Heat exchangers cool hot plug-in pc boards

When device power levels and packing densities rise, the thermal deficiencies of printed-circuit boards must be compensated by efficient heat exchangers

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☐ On most counts, the plug-in printed-circuit board deserves its status as today's unofficial industry-wide standard. Mounting vertically in an equipment case, it can easily be withdrawn when replacement is necessary. Yet it is well protected from shock and vibration, being held rigidly in place by card slides.

Thermally, however, the plug-in printed-circuit board is much less impressive. Neither epoxy-glass nor paper-based boards are good heat conductors. Also, the thermal paths from hot devices on the boards to the outside world are often long and hinder cooling.

Proof of the board's inadequacy as a heat conductor is that a temperature gradient of 707°C is required to drive just 1 watt of heat through a piece of epoxy-glass board only 1 inch square and 20 mils thick. This determination was made from an equation that enables the designer to calculate thermal gradient whenever heat flow can be considered unidirectional. The equation is:

$$\Delta T_{max} = QL/8Ktw$$

where

ΔT_{max} = temperature gradient to the hottest spot in the board, in degrees centigrade

Q = heat transfer by conduction along the board, in watts

L = span of the board between card guides, in inches

K = thermal conductivity of the board, in watts per inch°C

t = board thickness, in inches

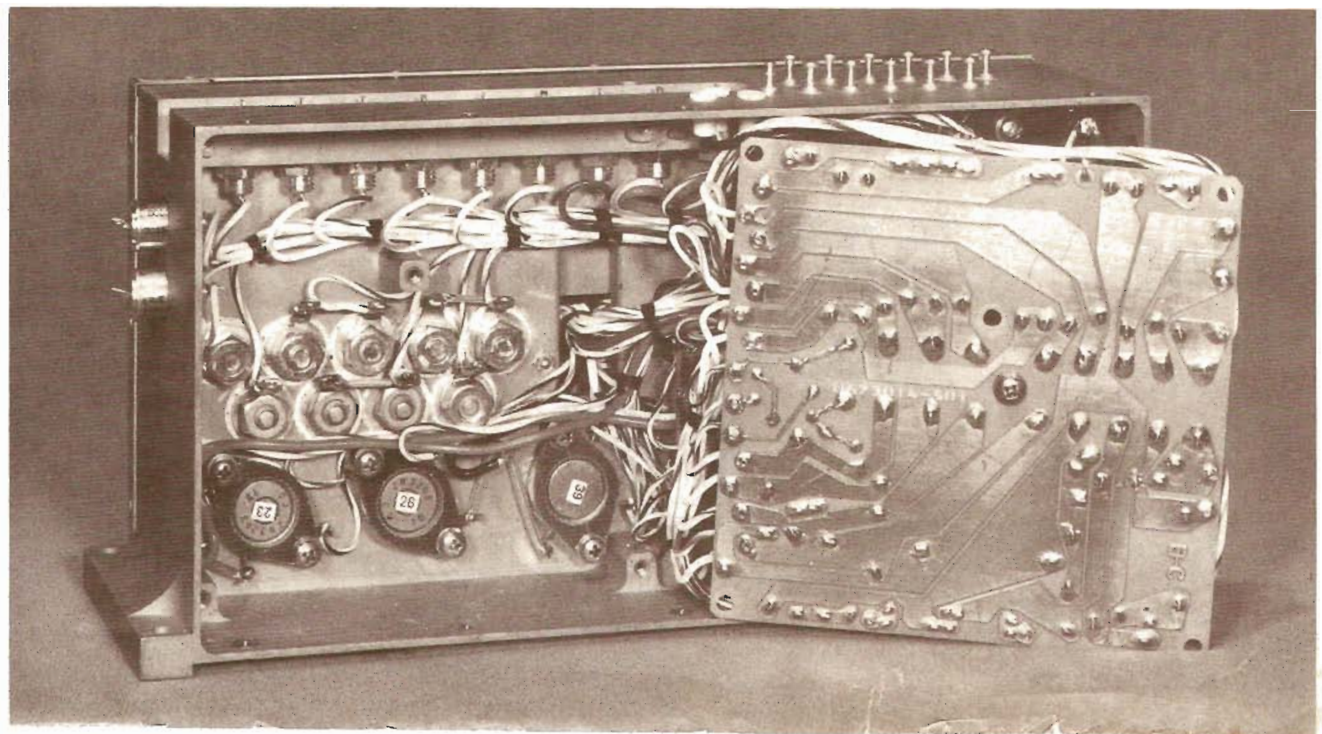
w = length of each interface between board and card guide, in inches.

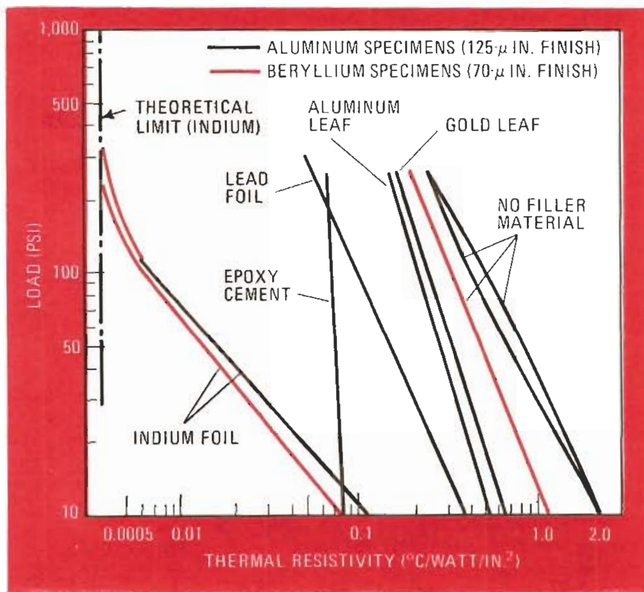
The equation assumes uniform power dissipation over the surface of the board and is realistic if the designer has optimized both heat spreading and component location on the board's surface.

One way to improve heat flow through a board is to use the copper conductors on its surface to transfer heat. Being a fine thermal conductor, the copper lowers thermal resistance significantly—though precisely how much it is lowered is difficult to calculate because the pattern etched into the conductor markedly reduces heat transfer. For instance, if just 10% of the copper is removed from a fully-clad board, thermal resistance of the overall board can increase by a factor of 17.[1]

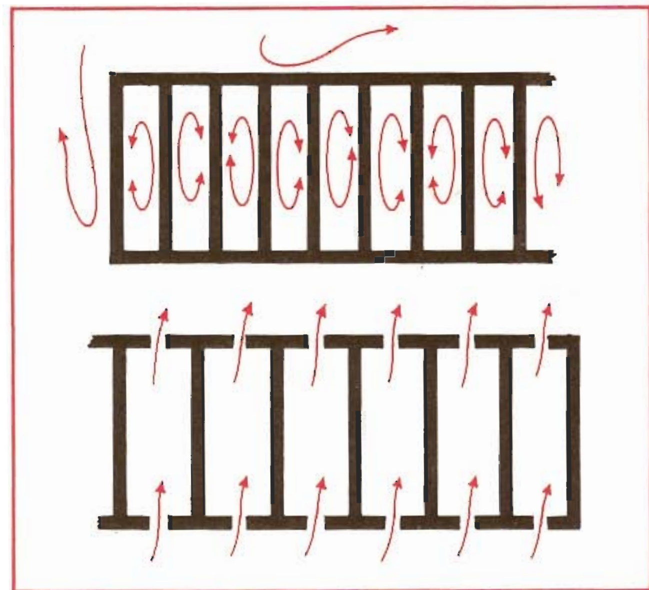
Materials other than epoxy-glass can be used for pc-board construction to upgrade their heat-transfer char-

1. Destined for the moon. Rarefied atmospheres deny package designers the advantage of convective cooling. This assembly, part of a radar system carried on the Apollo 17, employs a highly conductive frame to absorb heat from the printed-circuit board.





2. Beware of the boundary. Fillers between printed-circuit boards and the card slide, and high-compression forces aid heat flow across the interface. Data is based on research performed by MIT Instrumentation Laboratory.



3. Alternatives. Closed convective system shown in (a) prevents contaminants from being swept in by a moving air stream. But switching to an open system (b) boosts cooling capability per unit volume tenfold—10 watts/in.³ versus 1 watt/in.³

acteristics. But generally they fail to improve heat transfer enough to compensate for the electrical constraints they impose. Instead, it is frequently better to switch to a heat-conducting frame to support the pc board.

The heat-conducting frame

This technique was used to good effect for the Apollo 17, in a pc-board assembly that was part of the coherent synthetic aperture radar (CSAR). Figure 1 shows details of that assembly. Effective conductive cooling is a must in space, where the lack of atmosphere robs the designer of convective cooling. In the CSAR assembly, heat flowed from the board to the housing through the threaded bosses on which the board was mounted. Maximum temperature rise was kept low because the thermal path to a boss from any heat-producing component was kept short.

The Standard Hardware Program (SHP) developed by the U.S. Navy also utilizes heat-conducting frames to guarantee adequate heat transfer from its modules.[2]

However, only a poor thermal path from board to frame is provided by the usual card slides. The problem is that ease of maintainability and accessibility demands boards that slip easily in and out of card slides—but the thermal interface between such boards and slides is not good. Fortunately, card slides can often be modified to provide a large positive area of contact that will optimize heat flow across the interface.

Figure 2 summarizes the results of some interface resistivity studies.[3] It shows that various filler materials can be used to lower the thermal resistivity of board/slide interfaces. Note also the negative slope of the plots, which denotes that high compressive forces along the interface also lower thermal resistivity.

Enter the ambient

Regardless of how effectively such conduction paths are enhanced, convective transfer to the ambient fluid (usually air) often emerges as the principal heat-transfer mode in electronic equipment.

Two geometries are common in convective transfer. Figure 3a illustrates a closed system in which transfer is in effect a two-step process. Heat is moved from the board surfaces to the surrounding air by natural or forced convection, and the air is then cooled by natural- or forced-convection transfer to the equipment case.

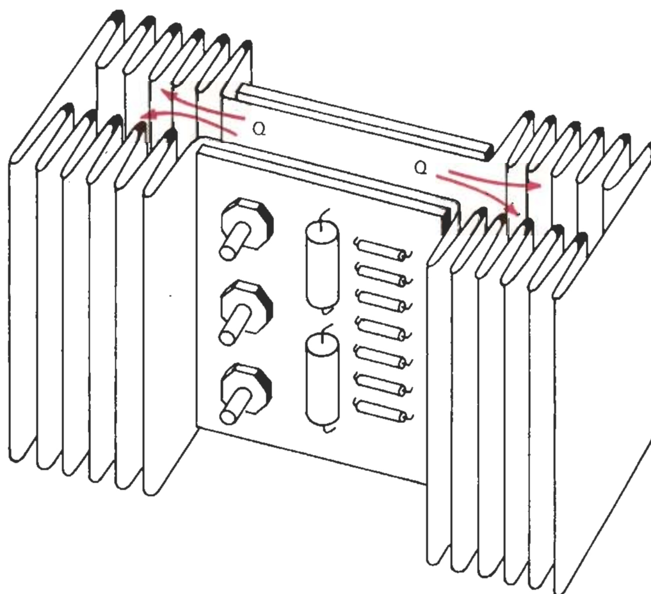
In the open system shown in Fig. 3b, the air is not entrapped, but enters the enclosure, sweeps across the pc boards, and then exits carrying the heat to the environment. There is no intermediate transfer to and from the equipment case. But such a system is often unacceptable because it can transport moisture and other harmful contaminants.

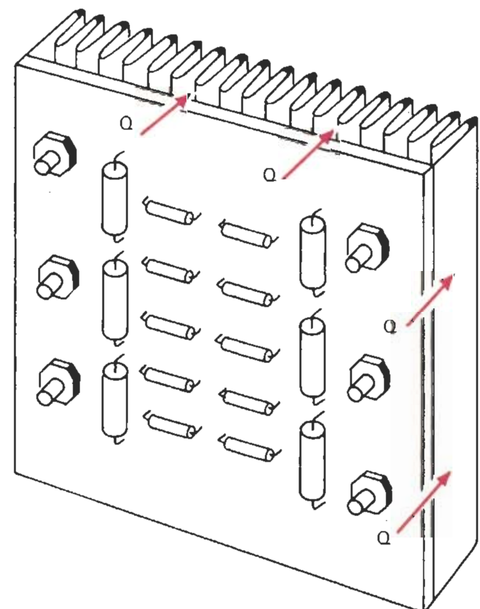
In either type of convective system, orientation and spacing of the boards play an important role in determining component temperatures. So do the flow rate and temperature of the cooling medium. Table 1 lists the range of typical heat-transfer rates for both open and closed plug-in pc-board designs. Note that the power density for a well-designed closed system where the exterior cooling is by natural convection ranges

TABLE 1: HOW COOLING MODE AFFECTS POWER DISSIPATION

System type	Internal cooling mode	Exterior cooling mode	
		Natural convection (W/in.³)	Forced convection (W/in.³)
Open		0.5 – 1	5 – 10
Closed	Natural convection	0.1 – 0.25	0.2 – 1.0
Closed	Forced convection	0.2 – 0.8	0.5 – 3.0
Closed	Conduction	0.4 – 1.5	2.0 – 4.0

	RESISTANCE (°C / W)		ΔT°C	
	IN-LINE	COPLANAR	IN-LINE	COPLANAR
CONDUCTIVE RESISTANCES				
R_{TJ} DEVICE-JUNCTION TO CASE	2.5	2.5	27.5	27.5
R_{T1} DEVICE-INTERFACE	1.12	1.12	12.3	12.3
R_{T2} THERMAL SPREADING RESISTANCE	0.65	0.65	7.1	7.1
R_{T3} BASEPLATE RESISTANCE (VERTICAL)	4.83	4.83	3.4	3.1
R_{T4} BASEPLATE RESISTANCE (HORIZONTAL)	1.97	–	39.8	0
CONVECTIVE RESISTANCES				
$T_{SO} - T_{SI}$ COOLING AIR TEMPERATURE RISE				26.3
1 / hA CONVECTION TRANSFER RESISTANCE	–	0.53	–	29.0*
* INCLUDES $T_{SO} - T_{SI}$				
TOTAL TEMPERATURE RISE °C			90.1	79.0





4. Go coplanar. The in-line construction (a) accounts for the large temperature rise—90.1°C. By contrast, the heat-transfer path in the coplanar structure (b) is very short, and temperature rise is significantly less—66.1°C.

from 0.10 W/in.^{3} to 1.5 W/in.^{3}. Also, for the internal forced-air cooling modes, total volume must not be so large that the space consumed by blowers and ducting becomes a significant fraction of the total volume. Otherwise, the listed values become invalid.

Looking at one design

Assume that a designer attempts to house a 100-watt uhf radio transmitter-receiver combination in a standard cabinet designed to mount printed-circuit boards. Detailed analysis of a particular design reveals that the maximum power dissipation that can be rejected in such a cabinet (4.87 in. wide by 7.62 in. high by 19.56 in. deep) is limited to 56 W at sea level and to 28 W if the equipment is operated at high altitude, where there is little convective cooling. Clearly, plug-in pc-board construction would not be appropriate for this equipment.

The power dissipation of the equipment, broken down by its component modules, is given in Table 2. Checking the power densities of each module against the criteria of Table 1 indicates that forced-air convection is necessary in two modules—the transmitter and the power supply. Since the equipment is intended for the military, however, an open system with forced-air convection directly over the circuit cards would be unacceptable because of possible contamination. So a closed-system, forced-air cold plate is a likely alternative.

Forced-air cooling differs from natural convection in that the driving force circulating the air is a mechanical pump rather than natural buoyancy induced by temperature gradients. This significantly increases the value of the parameter known as film coefficient (h), thereby upgrading the effectiveness of the surface area (A) of the heat-exchanging structure.

The basic relationship for convective transfer across a boundary is:

$$Q = hA\Delta T$$

where

Q = power, in watts

h = film coefficient, in W/ft^2-°C

A = area, in square feet

ΔT = temperature gradient, in degrees centigrade.

It turns out that the film coefficient is about an order of magnitude higher in forced convective transfer than it is in natural convection—2.6 to 7.9 W/ft^2-°C compared with 0.2 to 0.4 W/ft^2-°C.

But this improvement has to be traded off against the energy that must be expended on forcing air past the surface that needs to be cooled. This usually translates as electric power driving a fan or blower and can be defined as:

$$P_f = KvH \quad (1)$$

where

P_f = fan power required to deliver the necessary air velocity, in watts

K = a constant of 0.023 W-minute/ft-lb

v = air flow rate, in ft^3/minute

H = frictional air pressure loss, in pounds/ft^2.

Thus design optimization comes down to the task of maximizing the heat transfer required in terms of Q and P_f.

TABLE 2:
THERMAL BUDGET FOR A 100 WATT TRANSMITTER-RECEIVER

Module	Peak power		Average power 50% duty cycle (W)	Power density (W/in.^{3})
	Transmit (W)	Receive (W)		
Guard receiver	1.8	1.8	1.8	0.09
Frequency/control	18.9	18.3	18.6	0.19
Main receiver	4.8	4.8	4.8	0.05
Transmitter	272.0	25.9	149	2.72
Power supply	98.5	35.2	66.5	0.76
Totals	396	86	241.7	0.96

As if this were not enough, the designer must usually restrict the physical size of the heat-exchanging structure to the smallest volume possible. In the case of the uhf transmitter-receiver, the space available for the rf power output stages, which dissipate 250 W, is 100 cubic inches, or roughly 8¾ by 5¾ by 2 in. The task requires that the junction temperatures of the rf power transistors be cooled to within safe limits.

Cold-plate considerations

Two cold-plate configurations were analyzed to determine the temperature fields which develop in each. Figure 4a is a straightforward variation of the plug-in printed-circuit board; the transistors are stud-mounted on an aluminum plate 90 mils thick that has integral heat sinks at both ends. Figure 4b shows how the board and the heat exchanger can be repackaged so that they become coplanar. The coplanar structure proved to be superior because it considerably shortened the conduction paths between each transistor and the heat exchanger.

The assumptions and design constraints used in the analysis of these two configurations are:

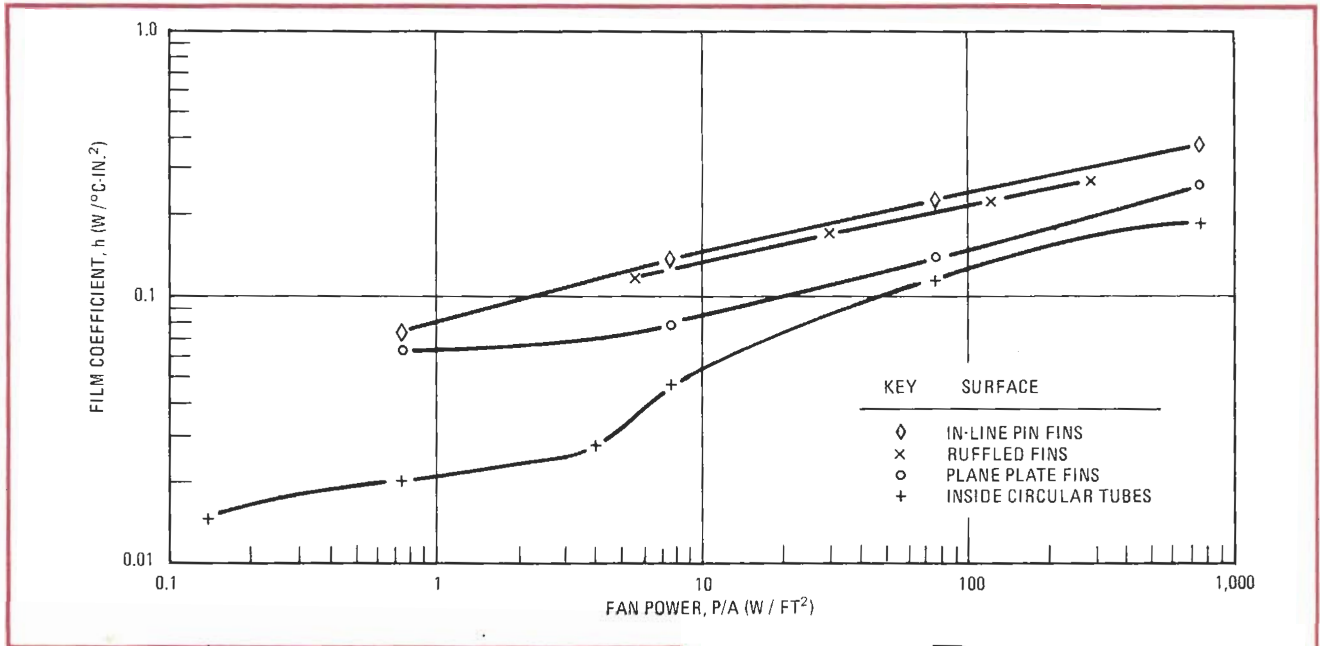
- Each transistor dissipates 11 W.
- Power is dissipated uniformly on the pc board at 2.3 W/in.^{2}.
- The equipment chassis is 90-mil-thick aluminum, with a thermal conductivity, K, of 4.4 W/in.-°C.
- Maximum transistor junction temperature is 150°C.
- Operating environmental temperature is 71°C.

The results of the analysis are listed in the table of Fig. 4. The critical ΔT, which is the temperature rise from the ambient to each device junction, can be expressed as:

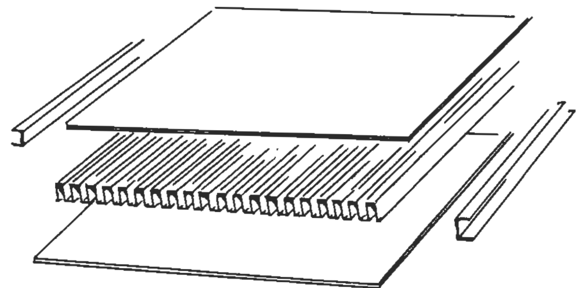
$$T_J - T_A = Q(R_{TJ} + R_{T1} + R_{T2} + R_{T3} + R_{T4} + 1/hA)$$

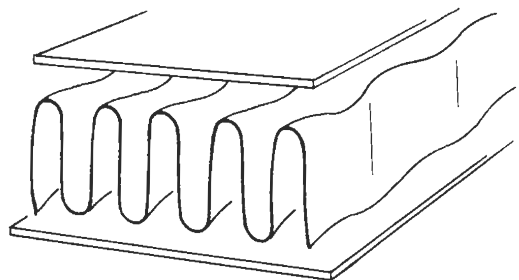
The values and definitions of the thermal resistances are in Fig. 4. The subscripts represent thermal resistances which are conductive paths. The quantity 1/hA is the thermal resistance across the convective interface of the heat-exchanger surface.

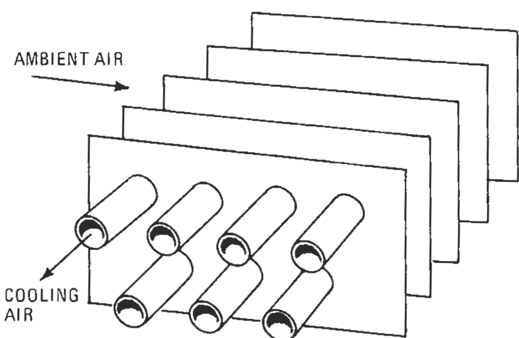
If the conductive resistances are assumed to be known, then the design goal is to assure that the value of 1/hA will be small enough to hold T_J below 150°C. The film coefficient h is determined by the fluid dynam-



5. Different geometries, equal areas. In-line pin fin is the most effective convective surface, according to these plots of film coefficient, h, versus fan power.







ics of the system and is largely a function of fan input power. The heat-transfer surface area (A) is a function of heat exchanger type and volume. Thus the required value of 1/hA can be achieved by proper selection of heat exchanger type and size, and adequate fan power.

The analysis demonstrates that the in-line configurations of Fig. 4a won't do the job. If the in-line construction were selected, the 90.1°C rise would boost the junction temperature to 161.1°C, above the design limit of 150°C. Just how big this rise would be in actuality would depend on the value of 1/hA, because 1/hA has been assumed to be zero in the in-line case. But it really doesn't matter because the allowable gradient budget has been consumed in conduction drops. It is therefore impossible to maintain the desired temperature, regardless of the heat exchanger selected.

The horizontal baseplate resistance (R_{t4}) with a resistance of 1.97°C/w is the major contributor to the temperature rise. If a designer wants to stick with the in-line design he might reduce this resistance by using a thick chassis.

However, the coplanar design will certainly do the job. Here a value of 0.53°C/w is required for 1/hA, to maintain the hottest transistor below the maximum allowable temperature of 150°C. There is obviously a tradeoff between supplying more air to the heat exchanger and providing more heat exchange surface so that the exchange can make a closer approach to the exit air temperature.

Once the basic packaging structure has been selected, the next step in the design is to select a forced-air heat exchanger.

Exchanging heat

The heat exchanger enables heat to cross the boundary from a conductive region to a moving fluid such as

air. Since its design is a major engineering challenge, it is worth summarizing the factors that go into a design analysis and to establish a design selection sequence.[4,5]

The prime considerations are the size and geometry of the heat exchanger structure. Heat transfer through the exchanger is expressed as:

$$Q = hA(T_H - T_S)$$

where

h = film coefficient of heat transfer, in W/in.^{2}-°C

A = area, in square inches

T_H = heat exchanger temperature

T_S = cooling air temperature.

As has been shown in the example, the designer wishes to maximize both h and A so as to minimize the temperature gradient ($T_H - T_A$).

The relationship which determines the air temperature rise in the heat exchanger is expressed in the equation:

$$Q = mc_p (T_{SO} - T_{SI})$$

where

m = mass flow rate, in pounds per second

c_p = specific heat of the fluid at constant pressure, in W-sec/lb-°C

T_{SO} = cooling air temperature at the exchanger outlet, in °C

T_{SI} = cooling air temperature at the exchanger inlet, in °C.

The design goal here is to maximize the air flow rate (m) so as to minimize the temperature drop to be provided by the exchanger. However, a price is paid in electrical power required to energize the fan as can be seen from Eq. 1. In this case, air flow rate, v, as well as the air pressure drop, h, must be minimized for minimum fan power consumption.

The key variables in this group of equations—h, H and v—are interrelated for a given type of heat-exchanger surface. By carefully evaluating these variables, it is possible to tailor a heat-exchange system to a given application.[4,5]

Surface considerations

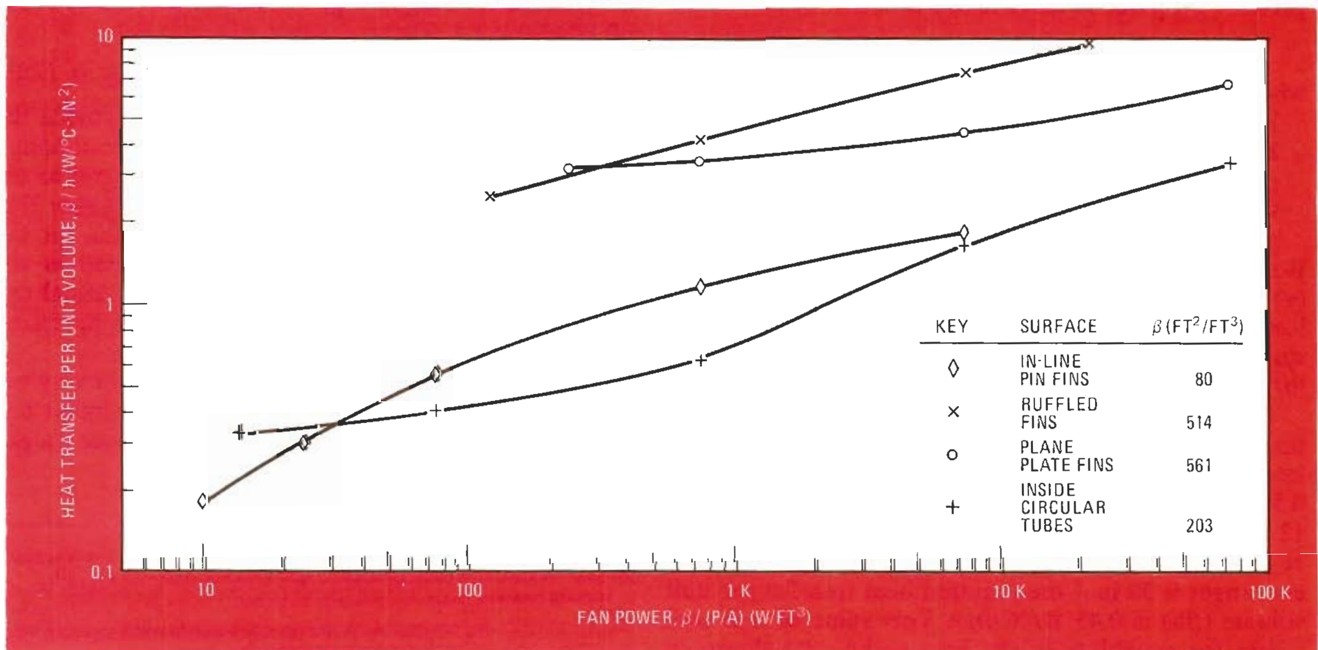
There is a considerable variation in the performance of various heat-exchanging surfaces. The value of h versus air power per unit cross-sectional area is plotted for a number of surfaces in Fig. 5. Note that the pin-fin exchanger delivers a value of h that is three and a half to four and a half times higher than the value of competing structures.

A useful figure of merit for evaluating a heat-exchanging surface is defined as the amount of heat exchanger surface contained in a unit volûme or A/V. It is assigned the symbol β. From the standpoint of maximum βh, the ruffled fin provides the most heat transfer per unit of volume and thus offers the designer a very compact exchanger.

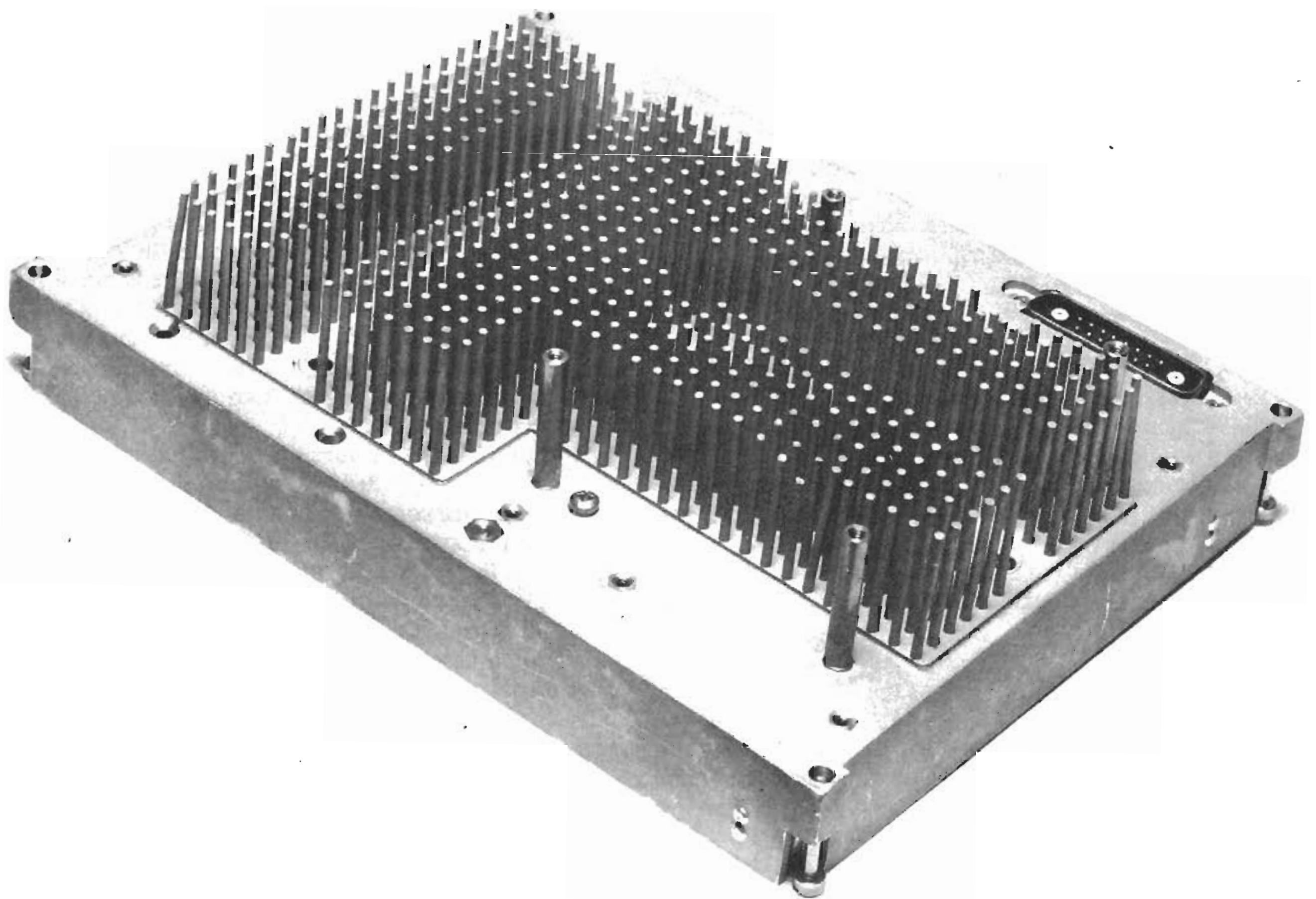
Figure 6 compares several surfaces on the basis of heat transfer per unit volume versus air friction power per unit volume. In effect, both the ordinate and abscissa in Fig. 5 have been multiplied by β. Thus the ordinate h becomes hA/V, expressed in W/in.^{3}-°C. The abscissa is the frictional air power per unit volume dissipated in the heat exchanger, expressed in W/in.^{3}. The values do not include other frictional losses or fan efficiency—typically 15% to 30% in small air-moving devices.

If the designer wants to include these losses, he can multiply the abscissa values by a number ranging from 7 to 13 to determine the approximate fan power. In the usual design operating range, this type of exchanger can reject 1.50 to 3.00 W/°C in.^{3} with a fan power requirement of 300 to 750 W/in.^{3}.

The form factor, which is the width-to-height ratio of a forced-air heat exchanger, depends heavily on the quantity of air passing through a given cross section. A



6. Equal volumes. By multiplying β (heat-exchanger surface area per unit volume) by the film coefficient and fan power, heat-exchanger surfaces can be compared on an equal volume basis. The ruffled-fin exchanger comes out on top.



7. Pin-fin exchanger. Die-cast heat exchanger safely dissipates 100 watts of power and fits into a volume of just 50 cubic inches. Stud-mounted transistors are in valleys between pins.

high-performance heat exchanger will generally require a large cross section to minimize air temperature rise and acoustic noise.

Pressure drops can build up quickly if there are long narrow ducts and many turns in the path or if there are expansions and contractions in the cross section. The pressure drop due to these effects is of the form:

$$P = k_1 \rho v^2 / 2g$$

where

P = air pressure, in lb

k_1 = a dimensionless constant related to geometry

ρ = density of air, in lb/ft^3

v = air velocity, in ft/min

g = 32.2 ft/s^2.

It is wise to keep air flow rate low so that the air velocity (v) does not exceed 500 to 800 ft to limit pressure-drop losses. A good value for air flow often used in military systems and a good starting point in any design is 4 lb/min/kW.

To return to the coplanar exchanger of Fig. 4b, a thermal budget for convective transfer can be calculated. The quantity 1/hA had a calculated value of 0.53°C/w for each transistor. If the exchanger contains 12 transistors, the total heat transfer requirement is 12 × 1/0.53 = 22.6W/°C. If the available volume for the exchanger is 50 in.,[3] the required heat transfer per unit volume (βh) is 0.45 w/°C-in.^{3}. This value of h is well within the capability of the heat exchangers shown in Fig. 6. Pin fins are selected because they can easily be integrated into the module enclosure. Pins spaced at 5.35 per lineal inch facilitate die-casting the exchanger.

To determine the fan power required, the following relationship can be used:

$$P_{\text{fan}} = (\beta P/A)(V\, r_p)$$

where (βP/A) is the power required per cubic foot, plotted as the abscissa in Fig. 6; V is the volume of the heat exchanger in cubic feet; and r_p is the ratio of fan power to core friction, assumed in this case to be 13. Since the required βh is 0.45 w/°C-in.^{3}, then for the in-line pin fin exchanger, Fig. 6 indicates a value of βP/A of 0.045 kW/ft^3. Then:

$$P_{\text{fan}} = (0.045)(50/1728)(13)kW = 0.017kW = 17W$$

Thus a fan with 17 w of fan power will provide the required heat transfer. Figure 7 shows the actual design of the exchanger. Note that the fins are integral to the chassis, thus doing away with the thermal losses that would accompany an attempt to fasten pins on the chassis. The semiconductors are stud-mounted in the two rows in the spaces between the pins. A thermal test program has confirmed the validity of the predicted temperature profile.

By applying such design principles from the very beginning of a packaging design, equipment designers can avoid the compromises in reliability and power output which have plagued designs in the past. □

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