec}} \tag{5.9}$$

Since 1 watt = 0.239 cal/sec,

$$Q = \frac{4.68}{0.239} = 19.6 \text{ watts}$$

The amount of heat lost by natural convection will be different if the plate is rotated 90°, as shown in Figure 5.5. The vertical height L is now 8 in

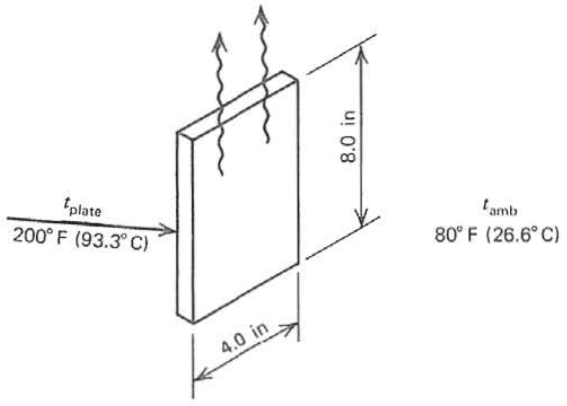


Figure 5.5 Vertical heated plate rotated 90°.

SEC et al. v. MRI
SEC Exhibit 1013.0126
IPR2023-00221

instead of 4 in. Using Eq. 5.4 once again, we obtain

$$h_c = 0.29\left(\frac{120}{8/12}\right)^{0.25} = 1.06\,\frac{\text{Btu}}{\text{hr ft}^2\,^\circ\text{F}}$$

Using Eq. 5.7 with the same plate area and temperature difference yields

$$Q = (1.06)\,(0.444)\,(120) = 56.5\,\frac{\text{Btu}}{\text{hr}} \tag{5.10}$$

Since 1 watt = 3.413 Btu/hr,

$$Q = 16.5 \text{ watts}$$

When the plate is rotated 90°, the amount of heat lost by natural convection is reduced about 15.8% because of the increased plate height, which reduces the natural convection coefficient.

5.6 TURBULENT FLOW WITH NATURAL CONVECTION

Turbulent flow conditions can develop during natural convection conditions when the temperature difference between the heated surface and the surrounding ambient air is high enough. The convection coefficient for turbulent conditions can be approximated by considering only the temperature difference. For heated vertical plates, the natural convection coefficient for turbulent flow is shown by Eq. 5.11. For heated horizontal plates facing upward, the natural convection coefficient for turbulent flow is shown by Eq. 5.12. The temperature difference must be in °F for these conditions [15, 17, 24, 36].

$$\text{Vertical plates:} \quad h_c = 0.19\Delta t^{0.333} \tag{5.11}$$

$$\text{Horizontal plates:} \quad h_c = 0.22\Delta t^{0.333} \tag{5.12}$$

The temperature difference required to change from laminar flow to turbulent flow for a vertical plate 1 ft high can be determined by setting Eq. 5.4 equal to Eq. 5.11.

$$0.29\left(\frac{\Delta t}{1.0}\right)^{0.25} = 0.19(\Delta t)^{0.333}$$

$$\frac{(\Delta t)^{0.333}}{(\Delta t)^{0.25}} = 1.526$$

$$(\Delta t)^{0.083} = 1.526$$

$$\Delta t = 163^\circ F \tag{5.13}$$

SEC et al. v. MRI
SEC Exhibit 1013.0127
IPR2023-00221

This means that convection heat transfer for a vertical plate 1 ft high will change from laminar flow to turbulent flow when the temperature difference between the plate and the ambient is about 163°F. If the plate is less than 1 ft high, the temperature difference must be greater than 163°F to produce turbulent flow conditions.

The external surfaces of electronic boxes are often available to cool the equipment using natural convection. The heat dissipated by electronic components within the box may be conducted to the outside surfaces, where it is removed by convection. The amount of heat that can be removed is sharply influenced by the temperature difference between the surface of the box and the ambient air and also the surface area of the box.

5.7 SAMPLE PROBLEM—HEAT LOST FROM AN ELECTRONIC BOX

Determine the amount of heat that is lost by natural convection from the external surfaces of the box shown in Figure 5.6. The surface temperature is relatively uniform at 130°F and the ambient temperature is 80°F at sea level conditions. The box is aluminum with an iridited (or chromate) finish that has a low emissivity, so that heat lost by radiation is small and is ignored here.

SOLUTION

The natural convection coefficients for the vertical surfaces is obtained from Eq. 5.4 and Table 5.2.

$$h_c = 0.29\left(\frac{\Delta t}{L}\right)^{0.25}$$

Given $\Delta t = 130 - 80 = 50°\text{F}$

$L = 6.0 \text{ in} = 0.50 \text{ ft}$

$$h_c = 0.29\left(\frac{50}{0.50}\right)^{0.25} = 0.92 \frac{\text{Btu}}{\text{hr ft}^2\,°\text{F}} \tag{5.14}$$

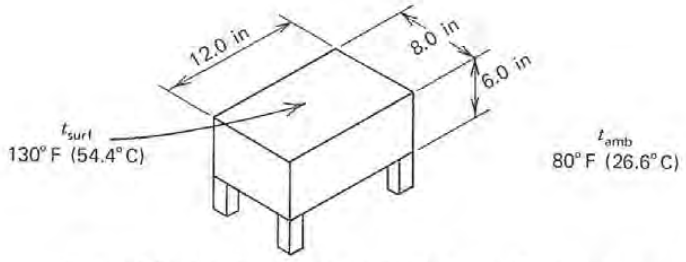


Figure 5.6 Electronic box cooled by natural convection.

SEC et al. v. MRI
SEC Exhibit 1013.0128
IPR2023-00221

The heat lost from the vertical sides of the box is determined from Eq. 5.7. Conversions from English units to metric units are made using Tables 1.3 and 1.10.

$$Q = h_c A\ \Delta t$$

Given $h_c = 0.92$ Btu/hr ft² °F = 0.000124 cal/sec cm² °C

$$A = \frac{[(12)(2) + (8)(2)](6)\ \text{in}^2}{144\ \text{in}^2/\text{ft}^2}$$

$$A = 1.667\ \text{ft}^2 = 1548.6\ \text{cm}^2$$
$$\Delta t = 130 - 80 = 50°\text{F} = 27.8°\text{C}$$

In English units:

$$Q_{\text{sides}} = (0.92)(1.667)(50) = 76.7\ \frac{\text{Btu}}{\text{hr}} \tag{5.15}$$

In metric units:

$$Q_{\text{sides}} = (0.000124)(1548.6)(27.8) = 5.34\ \frac{\text{cal}}{\text{sec}} \tag{5.15a}$$

The natural convection coefficient for the top horizontal surface is obtained from Eq. 5.5 and Table 5.2.

$$h_c = 0.27\left(\frac{\Delta t}{L}\right)^{0.25}$$

Given $\Delta t = 130 - 80 = 50°\text{F}$

$$L = \frac{2(12)(8)}{12 + 8} = 9.6\ \text{in} = 0.80\ \text{ft}$$

$$h_c = 0.27\left(\frac{50}{0.80}\right)^{0.25}$$

$$h_c = 0.76\ \frac{\text{Btu}}{\text{hr ft}^2\ °\text{F}} = 0.000103\ \frac{\text{cal}}{\text{sec cm}^2\ °\text{C}} \tag{5.16}$$

The heat lost from the top surface of the box is determined from Eq. 5.7.

Given $A = \frac{(12)(8)\ \text{in}^2}{144} = 0.667\ \text{ft}^2 = 619.6\ \text{cm}^2$

SEC et al. v. MRI
SEC Exhibit 1013.0129
IPR2023-00221

In English units:

$$Q_{\text{top}} = (0.76)(0.667)(50) = 25.3\,\frac{\text{Btu}}{\text{hr}} \tag{5.17}$$

In metric units:

$$Q_{\text{top}} = (0.000103)(619.6)(27.7) = 1.77\,\frac{\text{cal}}{\text{sec}} \tag{5.17a}$$

The natural convection coefficient for the bottom horizontal surface is obtained from Eq. 5.6 and Table 5.2. Conversions to metric units are made with Tables 1.3 and 1.10.

$$h_c = 0.13\left(\frac{\Delta t}{L}\right)^{0.25}$$

Given $\Delta t = 130 - 80 = 50°\text{F}$

$$L = \frac{2(12)(8)}{12 + 8} = 9.6\text{ in} = 0.80\text{ ft}$$

$$h_c = 0.13\left(\frac{50}{0.80}\right)^{0.25}$$

$$h_c = 0.36\,\frac{\text{Btu}}{\text{hr ft}^2\,°\text{F}} = 0.0000486\,\frac{\text{cal}}{\text{sec cm}^2\,°\text{C}} \tag{5.18}$$

The heat lost from the bottom surface is determined from Eq. 5.7.

Given $A = \frac{(12)(8)\text{ in}^2}{144} = 0.667\text{ ft}^2 = 619.6\text{ cm}^2$

In English units:

$$Q_{\text{bot}} = (0.36)(0.667)(50) = 12.0\,\frac{\text{Btu}}{\text{hr}} \tag{5.19}$$

In metric units:

$$Q_{\text{bot}} = (0.0000486)(619.6)(27.8) = 0.84\,\frac{\text{cal}}{\text{sec}} \tag{5.19a}$$

The total heat lost is the sum of the sides, top, and bottom.

SEC et al. v. MRI
SEC Exhibit 1013.0130
IPR2023-00221

In English units:

$$Q_{\text{total}} = 76.7 + 25.3 + 12.0 = 114.0 \frac{\text{Btu}}{\text{hr}} \tag{5.20}$$

Since 1 watt = 3.413 Btu/hr,

$$Q = 33.4 \text{ watts}$$

In metric units:

$$Q_{\text{total}} = 5.34 + 1.77 + 0.84 = 7.95 \frac{\text{cal}}{\text{sec}} \tag{5.20a}$$

Since 1 watt = 0.239 cal/sec,

$$Q = 33.3 \text{ watts}$$

5.8 FINNED SURFACES FOR NATURAL CONVECTION COOLING

The amount of heat that can be removed from an electronic box that is cooled by natural convection will be substantially increased if the surface area of the box can be substantially increased. One convenient method for increasing the surface area is to add fins with a low thermal resistance. The temperature of the fins will then be nearly equal to the surface temperature of the electronic box. The additional heat transferred to the atmosphere will then be approximately proportional to the increase in the surface area.

Fins will increase the size and weight of the electronic box. This may be a small penalty to pay if the cost is reduced and the reliability is increased by eliminating the need for a cooling fan.

The effectiveness of the finned surface will depend upon the temperature gradient along the fin as it extends from the surface of the electronic box. When the fin has a small temperature gradient, the temperature at the tip of the fin will be nearly equal to the temperature at the base of the fin or the chassis surface, and the fin will have a high efficiency. The natural convection coefficient h_c must be corrected when the fin efficiency is less than 100%. The fin efficiency is shown by Eq. 5.21 [15, 17, 24].

$$\eta = \frac{\tanh md}{md} = \text{fin efficiency} \tag{5.21}$$

SEC et al. v. MRI
SEC Exhibit 1013.0131
IPR2023-00221

where $m = \sqrt{\frac{2h_c}{K\delta}}$

h_c = convection coefficient
= Btu/hr ft² °F or cal/sec cm² °C
K = thermal conductivity of fin material
= Btu/hr ft °F or cal/sec cm °C
δ = fin thickness
= ft, in, cm
d = fin height (distance fin extends from surface of box)
= ft, in, cm

Fins should not be spaced too close together on the surface of a box, because this may result in pinching or choking the natural flow of cooling air. Closely spaced fins may increase the flow resistance and reduce the effective cooling available from the fins. When several fins are stacked together, there should be sufficient space to permit two convection films to develop, as shown in Figure 5.8.

The fin efficiency can also be determined from Figure 5.7 if the value of md is known.

If the fins are spaced closer together than about 0.60 in (1.52 cm) in perfectly still air at sea level, some pinching or choking may occur. However, under some natural convection conditions there is often additional air movement generated by the local ventilation system, so that the fins can be spaced closer together. Tests have shown that fins can be spaced about 0.50 in (1.27 cm) apart under these circumstances without choking the natural convection film coefficient. If the environmental conditions are unknown, a minimum clearance of 0.60 in is recommended to avoid trouble.

Fins can be cast integral with an electronic chassis, or they can be riveted, screwed, or cemented to the chassis structure. A low thermal interface

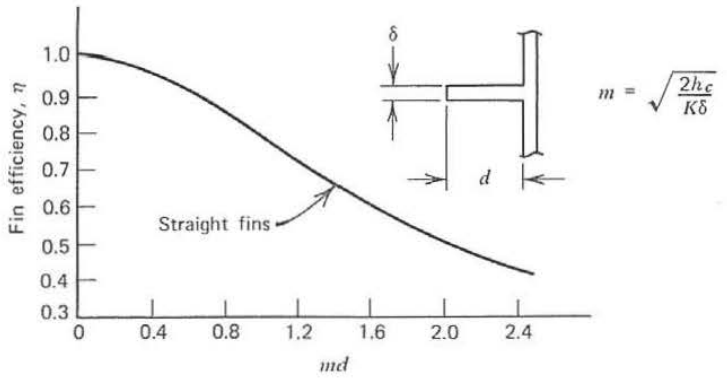


Figure 5.7 Fin efficiency factor for a single fin.

SEC et al. v. MRI
SEC Exhibit 1013.0132
IPR2023-00221

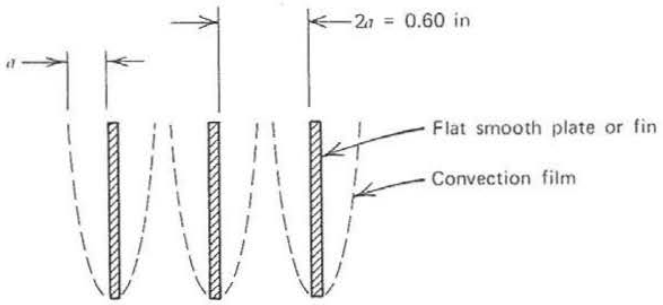


Figure 5.8 Convection film for several fins stacked together.

resistance must be provided between the fins and the chassis to ensure good heat transfer characteristics.

5.9 SAMPLE PROBLEM—COOLING FINS ON AN ELECTRONIC BOX

Determine the amount of heat that can be removed by natural convection from the electronic box in the preceding problem, when cooling fins are added as shown in Figure 5.9.

SOLUTION

The geometry of the electronic box is the same as Figure 5.6; only the vertical fins on the long sides have been added. The spacing between the

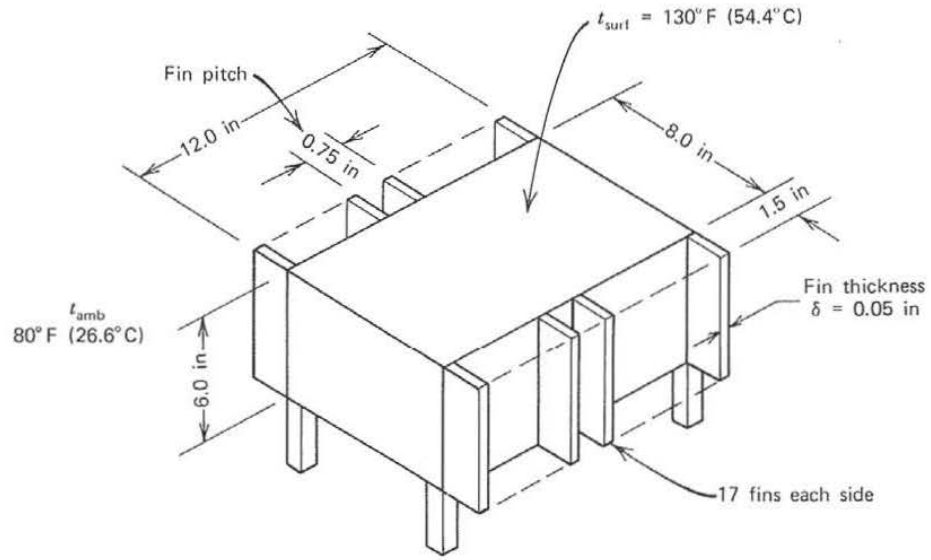


Figure 5.9 Electronic box with fins to improve natural convection cooling.

SEC et al. v. MRI
SEC Exhibit 1013.0133
IPR2023-00221

fin surfaces is 0.75 − 0.05 = 0.70 in (1.78 cm); therefore, no choking or pinching of the natural convection film will occur, even in perfectly still air.

The additional area supplied by the 17 fins on both sides is as follows:

$$A_{\text{fin}} = \frac{2(17)(6)(1.5)(2 \text{ sides})}{144 \text{ in}^2/\text{ft}^2} = 4.25 \text{ ft}^2 = 3948.3 \text{ cm}^2 \tag{5.22}$$

The temperature along the fin will not be constant, so that the fin efficiency η must be determined using Eq. 5.21 or Figure 5.7. The convection coefficient for the vertical sides of the box shown by Eq. 5.14 is used because the fins have the same height as the box.

Given $h_c = 0.92$ Btu/hr ft² °F = 0.000124 cal/sec cm² °C (ref. Eq. 5.14)
$K = 83$ Btu/hr ft °F = 0.343 cal/sec cm °C (aluminum fin conductivity)
$\delta = 0.05$ in/12 = 0.00416 ft = 0.127 cm (fin thickness)
$d = 1.5$ in = 0.125 ft = 3.81 cm (fin height)

In English units:

$$m = \sqrt{\frac{2(0.92)}{(83)(0.00416)}} = 2.308 \frac{1}{\text{ft}} \tag{5.23}$$
$$md = (2.308)(0.125) = 0.288 \text{ (dimensionless)}$$

In metric units:

$$m = \sqrt{\frac{(2)(0.000124)}{(0.343)(0.127)}} = 0.0755 \frac{1}{\text{cm}} \tag{5.24}$$
$$md = (0.0755)(3.81) = 0.288 \text{ (dimensionless)}$$

Substitute into Eq. 5.21 for fin efficiency.

$$\eta = \frac{\tanh 0.288}{0.288} = \frac{0.280}{0.288} = 97.2\% \tag{5.25}$$

Figure 5.7 could also have been used to obtain a fin efficiency of 97.2%.

The effective fin natural convection coefficient can now be determined with the fin efficiency factor using Eq. (5.14)

$$h_{\text{eff}} = \eta h_c = (0.972)(0.92) = 0.89 \frac{\text{Btu}}{\text{hr ft}^2 \text{ °F}} \text{ (English units)}$$

$$h_{\text{eff}} = (0.972)(0.000124) = 0.000121 \frac{\text{cal}}{\text{sec cm}^2 \text{ °C}} \text{ (metric units)}$$

SEC et al. v. MRI
SEC Exhibit 1013.0134
IPR2023-00221

The additional heat lost by the fins only using natural convection is determined from Eq. 5.7.

$$Q = h_{\text{eff}} A \, \Delta t$$

In English units:

$$Q = (0.89)(4.25 \text{ ft}^2)(50°\text{F}) = 189.1 \frac{\text{Btu}}{\text{hr}} \tag{5.26}$$

In metric units:

$$Q = (0.000121)(3948.3 \text{ cm}^2)(27.8°\text{C}) = 13.28 \frac{\text{cal}}{\text{sec}} \tag{5.26a}$$

The total heat loss from the box, in English units, is obtained by adding Eq. 5.20 to Eq. 5.26.

$$Q_{\text{total}} = 114.0 + 189.1 = 303.1 \frac{\text{Btu}}{\text{hr}} = 88.8 \text{ watts} \tag{5.27}$$

The total heat lost from the box, in metric units, is obtained by adding Eq. 5.20a to Eq. 5.26a.

$$Q_{\text{total}} = 7.95 + 13.28 = 21.23 \frac{\text{cal}}{\text{sec}} = 88.8 \text{ watts} \tag{5.27a}$$

The fins thus increase the heat transfer capability of the box from 33.3 watts to 88.8 watts for the same temperature conditions, but with a slight increase in the volume and weight requirements.

5.10 NATURAL CONVECTION ANALOG RESISTOR NETWORKS

Computers are often used to solve heat transfer problems involving natural convection. This usually requires some type of mathematical model to simulate the physical characteristics of the system. Some analog resistor network models are shown in Figures 3.9 through 3.13.

The natural convection resistance is defined by Eq. 5.28.

$$R = \frac{1}{h_c A} \tag{5.28}$$

Natural convection coefficients h_c for heated vertical plates can be determined from Eq. 5.4. Equation 5.5 can be used for heated horizontal

SEC et al. v. MRI
SEC Exhibit 1013.0135
IPR2023-00221

plates facing upward. The area A represents the physical size of the plane surface in consistent units of ft^2 for English units or cm^2 for metric units.

Considering only single surfaces for the plates shown in Figure 5.10, for a temperature rise of 75°F, the thermal resistances would be as follows:

Vertical plate:

$$h_c = 0.29\left(\frac{\Delta t}{L}\right)^{0.25} = 0.29\left(\frac{75}{3/12}\right)^{0.25} = 1.21\ \frac{\text{Btu}}{\text{hr ft}^2\ ^\circ\text{F}}$$

$$A = \frac{(3)(6)\ \text{in}^2}{144\ \text{in}^2/\text{ft}^2} = 0.125\ \text{ft}^2$$

$$R_{\text{vert}} = \frac{1}{h_c A} = \frac{1}{(1.21)(0.125)} = 6.61\ \frac{\text{hr}\ ^\circ\text{F}}{\text{Btu}} \tag{5.29}$$

Horizontal plate:

$$h_c = 0.27\left(\frac{\Delta t}{L}\right)^{0.25}$$

$$L = \frac{L \times w}{(L + w)/2} = \frac{(3)(6)}{(3 + 6)/2} = 4.0\ \text{in} = 0.333\ \text{ft}$$

$$\text{h}_c = 0.27\left(\frac{75}{0.333}\right)^{0.25} = 1.04\ \frac{\text{Btu}}{\text{hr ft}^2\ ^\circ\text{F}}$$

$$R_{\text{hor}} = \frac{1}{h_c A} = \frac{1}{(1.04)(0.125)} = 7.69\ \frac{\text{hr}\ ^\circ\text{F}}{\text{Btu}} \tag{5.30}$$

When a mathematical model for natural convection of a large vertical plate is being established, the conduction resistance between each node will be required, as well as the convection resistance to the ambient, as shown in Figure 5.11.

The small plate sections a through g are used to compute the conduction

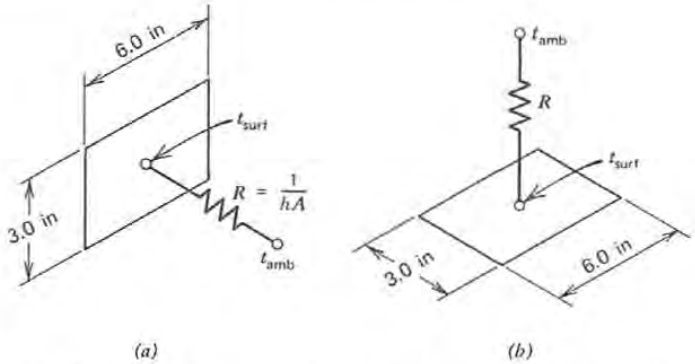


Figure 5.10 Natural convection models for vertical and horizontal plates.

SEC et al. v. MRI
SEC Exhibit 1013.0136
IPR2023-00221

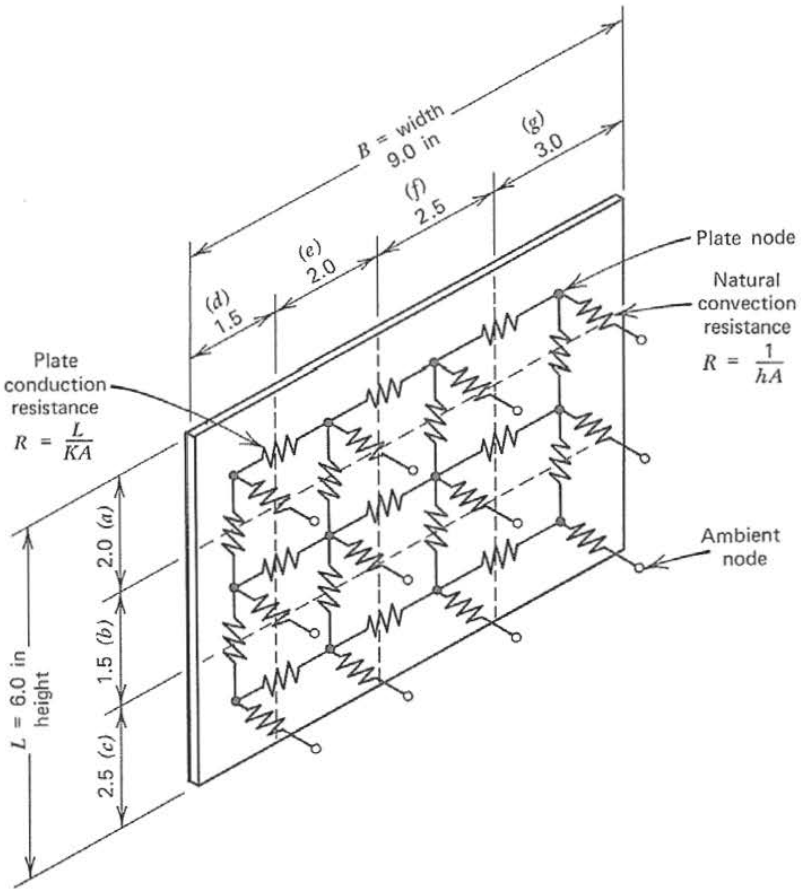


Figure 5.11 Mathematical model of a plate that combines conduction and natural convection thermal resistors.

resistances between each node point on the plate. However, the natural convection coefficient h_c depends upon the full height of the plate. Therefore, the full plate height of 6.0 in must be used for computing the natural convection coefficient, not the individual plate sections of a, b, and c.

5.11 NATURAL CONVECTION COOLING FOR PCBs

Printed circuit boards (PCBs) are often mounted within chassis that are completely open at the top and bottom to allow the free flow of cooling air. These PCBs should be mounted vertically with a minimum space of about 0.75 in (1.90 cm) free flow distance between the components and the adjacent PCB, to prevent pinching or choking of the natural convection flow, as shown in Figure 5.12.

Radiation cooling effects must be ignored here because the PCBs "see" each other. This simply results in a radiation heat interchange with no real

SEC et al. v. MRI
SEC Exhibit 1013.0137
IPR2023-00221

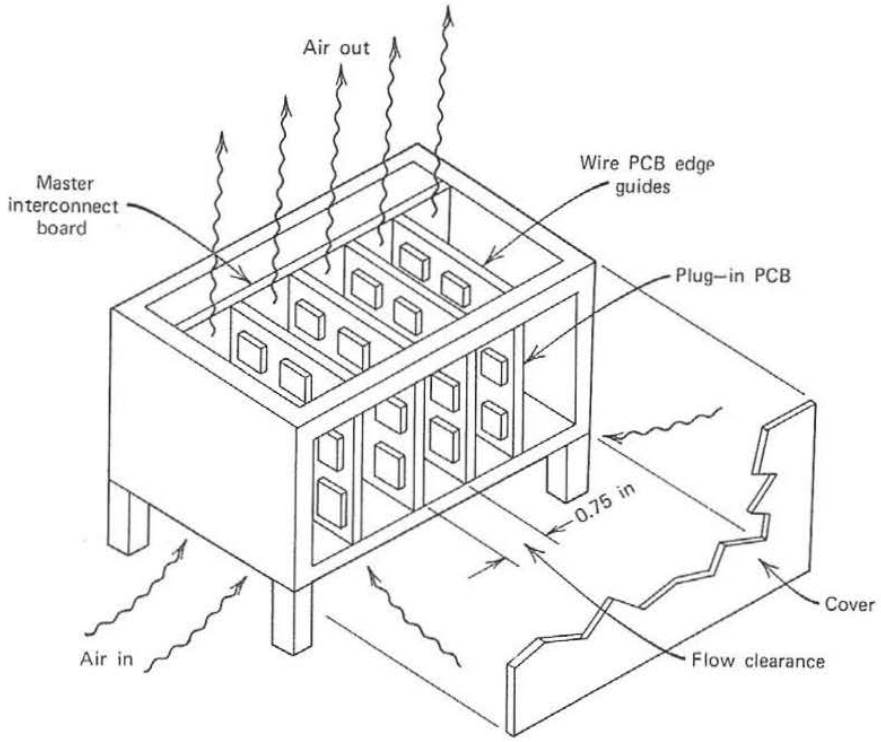


Figure 5.12 Plug-in PCBs cooled by natural convection.

cooling, except perhaps for the PCBs on each end of the chassis. These cooler end slots should be reserved for the highest-power-dissipating PCBs.

In a 160°F (71°C) ambient, a power dissipation of 0.10 watt/in² (0.0155 watt/cm²) will result in a component surface temperature of about 212°F (100°C) for a small PCB.

This considers only one side of the PCB, when electronic components are uniformly distributed with a uniform heat load along the surface of the PCB. When electronic components are mounted on both sides of the PCB with a uniform heat distribution, the total amount of power that can be dissipated from one PCB is double the power that can be dissipated from only one side of the PCB for a 30°C temperature rise. (54°F rise)

The foregoing approximation can be verified by considering a 6 by 9 in PCB, similar to Figure 5.11, with electronic components uniformly distributed on one side. The temperature rise from the component surface to the still ambient is 54°F (30°C). The natural convection coefficient is determined from Eq. 5.4.

$$h_c = 0.29\left(\frac{54}{6/12}\right)^{0.25} = 0.93\,\frac{\text{Btu}}{\text{hr ft}^2\,^\circ\text{F}}$$

SEC et al. v. MRI
SEC Exhibit 1013.0138
IPR2023-00221

The amount of heat lost by convection from one side of the PCB is determined from Eq. 5.7.

$$Q = 0.93\left(\frac{6 \times 9\ \text{in}^2}{144\ \text{in}^2/\text{ft}^2}\right)(54°\text{F}) = 18.8\ \frac{\text{Btu}}{\text{hr}} = 5.5\ \text{watts}$$

$$\text{power density} = \frac{Q}{A} = \frac{5.5\ \text{watts}}{6 \times 9\ \text{in}^2} = 0.10\ \frac{\text{watt}}{\text{in}^2} \tag{5.31}$$

Equation 5.31 shows that a value of 0.10 watt/in^2 is a good approximation for the allowable power dissipation of a PCB cooled by natural convection.

Electronic components are usually mounted on one side of a PCB with the lead wires passing through the PCB to the opposite side for wave or flow soldering. Some of the heat dissipated within the electronic components will flow through the electrical lead wires to the back side of the PCB, as shown in Figure 1.3. When the back side of the PCB has a large number of copper printed circuit runs, they will act as heat spreaders. The back side of the PCB will then be capable of providing additional surface area for heat removal by natural convection. Thermal tests on PCBs with plated throughholes and flow-soldered electronic components, show that the back side of the PCB can increase the total effective heat transfer area about 30%.

If the electronic components are mounted on one side of the PCB, and lap solder joints are used (the electrical lead wires do not extend through the PCB with lap solder joints), the heat will not flow through to the back side of the PCB as easily. The back side of the PCB will not provide additional surface area for cooling under these circumstances.

5.12 NATURAL CONVECTION COEFFICIENT FOR ENCLOSED AIR SPACE

Printed circuit boards that are to be cooled by natural convection are sometimes placed so close to a wall that the natural flow of cooling air is blocked. The resulting clearance is too small to allow air movement, so that the clearance becomes an enclosed air space. Although natural convection is suppressed, heat transfer may still take place by conduction across the air gap. An equivalent convection coefficient can then be defined, based upon the heat conducted across the air gap as shown by Eq. 3.1.

$$Q = KA\,\frac{\Delta t}{L} \qquad \text{(ref. Eq. 3.1)}$$

For convection, the general heat transfer relation is shown by Eq. 5.7.

$$Q = h_c A\,\Delta t \qquad \text{(ref. Eq. 5.7)}$$

SEC et al. v. MRI
SEC Exhibit 1013.0139
IPR2023-00221

When the heat transferred is the same in both cases, the equations must be equal so that an air gap convection coefficient h_{AG} can be obtained as shown in Eq. 5.32.

$$h_{AG}A\,\Delta t = KA\frac{\Delta t}{L}$$

$$h_{AG} = \frac{K}{L} \tag{5.32}$$

where K = thermal conductivity of air gap
L = thickness of air gap

The use of Eq. 5.32 can be demonstrated with a sample problem.

5.13 SAMPLE PROBLEM—PCB ADJACENT TO A CHASSIS WALL

An electronic chassis was designed for natural convection cooling, so that a clearance of 0.75 in (1.905 cm) was provided between the PCBs and components. However, a design change required the addition of another PCB, which might reduce the clearance too much unless the new PCB is placed very close to the side wall of the chassis, with a clearance of only 0.20 in (0.51 cm). The PCB measures 6 by 9 in and dissipates 5.5 watts. The electronic chassis must operate at sea level conditions in a maximum ambient temperature of 110°F (43.3°C). The maximum allowable component surface temperature is 212°F (100°C) with the chassis shown in Figure 5.13. The aluminum chassis has a polished finish that has a low emissivity, so that the heat lost by radiation is small. Determine if the design is adequate.

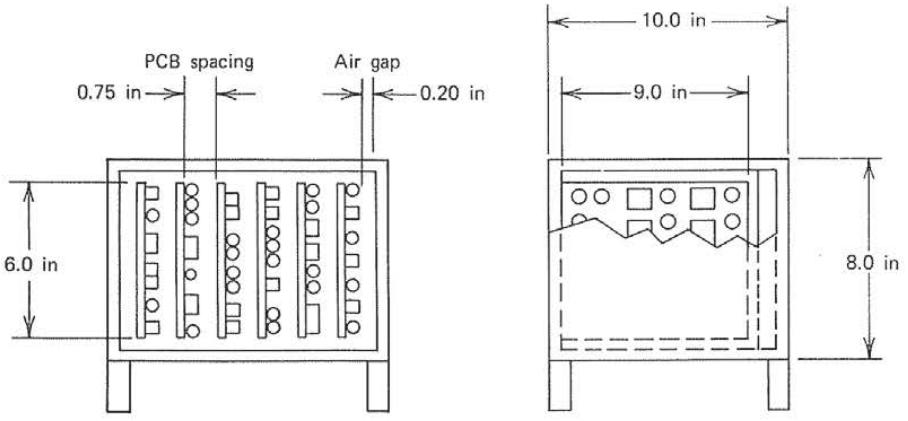


Figure 5.13 PCB spaced close to an end bulkhead.

SEC et al. v. MRI
SEC Exhibit 1013.0140
IPR2023-00221

SOLUTION

The previous sample problem, shown in Figure 5.12, demonstrated that a clearance of 0.75 in will result in a temperature rise of 54°F (30°C) for the conditions given. This will result in a component surface temperature of 164°F, which is satisfactory. Therefore, only the PCB mounted close to the end wall has to be examined.

Heat from the components must flow to the outside ambient. This will require the heat to flow across two major resistance areas, the internal air gap of 0.20 in (R_1) and the external convection film (R_2), as shown in Figure 5.14.

The thermal conductivity of the air in the air gap is unknown, so that an average air temperature of 175°F is assumed and verified later. Determine the resistance with the convection coefficient for the 0.20 in air gap.

$$R_1 = \frac{1}{h_{AG}A} \text{ (ref. Eq. 5.28)}$$

Given $K = 0.017$ Btu/hr ft °F (ref. Figure 6.28)

$$L = \frac{0.20}{12} = 0.0167 \text{ ft (air gap)}$$

$$h_{AG} = \frac{K}{L} = \frac{0.017}{0.0167} = 1.018 \frac{\text{Btu}}{\text{hr ft}^2\,°\text{F}} \text{ (air gap convection)}$$

$$A = \frac{6 \times 9 \text{ in}^2}{144} = 0.375 \text{ ft}^2 \text{ (PCB area)}$$

$$R_1 = \frac{1}{(1.018)(0.375)} = 2.62 \frac{\text{hr °F}}{\text{Btu}} \tag{5.33}$$

The external convection coefficient must be estimated because the tem-

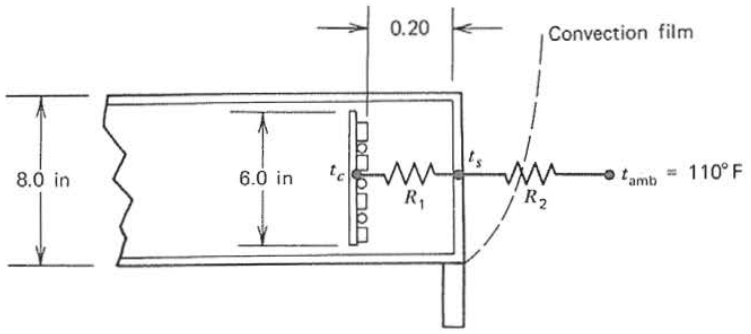


Figure 5.14 Thermal resistances in the heat flow path from PCB components to the outside ambient.

SEC et al. v. MRI
SEC Exhibit 1013.0141
IPR2023-00221

perature rise from the surface of the chassis to the ambient is unknown. In general, natural convection coefficients for this type of structure will range from about 0.6 to about 1.0 Btu/hr ft² °F. A value of 0.8 Btu/hr ft² °F is assumed to start. This can be changed if the analysis shows there is a large error.

$$R_2 = \frac{1}{h_c A} \qquad \text{(ref. Eq. 5.28)}$$

Given $h_c = 0.8$ Btu/hr ft² °F (assumed to start)

$$A = \frac{8 \times 10 \text{ in}^2}{144} = 0.555 \text{ ft}^2$$

$$R_2 = \frac{1}{(0.8)(0.555)} = 2.25 \frac{\text{hr °F}}{\text{Btu}} \tag{5.34}$$

The temperature rise across each thermal resistor is determined from Eq. 5.35. For resistor R_1:

$$\Delta t_1 = QR_1 \tag{5.35}$$

Given $Q = (5.5 \text{ watts})(3.413 \text{ Btu/hr/watt}) = 18.77 \text{ Btu/hr}$

$R_1 = 2.62$ hr °F/Btu (ref. Eq. 5.33)

$$\Delta t_1 = (18.77)(2.62) = 49.2°\text{F} \tag{5.36}$$

For resistor R_2:

$$\Delta t_2 = QR_2 = (18.77)(2.25) = 42.2°\text{F} \tag{5.37}$$

The natural convection coefficient was assumed to be 0.8 Btu/hr ft² °F. The actual value can now be determined from Eqs. 5.4 and 5.37 using a vertical chassis wall height of 8.0 in.

$$h_c = 0.29\left(\frac{\Delta t}{L}\right)^{0.25} = 0.29\left(\frac{42.2}{8/12}\right)^{0.25} = 0.81 \frac{\text{Btu}}{\text{hr ft}^2 \text{ °F}} \tag{5.38}$$

This compares well with the assumed value.

The surface temperature of the component on the end PCB can be determined as follows:

$$t_c = t_{\text{AMB}} + \Delta t_1 + \Delta t_2 \tag{5.39}$$

$$t_c = 110 + 49.2 + 42.2 = 201.4°\text{F}\ (94.1°\text{C}) \tag{5.40}$$

SEC et al. v. MRI
SEC Exhibit 1013.0142
IPR2023-00221

The average air temperature in the 0.20 in gap between the wall and the component can now be determined. A temperature of 175°F was assumed to obtain the air thermal conductivity.

The average air temperature in the gap is obtained from Eq. 5.40.

$$t_{av} = \frac{t_{comp} + t_{wall}}{2} = \frac{201.4 + (110 + 42.2)}{2} = 176.8°F$$

This compares well with the assumed value.

Therefore, since Eq. 5.40 shows that the component surface temperature is below the maximum allowable value of 212°F (100°C), the design is satisfactory.

If the inside and outside surfaces of the chassis are anodized or painted any color except silver, heat transfer by radiation will be increased and the surface of the PCB will be cooler.

5.14 HIGH ALTITUDE EFFECTS ON NATURAL CONVECTION

All of the previous natural convection equations were determined for sea level conditions, where the atmospheric pressure is 14.7 lb/in^2 (1034.4 g/cm^2). Natural convection depends upon a reduction in the density of the cooling air as it picks up heat, which forces it to rise in a gravity field.

At high altitudes the air density is much lower than it is at sea level conditions, so that the ability of the air to pick up heat is sharply reduced. This, in turn, reduces the ability of the air to rise, which results in a lower convection coefficient.

An examination of Eq. 5.1 shows that the convection coefficient h is related to the air density as $(\rho^2)^m$, where m has a value of 0.25. This means that the convection coefficient is related to the square root of the air density, or $\sqrt{\rho}$. When this is combined with the basic gas law $PV = WRT$, it shows that the air density is directly related to the air pressure when the air temperature does not change. Therefore, the convection coefficient h is also proportional to the square root of the air pressure, as shown in Eq. 5.41.

$$h_{alt} = h_{sl}\sqrt{\frac{\rho_{alt}}{\rho_{sl}}} = h_{sl}\sqrt{\frac{P_{alt}}{P_{sl}}} \tag{5.41}$$

For example, when the natural convection coefficient at sea level is 0.75 Btu/hr ft^2 °F, the natural convection coefficient at an altitude of 30,000 ft can be determined from the atmospheric pressure, which is 4.37 psia, as

SEC et al. v. MRI
SEC Exhibit 1013.0143
IPR2023-00221

shown in Table 6.5. When there is no change in the temperature,

$$h_{alt} = 0.75\sqrt{\frac{4.37}{14.7}} = 0.41\ \frac{\text{Btu}}{\text{hr ft}^2\ ^\circ\text{F}} \tag{5.42}$$

When there is a change in the temperature, the air density ratio should be used instead of the pressure ratio, for greater accuracy.

The thermal conductivity of air changes very little with a change in altitude or air pressure, up to an altitude of about 82,000 ft (see Figure 3.19). The thermal conductivity of air does change, however, with the air temperature, as shown in Figure 6.28. Therefore, at high altitudes the natural convection is reduced, but conduction heat transfer across small air gaps is not affected, as long as there is no change in the temperature. These altitude and temperature characteristics can be demonstrated with a sample problem.

5.15 SAMPLE PROBLEM—PCB COOLING AT HIGH ALTITUDES

For the previous sample problem (Section 5.13 and Figure 5.13) determine the component surface temperature that will result at an altitude of 30,000 ft. The aluminum chassis has an iridited finish, which has a low emissivity, so that heat lost by radiation is small and is ignored.

SOLUTION

The external convection coefficient from the outer surface of the box will be reduced at the high altitude. Its exact value is not known. Equation 5.41 might be used with the results from the previous sample problem, shown by Eq. 5.38, to obtain a rough approximation of the convection coefficient at 30,000 ft. However, this would not be accurate, because the outer surface of the box will be hotter at the high altitude condition than at the sea level condition, which will change the convection coefficient. A better approximation of the external convection coefficient at the 30,000 ft altitude condition can be obtained by combining the following equations:

$$h_{sl} = 0.29\left(\frac{\Delta t}{L}\right)^{0.25} \qquad \text{ref. Eq. 5.4)}$$

$$h_{alt} = h_{sl}\sqrt{\frac{P_{alt}}{P_{sl}}} \qquad \text{(ref. Eq. 5.41)}$$

$$h_{alt} = 0.29\left(\frac{\Delta t}{L}\right)^{0.25}\sqrt{\frac{P_{alt}}{P_{sl}}}$$

$$h_{alt} = 0.29\left(\frac{\Delta t}{L}\right)^{0.25}\sqrt{\frac{4.37}{14.7}} = 0.158\left(\frac{\Delta t}{L}\right)^{0.25}$$

SEC et al. v. MRI
SEC Exhibit 1013.0144
IPR2023-00221

Consider another equation:

$$Q = h_{\text{alt}} A\, \Delta t \qquad \text{(at altitude, ref. Eq. 5.7)}$$

or

$$h_{\text{alt}} = \frac{Q}{A\, \Delta t}$$

Equate the convection coefficients h_{alt} and solve for Δt.

$$\frac{Q}{A\, \Delta t} = \frac{0.158\, \Delta t^{0.25}}{L^{0.25}}$$

$$QL^{0.25} = 0.158A\, \Delta t^{1.25}$$

$$\Delta t = \left(\frac{QL^{0.25}}{0.158A}\right)^{0.8} \tag{5.43}$$

For a power dissipation of 5.5 watts (18.77 Btu/hr), a vertical height of 8.0 in (0.666 ft) and a surface area of $8 \times 10 = 80$ in^2 (0.555 ft^2), the resulting external temperature rise at an altitude of 30,000 ft becomes

$$\Delta t_2 = \left[\frac{(18.77)(0.666)^{0.25}}{(0.158)(0.555)}\right]^{0.8} = 67.4°\text{F} \tag{5.43a}$$

This compares to a temperature rise of only 42.2°F across the external convection film at sea level conditions, as shown by Eq. 5.37.

The resulting external convection coefficient at the 30,000 ft altitude can be obtained from Eq. 5.7.

$$h_{\text{alt}} = \frac{Q}{A\, \Delta t} = \frac{18.77 \text{ Btu/hr}}{(0.555 \text{ ft}^2)(67.4)}$$

$$h_{\text{alt}} = 0.50 \frac{\text{Btu}}{\text{hr ft}^2 \text{ °F}}$$

The thermal conductivity of the air in the 0.20 in air gap will increase to about 0.018 Btu/hr ft °F because the average air temperature in the gap will increase to about 200°F. This average air gap temperature can be verified after the preliminary temperature rise calculations. Then, following Eq. 5.32.

$$h_{AG} = \frac{K}{L} = \frac{0.018 \text{ Btu/hr ft °F}}{0.0167 \text{ ft air gap}} = 1.078 \frac{\text{Btu}}{\text{hr ft}^2 \text{ °F}} \tag{5.44}$$

SEC et al. v. MRI
SEC Exhibit 1013.0145
IPR2023-00221

Substitute into Eq. 5.28, using the area 0.375 ft², as shown in Eq. 5.33.

$$R = \frac{1}{(1.078)(0.375)} = 2.47 \frac{\text{hr °F}}{\text{Btu}}$$

From Eq. 5.35, using a heat value of $Q = 5.5$ watts = 18.77 Btu/hr, we obtain

$$\Delta t_1 = \left(18.77 \frac{\text{Btu}}{\text{hr}}\right)\left(2.47 \frac{\text{hr °F}}{\text{Btu}}\right) = 46.4°\text{F} \tag{5.45}$$

The component surface temperature at 30,000 ft is determined from Eq. 5.39. The ambient temperature is 110°F.

$$t_c = 110 + 46.4 + 67.4 = 223.8°\text{F (106.5°C)} \tag{5.46}$$

Since this exceeds the maximum allowable component surface temperature of 212°F (100°C), the design is not acceptable. Again, one more check must be made of the average temperature in the air gap to veryify the thermal conductivity used.

$$t_{av} = \frac{t_{comp} + t_{wall}}{2} = \frac{223.8 + (110 + 67.4)}{2} = 200.6°\text{F}$$

This is close to the assumed value of 200°F, so that the results are valid.

5.16 RADIATION COOLING OF ELECTRONICS

Hot bodies emit thermal radiation that is transferred by electromagnetic waves, which can travel through a vacuum or through a gas with relatively little absorption. The true nature of radiation and its transfer mechanism is not completely clear, but it is known that radiation in free space travels at the speed of light.

When radiation waves strike a second body, part of the radiated energy may be absorbed, part may be reflected, and part may be transmitted through the body, as in the case of glass and quartz. The relative amounts of energy transferred in each case depends upon the temperature, the surface characteristics, the body geometry, the material, and the wavelength.

The amount of heat radiated from a surface is related to a factor called the emittance e. This is dependent upon the condition of the surface, such as its temperature, roughness, finish, coating, and if it is a metal, the amount of oxidation. The emittance is a ratio of the amount of radient energy emitted by a body to that emitted by the ideal radiator, which is a black body. The emittance is always less than 1.0 because the perfect black body

SEC et al. v. MRI
SEC Exhibit 1013.0146
IPR2023-00221

does not really exist in nature. The black body is used as a standard for comparison with the radiation characteristics of other bodies.

A perfect black body is one that absorbs 100% of all the energy it receives, without reflecting any energy. The emissive power of a black body depends only upon its temperature. The black body radiates the maximum amount of energy at any wavelength, as shown in Figure 5.15.

The ability of a body to absorb part of the total radiated energy it receives is called the absorptivity α. Again, the amount of energy that the body receives is dependent upon the material and the temperature (or wavelength) of the emitter.

When a body is at thermal equilibrium, the emissivity and the absorptivity of that body will be equal. When this condition exists for the ideal radiator or black body, the emissivity and the absorptivity both equal unity.

The total area under any one of the indicated temperature curves in Figure 5.15 is used to determine the total energy radiated by the black body at that temperature.

Metals and plastics normally used in electronics are very opaque to radiation. Therefore, radiation can be considered as a surface phenomenon, where the effective surface depth for metals is only about 0.0001 in (0.000254 cm) and for nonmetals about 0.02 in (0.0508 cm).

The emissivities of clean metallic materials are generally very low while the emissivities of nonmetallic materials are much higher. Also, nonmetallic materials, in contrast to metals, show decreasing emissivities with an increase in temperature.

A spacecraft in close orbit around the earth or around the moon can receive radiation from the sun and the earth (or moon) at the same time. Most of the radiation is in the ultraviolet and visible light frequency range,

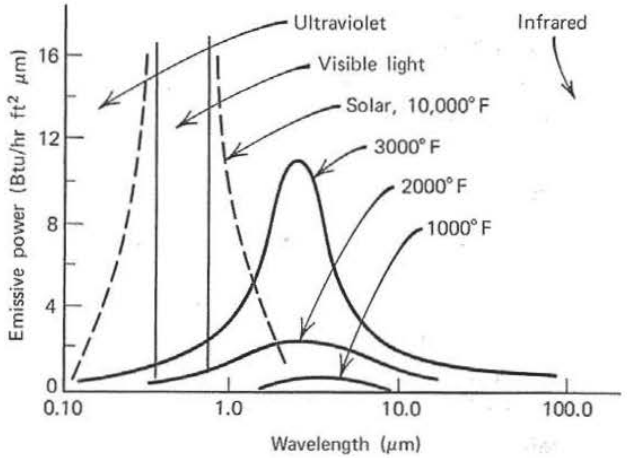


Figure 5.15 Change of emissive power with relation to the temperature of an ideal radiator or black body.

SEC et al. v. MRI
SEC Exhibit 1013.0147
IPR2023-00221

which has a wavelength up to about 1 micrometer (μm), as shown in Figure 5.15 [4, 5, 15].

In general, dull dark surfaces are good absorbers and good emitters of heat, so they have high emittance values. Polished metal surfaces have very low emissivities. However, as metals oxidize, their emissivities increase rapidly. Sandblasted metal surfaces also show emissivities that are much higher than those for polished surfaces.

That emissivity is a surface phenomenon can be demonstrated by placing a thin coat of paint on a highly polished metal surface. Before painting a polished copper surface, the emissivity will be about 0.03. However, with just a thin coat of lacquer, 0.0005 in (0.00127 cm) thick, the emissivity of the same surface will increase sharply to about 0.80.

For temperatures normally encountered in electronic equipment, the radiation wavelength is relatively long, about 7 μm, and is not perceived by the human eye. This represents the infrared region, where color is relatively unimportant, so that black paints have radiation characteristics that are very similar to white paints. In this region a good emitter is a good receiver of radient energy. Dark colors will not emit or absorb any more heat than

Table 5.3 Typical Emissivities 100°C [7, 8]

Material	Emissivity, e
Aluminum	
Polished	0.06
Commercial sheet	0.09
Rough polish	0.07
Iridited	0.10
Gold, highly polished	0.018–0.035
Steel, polished	0.06
Iron, polished	0.14–0.38
Cast iron, machine cut	0.44
Brass, polished	0.06
Copper, polished	0.023–0.052
Polished steel casting	0.52–0.56
Glass, smooth	0.85–0.95
Aluminum oxide	0.33
Anodized aluminum	0.81
White enamel on rough iron	0.9
Black shiny lacquer on iron	0.8
Black or white lacquer	0.80–0.95
Aluminum paint and lacquer	0.52
Rubber, hard or soft	0.86–0.94
Water (32–212°F)	0.95–0.96

SEC et al. v. MRI
SEC Exhibit 1013.0148
IPR2023-00221

light colors in the infrared region, because the emissivities and absorptivities are not related to the paint color.

Sometimes a surface will have characteristics similar to a black body, as shown in Figure 5.15. The shape or its emissive power, as a function of the temperature, will then be the same as the black body, except that the height is reduced by the numerical value of the emissivity. This type of surface is defined as being gray. In most electronic heat transfer calculations, the surfaces are assumed to be gray even though they do not meet the exact requirements for a gray surface.

As the temperature of a body increases, the radiation wavelength decreases until the visible spectrum is reached at about 0.7 μm. In the visible light spectrum, the amount of radiation energy absorbed by a body is sharply affected by the color. Dark colors in the visible spectrum absorb more radient energy than light colors. This can be dramatically shown by placing your hand on a black automobile roof and a white automobile roof at the same time, after both cars have been sitting in the hot sun for several hours. The black roof will be substantially hotter than the white roof.

Heat from both the black and white automobile surfaces will be radiated back to the surrounding areas in the long wavelength, or infrared region, where these paints act like gray surfaces.

Typical emissivity values for various materials are shown in Table 5.3.

Values of α/e for several different materials are shown in Table 5.4.

Relations for the emissivities for two different surfaces are shown in Table 5.5.

Radiation really involves an exchange of energy, since part of the energy received is always reflected back. For normal engineering purposes, the reflected energy is taken care of by the emissivity and the absorptivity of each body involved in the exchange.

The general radiation equation for the exchange of radiant heat energy between two non-black bodies is shown in Eq. 5.47 [7, 8, 15, 17, 24].

$$Q = \sigma f e A\ (T_1^4 - T_2^4) \tag{5.47}$$

Consistent sets of units must be used as defined for English units and metric units in Table 5.6.

The heat transmitted by radiation is directly related to the surface area A. If the surface is irregular, the projected area should be used. This is the area that would result when a tightly stretched string is held over various parts of the surface, as shown in Figure 5.16.

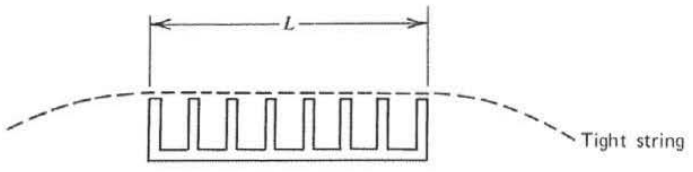


Figure 5.16 Projected area of an irregular surface.

SEC et al. v. MRI
SEC Exhibit 1013.0149
IPR2023-00221

Table 5.4 Some Representative Values of α/e

Material	Solar Absorptivity (α)	Total Normal Emissivity at 100°F (e)	α/e
Aluminum, freshly evaporated	0.10	0.025	4
Gold	0.16	0.02	8
Molybdenum	0.43	0.03	14
Palladium	0.41	0.03	14
Platinum	0.33	0.03	11
Rhodium	0.28	0.02	14
Silver	0.07	0.01	7
Tantalum	0.59	0.02	29
Aluminum, vapor deposit on 0.5 mil Mylar			
Aluminum side	0.13	0.04	3.25
Mylar side	0.18	0.90	0.20
White paint, 1 mil thick	0.15	0.94	0.16
Black paint, 1 mil thick	0.97	0.94	1.03
Clear varnish on aluminum 1 mil	0.20	0.80	0.25
Clear varnish on aluminum 0.05 mil coat	0.20	0.10	2.0
Black lacquer, 0.24 mil coat	0.96	0.48	2.0

Table 5.5 Emissivity Factors for Different Configurations

Configuration	e
Infinite parallel planes	$\dfrac{1}{1/e_1 + 1/e_2 - 1}$
One surface (1) completely surrounded by another (2)	$\dfrac{1}{1/e_1 + A_1/A_2\,[1/e_2 - 1]}$
Completely enclosed body 1, small compared with enclosing body 2	e_1
General case for two surfaces	$e_1 \times e_2$

SEC et al. v. MRI
SEC Exhibit 1013.0150
IPR2023-00221