

Table 5.6 Units Used with Radiation Exchange Equation

Item	Symbol	English Units	Metric Units
Radiant heat exchange	Q	$\frac{\text{Btu}}{\text{hr}}$	$\frac{\text{cal}}{\text{sec}}$
Stefan-Boltzmann constant	σ	$\frac{0.1713 \times 10^{-8} \text{ Btu}}{\text{hr ft}^2 \text{ }^\circ\text{R}^4}$	$\frac{1.355 \times 10^{-12} \text{ cal}}{\text{sec cm}^2 \text{ }^\circ\text{K}^4}$
View factor	f	Dimensionless	Dimensionless
Emissivity	e	Dimensionless	Dimensionless
		$\frac{1}{1/e_1 + 1/e_2 - 1}$	$\frac{1}{1/e_1 + 1/e_2 - 1}$
Area	A	ft^2	cm^2
Absolute temperature	T	$^\circ\text{F} + 460 = ^\circ\text{R}$	$^\circ\text{C} + 273 = ^\circ\text{K}$

The finned surface increases the effective area for convection heat transfer but has very little affect on the effective area for radiation heat transfer. Radiating surfaces must be able to “see” each other in order to transfer heat. The fins block the line of sight so that only the projected area bounded by the tight string is effective for the transfer of heat.

5.17 RADIATION VIEW FACTOR

The view factor f is defined as the fraction of the radiation leaving surface 1 in all directions, which is intercepted by surface 2, as shown in Figure 5.17.

Heat transferred by radiation can be evaluated as taking place from surface 1 to surface 2, or from surface 2 to surface 1, if both are black bodies, whichever is more convenient. The product of the area times the

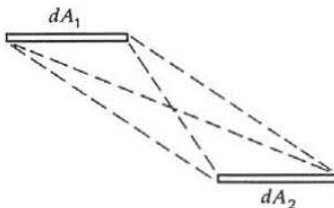


Figure 5.17 Areas involved in a radiation interchange.

view factor works in both directions, as shown in Eq. 5.48 [15, 24].

$$A_1 f_{1-2} \equiv A_2 f_{2-1} \quad (5.48)$$

Many complex equations have been derived, and many computer programs are available, which relate the view factor between two surfaces as a function of the geometry. Sometimes it is possible to determine the view factor between two rectangular plates with a piece of string, using the cross string method.

For example, consider two long, narrow, rectangular parallel plates as in Figure 5.18, where only the narrow edges are shown. Using a piece of string, the dimensions across the free edges are shown by the dashed lines.

As shown by Eq. 5.48, the product of the area times the view factor, per unit length of depth, is the sum of the lengths of the crossed strings stretched from the edges of the plates, minus the sum of the lengths of the noncrossed strings similarly stretched from the edges of the plates, all divided by 2. This is shown as follows:

$$A_1 f_{1-2} = A_2 f_{2-1} = \frac{\text{sum cross strings} - \text{sum noncross strings}}{2} \quad (5.49)$$

$$A_1 f_{1-2} = A_2 f_{2-1} = \frac{(8.54 + 5.0) - (5.0 + 3.0)}{2} = 2.77 \frac{\text{in}^2}{\text{in of depth}}$$

The view factor from plates 1 to 2 (f_{1-2}) is

$$f_{1-2} = \frac{A_2 f_{2-1}}{A_1} = \frac{2.77 \text{ in}^2/\text{in}}{(4)(40) \text{ in}^2} (40 \text{ in}) = 0.692 \text{ (dimensionless)} \quad (5.50)$$

The view factor from plates 2 to 1 (f_{2-1}) is

$$f_{2-1} = \frac{A_1 f_{1-2}}{A_2} = \frac{2.77 \text{ in}^2/\text{in}}{(8)(40) \text{ in}^2} (40 \text{ in}) = 0.346 \text{ (dimensionless)} \quad (5.51)$$

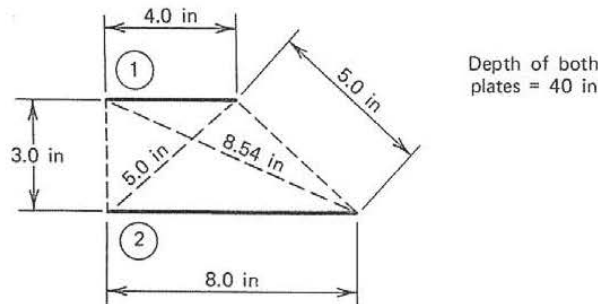


Figure 5.18 End view of two long rectangular parallel plates.

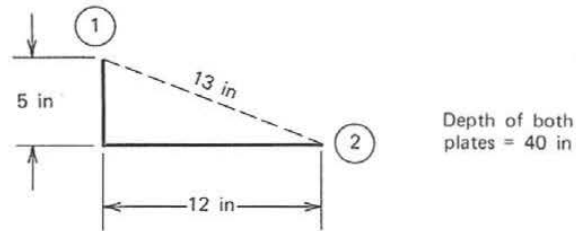


Figure 5.19 End view of two long rectangular perpendicular plates.

The view factor for two long rectangular plates that intersect in a right angle, as shown in Figure 5.19, can be determined with Eq. 5.49. Here the cross strings are the width of each plate, and the noncross string is the hypotenuse of the triangle.

$$A_1 f_{1-2} = A_2 f_{2-1} = \frac{5 + 12 - 13}{2} = 2 \frac{\text{in}^2}{\text{in depth}}$$

The view factor from plates 1 to 2 (f_{1-2}) is

$$f_{1-2} = \frac{A_2 f_{2-1}}{A_1} = \frac{2 \text{ in}^2/\text{in}}{(5)(40) \text{ in}^2} (40 \text{ in}) = 0.40 \text{ (dimensionless)} \quad (5.52)$$

The view factor from plates 2 to 1 (f_{2-1}) is

$$f_{2-1} = \frac{A_1 f_{1-2}}{A_2} = \frac{2}{(12)(40)} \times 40 = 0.167 \text{ (dimensionless)} \quad (5.53)$$

Figure 5.20 shows two infinite rectangular parallel plates of width b , directly above one another and separated by a distance h . The view factor

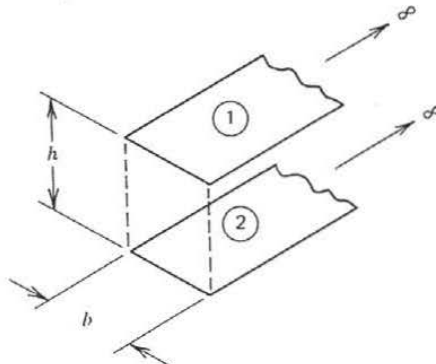


Figure 5.20 Two infinite parallel plates.

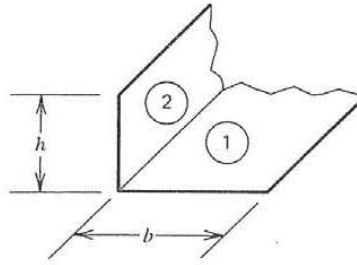


Figure 5.21 Two infinite perpendicular plates.

f_{1-2} is shown by Eq. 5.54.

$$f_{1-2} = \sqrt{1 + \left(\frac{h}{b}\right)^2} - \frac{h}{b} \quad (5.54)$$

When the distance between the plates (h) is equal to the width (b) of the plates, the view factor f_{1-2} is shown by Eq. 5.55.

$$f_{1-2} = \sqrt{1 + (1)^2} - 1 = 1.414 - 1.0 = 0.414 \quad (5.55)$$

Figure 5.21 shows two infinite rectangular plates of width b and h that intersect each other at an angle of 90° . The view factor f_{1-2} is shown by Eq. 5.56.

$$f_{1-2} = \frac{1}{2} \left[1 + \frac{h}{b} - \sqrt{1 + \left(\frac{h}{b}\right)^2} \right] \quad (5.56)$$

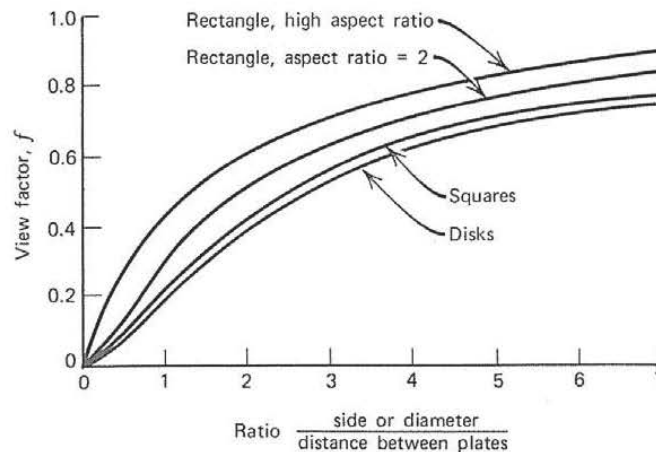


Figure 5.22 View factor for flat parallel plates directly opposed. (From Heat Transmission, by McAdams, McGraw Hill Book Co.)

When the width of both plates are equal, the view factor f_{1-2} is shown by Eq. 5.57.

$$f_{1-2} = \frac{1}{2} [1 + 1 - \sqrt{1 + (1)^2}] = 0.293 \quad (5.57)$$

Figure 5.22 shows view factors for two rectangular parallel plates of equal size but with different ratios of length to width, or aspect ratio [15].

Figure 5.23 shows view factors for two rectangular plates of different sizes that intersect at an angle of 90° [15].

The view factor f is not easily determined for radiation involving irregular shapes, unless the radiating surface (1) is completely enclosed by another surface (2), as shown in Figure 5.24. The view factor from surface 1 to surface 2 (f_{1-2}) is 1.0 for this condition.

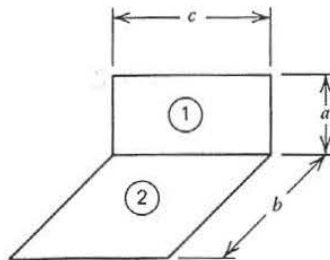
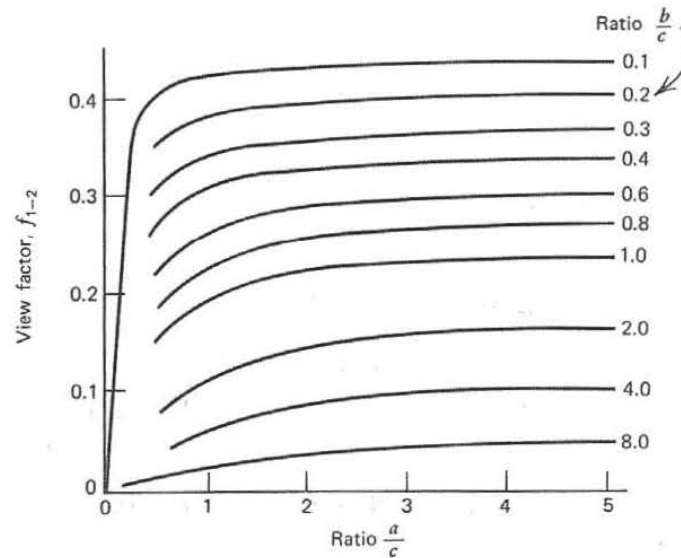


Figure 5.23 View factor for perpendicular and adjacent rectangles with a common side. (From Heat Transmission, by McAdams, McGraw Hill Book Co.)

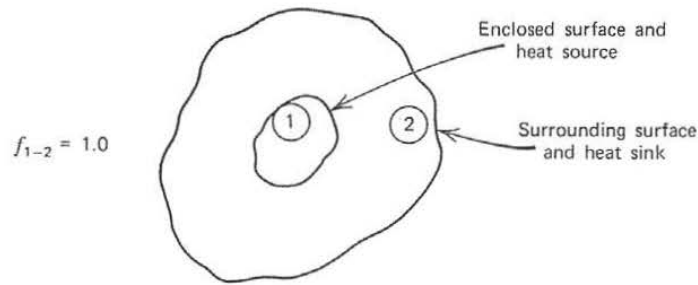


Figure 5.24 View factor for a completely enclosed irregular body.

When partial surfaces are involved in the radiation interchange, the view factor must be calculated or estimated. In electronic systems, the view factor from a transistor on a printed circuit board (PCB) to a cooler wall may be extremely complex and involve lengthy mathematics. If there are 50 transistors in different positions on different PCBs, the calculations may take several weeks. The amount of time required to perform these calculations may not be cost effective, so that the view factors for each transistor may simply be estimated.

Radiation heat transfer depends upon the line of sight. Therefore, it is important to determine approximately what fraction of the radiation leaving the surface of each transistor in all directions is intercepted by the cooler wall. This is what determines the view factor from the transistor to the wall.

The view factor from a hot transistor case on one PCB to the surface of an adjacent PCB is meaningless unless the adjacent PCB is much cooler than the transistor case. In general, there is very little heat lost by radiation between the hot components on adjacent PCBs. These hot components just "see" each other, so that there is a radiation interchange but no heat loss (see Figure 5.25).

Electronic components mounted on the end PCBs will usually have a good view of the end walls. Radiation heat transfer for these components will be relatively good if the end walls are cool.

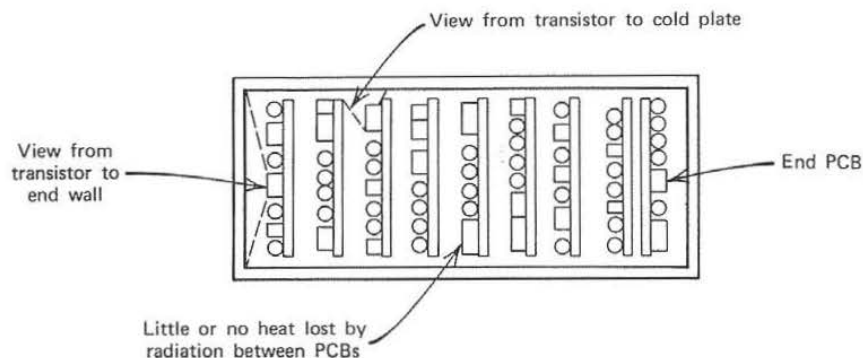


Figure 5.25 Radiation heat transfer within an electronic box.

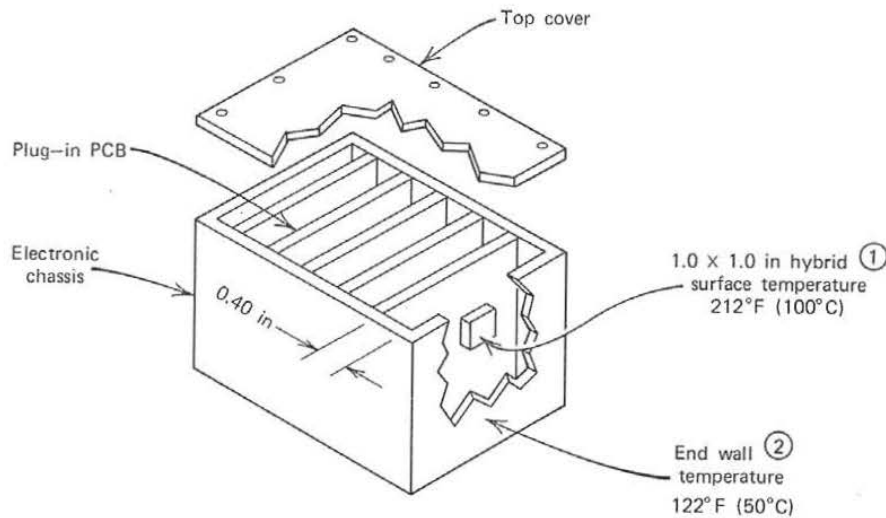


Figure 5.26 Hybrid mounted on a PCB that faces a chassis end wall.

5.18 SAMPLE PROBLEM—RADIATION HEAT TRANSFER FROM A HYBRID

A 1.0 in. square (2.54 cm) $\times$ 0.180 in. high (0.457 cm) flat pack hybrid is mounted on the end printed circuit board so that it faces the end wall of an electronic chassis, which has a temperature of 122°F (50°C), as shown in Figure 5.26. The hybrid is about 0.40 in (1.02 cm) from the wall, so that natural convection and conduction heat transfer will be negligible. The maximum allowable case temperature of the hybrid is 212°F (100°C). Determine the maximum allowable power dissipation for the hybrid with and without a conformal coating. The inside surfaces of the aluminum chassis are painted light blue.

SOLUTION

The top surface of the hybrid and its electrical lead wires have a good view of the chassis end wall. The view from the side walls of the hybrid to the side walls of the chassis is blocked by other electronic components around the hybrid. The bottom surface of the hybrid has no view of any wall. Ignoring radiation from the lead wires, the view factor from the top surface of the hybrid to the end wall will be very high, because the end wall is so close to the hybrid and the end wall is so much larger than the hybrid. The view factor f from the hybrid case to the chassis wall in this case would be about 0.95.

Equation 5.47 can be used to determine the amount of heat the hybrid is permitted to dissipate, in English and metric units.

$$\text{Given } \sigma = 0.1713 \times 10^{-8} \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R}^4 = 1.355 \times 10^{-12} \text{ cal/sec cm}^2 \text{ } ^\circ\text{K}^4$$

$$f = 0.95 \text{ estimated, based on geometry}$$

$$e_1 = 0.066 \text{ emissivity of bare hybrid steel case}$$

$$e_2 = 0.90 \text{ emissivity of painted aluminum wall}$$

$$e = \frac{1}{1/e_1 + 1/e_2 - 1} = \frac{1}{1/0.066 + 1/0.90 - 1} = 0.0655$$

(combined emissivity)

$$A_1 = 1.0 \text{ in}^2 = 0.00694 \text{ ft}^2 = 6.45 \text{ cm}^2 \text{ (area)}$$

$$t_1 = 212^\circ\text{F} = 100^\circ\text{C} \text{ (hybrid case temperature)}$$

$$t_2 = 122^\circ\text{F} = 50^\circ\text{C} \text{ (chassis wall temperature)}$$

Substitute into Eq. 5.47 for heat transferred from the hybrid by radiation, without a conformal coating, using English units.

$$Q = \left(0.1713 \times 10^{-8} \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{R}^4} \right) (0.95)(0.0655)(0.00694 \text{ ft}^2)$$

$$[(460 + 212)^4 - (460 + 122)^4] \text{ } ^\circ\text{R}^4$$

$$Q = (0.1713)(0.95)(0.0655)(0.00694) \left[\left(\frac{672}{100} \right)^4 - \left(\frac{582}{100} \right)^4 \right]$$

$$Q = 0.065 \frac{\text{Btu}}{\text{hr}} \times 0.293 \frac{\text{watt}}{\text{Btu/hr}} = 0.019 \text{ watt} \quad (5.58)$$

In metric units for the hybrid with no coating:

$$Q = \left(1.355 \times 10^{-12} \frac{\text{cal}}{\text{sec cm}^2 \text{ } ^\circ\text{K}^4} \right) (0.95)(0.0655)(6.45 \text{ cm}^2)$$

$$[(273 + 100)^4 - (273 + 50)^4] \text{ } ^\circ\text{K}^4$$

$$Q = (1.355)(0.95)(0.065)(6.45) \left[\left(\frac{373}{1000} \right)^4 - \left(\frac{323}{1000} \right)^4 \right]$$

$$Q = 0.00457 \frac{\text{cal}}{\text{sec}} \times 4.187 \frac{\text{watts sec}}{\text{cal}} = 0.019 \text{ watt} \quad (5.58a)$$

When a thin conformal coating is applied to the hybrid case, the emissivity will jump to about 0.80. The combined emissivity will jump to

$$e = \frac{1}{1/0.80 + 1/0.90 - 1} = 0.734$$

An examination of Eqs. 5.58 and 5.58a shows that the amount of heat transferred by radiation is directly proportional to the emissivity. Therefore,

a direct ratio of the combined emissivities can be used to determine the heat transferred when a conformal coating is added to the hybrid case.

$$Q = 0.019 \text{ watt} \left(\frac{e_{\text{coat}}}{e_{\text{no coat}}} \right) = 0.019 \left(\frac{0.734}{0.0655} \right)$$

$$Q = 0.213 \text{ watt} \quad (5.59)$$

This shows that just adding a conformal coating to the hybrid will increase its heat transfer capability by a factor of 11.

5.19 SAMPLE PROBLEM—JUNCTION TEMPERATURE OF A DUAL FET SWITCH

Figure 5.27 shows a dual field effect transistor (FET) switch in a TO-8 size case that dissipates 0.150 watt. The switch is mounted on a PCB within an electronic chassis that has painted walls at a temperature of 185°F (85°C). The maximum allowable junction temperature is 302°F (150°C) and the thermal impedance (Θ_{jc}) from the junction of the switch to the case is 49.2°F/watt (27.3°C/watt) as rated by the manufacturer. The case of the switch is made of polished iron and the view factor f from the case to the chassis walls is estimated to be about 0.80. The chassis is enclosed, so that natural convection and conduction heat transfer from the switch are negligible. Determine if the design is satisfactory.

SOLUTION

The following information is required for Eq. 5.47.

- $f = 0.80$ view factor, estimated from geometry
- $e_1 = 0.24$ emissivity of switch with bare iron case
- $e_2 = 0.84$ emissivity of painted chassis walls

$$e = \frac{1}{1/e_1 + 1/e_2 - 1} = \frac{1}{1/0.24 + 1/0.84 - 1} = 0.23$$

(combined emissivity)

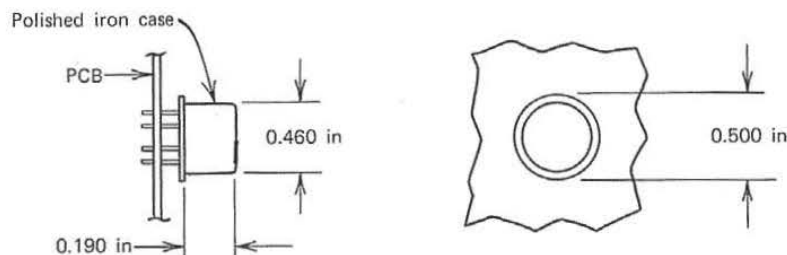


Figure 5.27 Dual FET switch mounted on a PCB.

The bottom surface of the switch faces the PCB, which is a poor heat conductor, so that its area is ignored. Also, radiation from the lead wires is ignored. The switch surface area is:

$$A = \frac{\pi(0.460)(0.190) + (\pi/4)(0.500)^2 \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 0.00327 \text{ ft}^2$$

$$Q = 0.150 \text{ watt} \times 3.413 \frac{\text{Btu/hr}}{\text{watt}} = 0.512 \frac{\text{Btu}}{\text{hr}} \text{ (heat dissipation)}$$

Substitute into Eq. 5.47 for the case temperature.

$$0.512 = 0.1713 \times 10^{-8} (0.8)(0.23)(0.00327)[(460 + t_1)^4 - (460 + 185)^4]$$

$$0.512 = 1.031 \times 10^{-12} (460 + t_1)^4 - 0.1784$$

$$460 + t_1 = \left(\frac{0.512 + 0.1784}{1.031 \times 10^{-12}} \right)^{1/4}$$

$$t_1 = 444.6^\circ\text{F} (229.2^\circ\text{C}) \text{ (case temperature)} \quad (5.60)$$

The temperature rise from the switch case to the junction is determined from the manufacturer's thermal impedance from junction to case (Θ_{jc}), 49.2°F/watt .

$$\Delta t_{jc} = 49.2 \frac{^\circ\text{F}}{\text{watt}} \times 0.150 \text{ watt} = 7.4^\circ\text{F} \quad (5.61)$$

The junction temperature becomes

$$t_j = 444.6 + 7.4 = 452^\circ\text{F} (233.3^\circ\text{C}) \quad (5.62)$$

This is substantially above the maximum allowable junction temperature of $302^\circ\text{F} (150^\circ\text{C})$, so that the design is not satisfactory.

To improve the radiation heat transfer, a black oxide finish is added to the surface of the switch, which raises its emissivity to a value of 0.95. The combined emissivity is now

$$e = \frac{1}{1/0.95 + 1/0.84 - 1} = 0.804 \text{ (combined emissivity)}$$

Substitute back into Eq. 5.47 for the case temperature.

$$0.512 = 0.1713 \times 10^{-8} (0.8)(0.804)(0.00327)[(460 + t_1)^4 - (460 + 185)^4]$$

$$460 + t_1 = \left(\frac{0.512 + 0.623}{3.60 \times 10^{-12}} \right)^{1/4}$$

$$t_1 = 289^\circ\text{F} (142.7^\circ\text{C}) \text{ (case temperature)} \quad (5.63)$$

The thermal impedance Θ_{jc} from the junction to the case is still the same. The junction temperature of the switch with the black case is shown as follows:

$$t_j = 289 + 7.4 = 296.4^\circ\text{F} \ (146.9^\circ\text{C}) \quad (5.64)$$

Since the junction temperature is less than 302°F (150°C), the design is satisfactory.

5.20 RADIATION HEAT TRANSFER IN SPACE

In the hard vacuum environment of space, the amount of radiant energy that a body receives will depend upon its distance from the sun, the view the body has of the sun, and its proximity to other planets or satellites. A spacecraft in close orbit around the earth or around the moon can receive radiation from the sun, the earth, and the moon at the same time. Most of this radiation is in the ultraviolet and visible light frequency range, which has a wavelength up to about $1\ \mu\text{m}$, which is $1 \times 10^{-6}\ \text{m}$, as shown in Figure 5.15. In a near earth orbit the direct radiation intensity is about $444\ \text{Btu/hr ft}^2$ ($130\ \text{watts/ft}^2$) ($0.0334\ \text{cal/sec cm}^2$) [4, 5].

In addition to the direct solar radiation energy, there is reflected radiation energy, or albedo. Albedo is defined as the fraction of solar radiation that is returned to space due to reflections of solar energy from the atmosphere, clouds, and surface of a planet. For the earth, the albedo is about 0.40, depending upon the amount of cloud cover in the sky. For the moon, the albedo is very low, about 0.07, because the moon has virtually no atmosphere.

When the earth is viewed from a distance, its average temperature appears to be about -20°F (-29°C). The mean surface temperature appears to have a value of about 57°F (14°C), and the mean temperature of the upper atmosphere, at about 150,000 ft, is about -85°F (-65°C). The sun, when viewed from a distance, appears to have a temperature of about $10,400^\circ\text{F}$ (5760°C).

The amount of radiant energy that a body in space will receive will depend upon the proximity and the view the body has of the sun and, or the moon, or perhaps some other planet. Table 5.7 shows how much radiant energy various planets receive from the sun. This is the energy that reaches the planet's upper atmosphere, not the energy that reaches the surface (except for the moon).

The upper atmosphere of the earth attenuates the radiation energy received from the sun, and this reduces the amount of solar radiation that actually reaches the surface of the earth. The amount of radiation energy that actually reaches the surface of the earth depends upon the month, hour, parallel of latitude, and cloud cover. Some typical maximum values

Table 5.7 Average Radiation Characteristics of the Planets

Planet	Incident Radiation Intensity (Btu/hr ft ²)	Reflected Radiation (Albedo)	Equivalent Black Body Temperature (°F)
Mercury	2922	0.06	342
Venus	860	0.61	127
Earth	444	0.40	-5
(Moon)	434	0.072	51
Mars	192	0.15	-56
Jupiter	16.4	0.41	-242
Saturn	4.9	0.42	-297
Uranus	1.2	0.45	-346
Neptune	0.5	0.54	-366
Pluto	0.3	0.15	-380

Night sky = -40°F

on a clear day are as follows: [37, 38]:

40° north latitude USA: 300 Btu/hr ft²

32° north latitude USA: 310 Btu/hr ft²

24° north latitude USA: 320 Btu/hr ft²

Deep space has the ability to absorb virtually an infinite amount of heat that makes it a very useful heat sink with a temperature of -460°F (-273°C), absolute zero. The view factor of a body in deep space is 1.0, because the body can radiate to deep space in all directions.

If two aluminum spheres are placed in deep space and if one sphere has a polished aluminum surface while the other sphere has a surface that is painted white, the white sphere will show a steady state surface temperature of about 80°F (26.6°C) and the polished sphere will show a surface temperature of about 450°F (232°C), as shown in Figure 5.28.

5.21 EFFECTS OF α/e ON TEMPERATURES IN SPACE

When thermal radiation energy strikes a body, the fraction of the incident radiation energy that is absorbed is called the absorptivity α . In the infrared, or long wavelength range, the color has no effect on the absorptivity, and a black surface will absorb about the same amount of thermal radiation energy as a white surface. In the visible, or short wavelength range (see Figure 5.15), the amount of radiation energy absorbed by a body is sharply

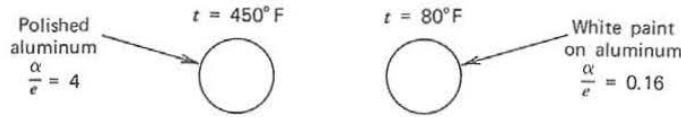


Figure 5.28 Temperatures of painted and unpainted aluminum spheres in space.

affected by its color. The stabilized temperature a body reaches in space will therefore be related to the ratio of the solar absorptivity (α_s) to the infrared emissivity (e), which is shown as the ratio α_s/e . For the polished aluminum sphere this ratio is shown in Eq. 5.65 and for the white-painted aluminum sphere, this ratio is shown in Eq. 5.66.

Polished aluminum sphere in space:

$$\frac{\alpha_s}{e} = \frac{\text{solar absorptivity}}{\text{infrared emissivity}} = \frac{0.10}{0.025} = 4 \quad (5.65)$$

White-painted aluminum sphere in space:

$$\frac{\alpha_s}{e} = \frac{\text{solar absorptivity}}{\text{infrared emissivity}} = \frac{0.15}{0.94} = 0.16 \quad (5.66)$$

A high ratio of α/e means that more solar radiation energy will be absorbed, so that the resulting temperature of a body in space will be higher. Conversely, a low α/e means that less solar radiation energy will be absorbed, so that the resulting temperature will be lower. The equilibrium temperature of a body in space can be controlled, to a great extent, by controlling the ratio of α_s/e .

A body in space will receive radiation from the sun and emit radiation to space at -460°F , which is absolute zero. If the amount of solar energy absorbed by a body (including albedo) is shown as $\alpha_s Q_s A_s$, the general radiation relation shown by Eq. 5.47 can be modified as shown by Eq. 5.67.

$$Q_i + \alpha_s Q_s A_s = \sigma e A T^4 \quad (5.67)$$

where α_s = absorptivity of surface receiving solar radiation
 Q_s = solar irradiation on absorbing surface
 Q_i = internal power dissipation
 A_s = projected area of body normal to sun
 σ = Stefan-Boltzmann constant
 e = emissivity of surface radiating to space
 A = surface area of body radiating to space
 T = absolute temperature of body in space

When the body in space is away from all the planets, with no internal power

dissipation and no heat transferred into or out of the back side of the body, and when the projected area of the body is normal to the sun. The absolute temperature of the body in space is shown by Eq. 5.68.

$$T = \left(\frac{\alpha_s Q_s}{\sigma e} \right)^{0.25} \quad (5.68)$$

In a deep space orbit similar to that of Venus, but away from all planets, the direct solar radiation intensity Q_s is about 860 Btu/hr ft² (252 watts) (0.0648 cal/sec cm²). Substitute this value into Eq. 5.68 to obtain the relation for the absolute temperature of a body in a deep space orbit similar to that of Venus.

$$T = \left(\frac{860 \alpha_s}{0.1713 \times 10^{-8} e} \right)^{0.25} = 842 \left(\frac{\alpha_s}{e} \right)^{0.25} \quad (5.69)$$

Since the solar absorptivity (α_s) and the infrared emissivity (e) are both characteristics of the body surface, the control of these values will help to control the body temperature.

5.22 SAMPLE PROBLEM—TEMPERATURES OF AN ELECTRONIC BOX IN SPACE

An electronic box has an insulated panel with a surface area of 1.0 ft². It is in deep space approximately the same distance from the sun as the earth. The panel is painted white and always faces the sun. Determine the stabilized temperature of the panel with no internal power dissipation and with an internal power dissipation of 50 watts.

SOLUTION

Given $Q_s = 444$ Btu/hr ft² solar radiation on panel in an orbit similar to that of Earth

$\alpha_s = 0.15$ solar absorptivity of white paint (ref. Table 5.4)

$\sigma = 0.1713 \times 10^{-8}$ Btu/hr ft² °R⁴ constant

$e = 0.9$ emissivity of white paint (ref. Table 5.4)

Substitute into Eq. 5.68 for the stabilized temperature of the panel with no internal power dissipation.

$$T = \left[\frac{(0.15)(444)}{(0.1713 \times 10^{-8})(0.9)} \right]^{0.25} = 455.9^\circ\text{R}$$

$$t = 455.9 - 460 = -4.1^\circ\text{F} \quad (5.70)$$

The stabilized temperature of the panel with power on can be obtained with Eqs. 5.67 and 5.68, which requires adding the internal power of 50 watts (170.6 Btu/hr).

$$T = \left[\frac{(0.15)(444) + 170.6}{(0.1713 \times 10^{-8})(0.9)} \right]^{0.25} = 626.3^{\circ}\text{R}$$

$$t = 626.3 - 460 = 166.3^{\circ}\text{F} \quad (5.71)$$

If the electronic box was in an orbit around the earth, where it was possible for the panel to receive reflected albedo radiation (40%) from the earth, the panel would receive an additional incident radiation of 444×0.40 or 177.6 Btu/hr ft. Substituting into Eq. 5.68 for the panel temperature, we obtain

$$T = \left[\frac{(0.15)(444 + 177.6) + 170.6}{(0.1713 \times 10^{-8})(0.9)} \right]^{0.25} = 643.2^{\circ}\text{R}$$

$$t = 643.2 - 460 = 183^{\circ}\text{F} \quad (5.72)$$

There would also be an interchange of radiation energy between the electronic box and the earth, which would depend upon the orbit of the electronic box around the planet. This energy interchange was ignored in the problem.

5.23 SIMPLIFIED RADIATION HEAT TRANSFER EQUATION

It is often convenient to express the radiation heat loss in a form similar to the convection heat loss, as shown by Eq. 5.7. This can be accomplished by setting Eq. 5.7 equal to Eq. 5.47. The radiation heat transfer equation is then simplified as shown by Eq. (5.73) [15, 17, 24],

$$Q = h_r A \Delta t \quad (5.73)$$

In this expression, h_r represents the radiation heat transfer coefficient in terms of Btu/hr ft² °F (cal/sec cm² °C). The radiation coefficient is then defined as shown in Eq. 5.74 for English units and in Eq. 5.74a for metric units.

$$h_r = \frac{0.1713 fe \{[(t_1 + 460)/100]^4 - [(t_2 + 460)/100]^4\}}{t_1 - t_2}$$

$$= \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^{\circ}\text{F}} \quad (5.74)$$

$$h_r = \frac{1.355 f e \{[(t_1 + 273)/1000]^4 - [(t_2 + 273)/1000]^4\}}{t_1 - t_2}$$

$$= \frac{\text{cal}}{\text{sec cm}^2 \text{ } ^\circ\text{C}} \quad (5.74a)$$

A plot of Eq. 5.74 is shown in Figure 5.29 for various heat receiver temperatures and various temperature differences between the emitting and receiving surfaces [70].

5.24 SAMPLE PROBLEM—RADIATION HEAT LOSS FROM AN ELECTRONIC BOX

An electronic box, for a space application, has hybrid circuits mounted on the cover facing the inside of the box. The backside of the cover can “see” the back side of a space radiator panel within the spacecraft, as shown in Figure 5.30. The maximum allowable hybrid case temperature is 212°F (100°C). The temperature rise from the component case to the cover is 12°F (6.7°C), which means that the maximum allowable cover temperature is 200°F (93.3°C). The radiator panel has a temperature of 20°F (−6.7°C) and an emissivity e_2 of 0.75. The electronic box cover has a painted finish with an emissivity e_1 of 0.85. Determine the maximum allowable power the electronic components on the cover can dissipate.

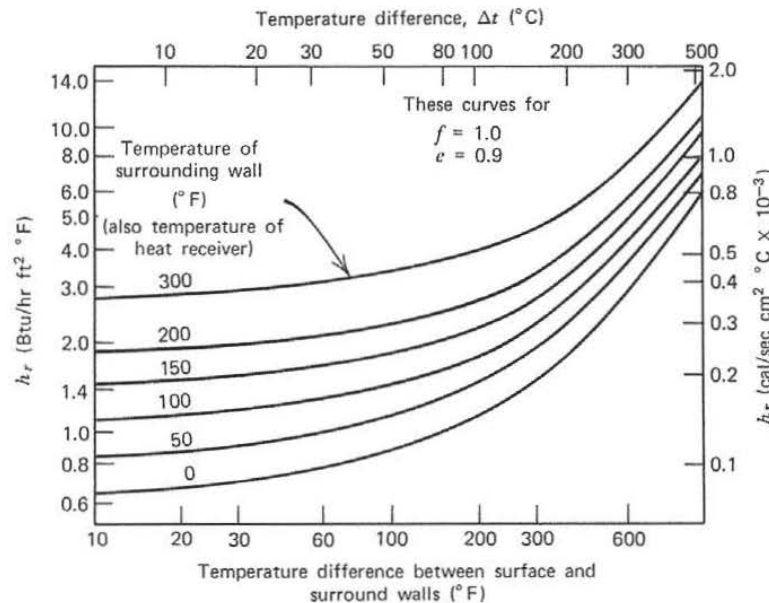


Figure 5.29 Radiation heat transfer as a function of temperature. (From General Electric Co. Heat Transfer/Fluid Flow Data Book)

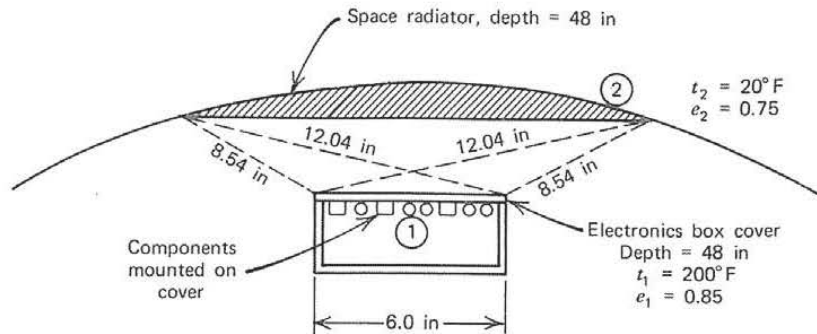


Figure 5.30 Electronic box mounted in a spacecraft.

SOLUTION

The cross string method shown in Eq. 5.49 is used to determine the view factors between the radiating surfaces.

$$A_1 f_{1-2} = A_2 f_{2-1} = \frac{2(12.04) - 2(8.54)}{2} = 3.5 \frac{\text{in}^2}{\text{in depth}}$$

$$f_{1-2} = \frac{A_2 f_{2-1}}{A_1} = \frac{3.5 \text{ in}^2/\text{in}}{(6)(48) \text{ in}^2} \times 48 \text{ in} = 0.583$$

$$f_{2-1} = \frac{A_1 f_{1-2}}{A_2} = \frac{3.5}{(12)(48)} \times 48 \text{ in} = 0.292$$

The combined emissivity is obtained from Table 5.5.

$$e = \frac{1}{1/e_1 + 1/e_2 - 1} = \frac{1}{1/0.85 + 1/0.75 - 1} = 0.662$$

The view factor from area 1 to area 2 ($f_{1-2} = 0.583$) will be used for the radiation heat transfer, so area 1 must be computed.

$$A_1 = \frac{(6)(48) \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 2.0 \text{ ft}^2 = 1858 \text{ cm}^2 \quad (5.75)$$

Substitute into Eq. 5.74 to obtain the radiation coefficient in English Units.

$$h_r = \frac{0.1713(0.583)(0.662) \{[(200 + 460)/100]^4 - [(20 + 460)/100]^4\}}{200 - 20}$$

$$h_r = 0.502 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} \quad (5.76)$$

Substitute into Eq. 5.74a to obtain the radiation coefficient in metric units.

$$h_r = \frac{1.355(0.583)(0.662)\{[(93.33 + 273)/1000]^4 - [(-6.7 + 273)/1000]^4\}}{93.3 - (-6.7)}$$

$$h_r = 0.0000679 \frac{\text{cal}}{\text{sec cm}^2 \text{ } ^\circ\text{C}} \quad (5.76a)$$

Substitute Eqs. 5.75 and 5.76 into Eq. 5.73 for the maximum heat that can be transferred by radiation, using English units.

$$Q = \left(0.502 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}}\right) (2.0 \text{ ft}^2)(200 - 20)^\circ\text{F}$$

$$Q = 180.7 \frac{\text{Btu}}{\text{hr}} \times 0.293 \frac{\text{watt}}{\text{Btu/hr}} = 52.9 \text{ watts} \quad (5.77)$$

Substitute into Eq. 5.73 for the maximum heat that can be transferred by radiation using metric units.

$$Q = \left(0.0000679 \frac{\text{cal}}{\text{sec cm}^2 \text{ } ^\circ\text{C}}\right) (1858 \text{ cm}^2)[(93.3 - (-6.7))]$$

$$Q = 12.61 \frac{\text{cal}}{\text{sec}} \times 4.187 \frac{\text{watts}}{\text{cal/sec}} = 52.8 \text{ watts} \quad (5.77a)$$

The heat transfer coefficient h_r could also have been obtained from Figure 5.29 for English units and metric units. Notice that Figure 5.29 is based upon a view factor of 1 and a combined emissivity of 0.9. If the geometry under consideration has a different view factor and a different emissivity, corrections must be applied to obtain the true radiation coefficient.

For the previous problem, using a heat sink temperature of 20°F and a ΔT of $200 - 20 = 180^\circ\text{F}$, $h_r = 1.18 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F}$ for English units. Corrections must be applied for view factors and emissivity.

$$h_r = 1.18 \times 0.583 \times \left(\frac{0.662}{0.9}\right) = 0.506 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} \quad (5.78)$$

Equation 5.78 compares well with Eq. 5.76.

5.25 COMBINING CONVECTION AND RADIATION HEAT TRANSFER

In many electronic boxes, radiation heat transfer and convection (both natural and forced) occur simultaneously. Therefore, it is convenient to write the combined heat transfer equation in a form similar to the general

relation shown by Eqs. 5.7 and 5.73. The combined relation is shown in Eq. 5.79 [15, 17, 24].

$$Q = (h_c + h_r)A \Delta t \quad (5.79)$$

where h_c = Btu/hr ft² °F or cal/sec cm² °C (convection coefficient)

h_r = Btu/hr ft² °F or cal/sec cm² °C (radiation coefficient)

A = ft² or cm² (area)

Δt = °F or °C (temperature difference, surface to ambient)

For most small electronic boxes, typical values of h_c are about 0.8 Btu/hr ft² °F (0.000108 cal/sec cm² °C) and typical values of h_r are about 1.2 Btu/hr ft² °F (0.000162 cal/sec cm² °C), which results in a combined heat transfer coefficient, $h_c + h_r$, of about 2.0 Btu/hr ft² °F (0.00027 cal/sec cm² °C).

5.26 SAMPLE PROBLEM—ELECTRONIC BOX IN AN AIRPLANE COCKPIT AREA

An electronic box is mounted in the cockpit area of an airplane that must operate at 50,000 ft without a pressurized cabin. Plug-in PCBs within the box are cooled by conducting their heat to the side walls of the electronic box. Natural convection and radiation are used to cool the external surfaces of the box. The maximum expected cockpit temperature is 100°F and the maximum allowable box surface temperature is 140°F. Because the top and bottom covers are fastened to the box with a few small screws, the covers make poor contact with the box and will not be effective heat transfer surfaces (see Figure 5.31). Determine the maximum amount of heat that can

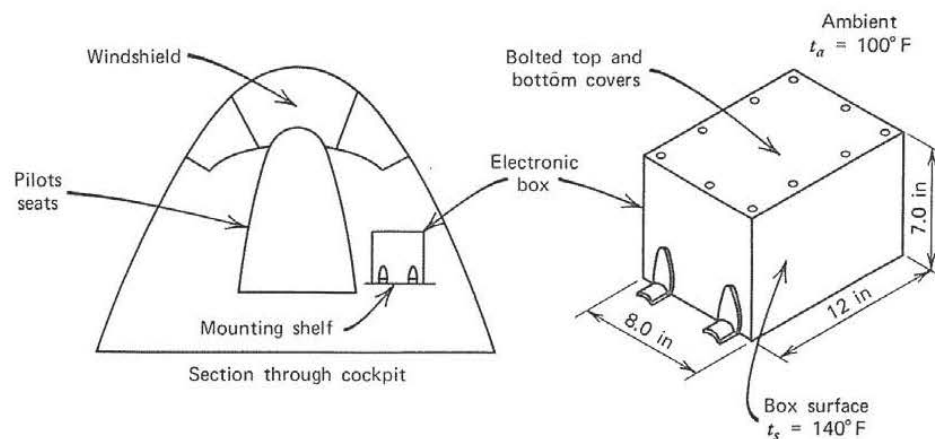


Figure 5.31 Electronic box mounted on a shelf in an airplane.

be transferred from the box to the surrounding ambient at sea level and at 50,000 ft. The box has painted external surfaces.

SOLUTION

Only the vertical side and end walls will be effective for heat transfer by natural convection and radiation. The top and bottom covers are ignored here because the covers make poor contact with the chassis. The natural convection coefficient for the vertical side and end walls is determined first for the sea level condition.

$$h_c = 0.29 \left(\frac{\Delta t}{L} \right)^{0.25} \quad (\text{ref. Eq. 5.4})$$

Given $\Delta t = 140 - 100 = 40^\circ\text{F}$ (temperature difference)

$$L = \frac{7.0 \text{ in}}{12 \text{ in/ft}} = 0.583 \text{ ft (vertical height)}$$

$$h_c = 0.29 \left(\frac{40}{0.583} \right)^{0.25} = 0.83 \frac{\text{Btu}}{\text{hr ft}^2 ^\circ\text{F}} \text{ (convection coefficient)} \quad (5.80)$$

The radiation heat transfer coefficient is determined next, using an emissivity of 0.9 for the painted finish and a view factor of 1.0 for the electronic box. Using Figure 5.29 with an ambient heat sink temperature of 100°F and a Δt of 40°F ,

$$h_r = 1.20 \frac{\text{Btu}}{\text{hr ft}^2 ^\circ\text{F}} \text{ (radiation coefficient)} \quad (5.81)$$

The surface area available for heat transfer is the sides and ends of the box only.

$$A = \frac{(2 \times 12 + 2 \times 8)(7) \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 1.94 \text{ ft}^2 \quad (5.82)$$

Substitute Eqs. 5.80 through 5.82 into Eq. 5.79 for the heat transferred at sea level conditions.

$$\begin{aligned} Q &= (0.83 + 1.20)(1.94)(40) \\ Q &= 157.5 \frac{\text{Btu}}{\text{hr}} \times 0.293 \frac{\text{watt}}{\text{Btu/hr}} = 46.1 \text{ watts} \end{aligned} \quad (5.83)$$

At an altitude of 50,000 ft, there will be a small change in the radiation characteristics of the box, but there will be a big change in the natural

convection characteristics. (The surface temperature of the box will increase slightly at altitude.)

$$h_{\text{alt}} = h_c \sqrt{\frac{\text{altitude pressure}}{14.7 \text{ psia}}} \quad (\text{ref. Eq. 5.41})$$

Using the pressure altitude from Table 6.5, the pressure at 50,000 ft is 1.692 lb/in².

$$h_{\text{alt}} = 0.83 \sqrt{\frac{1.69 \text{ psia}}{14.7 \text{ psia}}} = 0.28 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} \quad (5.84)$$

Equation 5.79 is used once again to determine the amount of heat that can be transferred at the 50,000 ft altitude.

$$\begin{aligned} Q &= (h_{\text{alt}} + h_r) A \Delta T \quad (\text{ref. Eq. 5.79}) \\ Q &= (0.28 + 1.20)(1.94)(40) \\ Q &= 114.8 \frac{\text{Btu}}{\text{hr}} \times 0.293 \frac{\text{watt}}{\text{Btu/hr}} = 33.6 \text{ watts} \end{aligned} \quad (5.85)$$

This shows that the amount of heat that can be removed at the 50,000 ft altitude has been reduced about 27%.

5.27 EQUIVALENT AMBIENT TEMPERATURE FOR RELIABILITY PREDICTIONS

Reliability failure rates for electronic component parts are usually shown to be related to the *ambient* temperature in which the component is operating. MIL-HDBK-217B, for example, shows failure rates for various components as a function of the ambient temperature [39].

In many electronic systems, the electronic components are mounted on thermally conductive circuit boards, which spread the heat to eliminate hot spots and also carry the heat to cold plates and heat exchangers. A low thermal resistance path is provided from the component surface directly to the chassis heat sink. This technique is very effective in providing a low component *surface* temperature.

The reliability engineer is not interested in component surface temperatures, because there are very few data relating surface temperatures of components to failure rates. Most data will show that component failure rates are related to the ambient temperature. Therefore, reliability engineers will usually request information relating to ambient temperatures when they are computing failure rates for a system.

When components are cooled by conduction, there are no real ambient operating temperatures that can be used to predict failure rates. However, an equivalent ambient temperature can be calculated based upon the surface temperature of the component. Thus, for a given set of conditions and for specific components, the equivalent ambient temperature can be determined from the component surface temperature. This permits failure rates to be determined for components when only the surface temperatures are known.

Since heat travels from high temperature areas to low temperature areas, it follows that a heat dissipating component will be hotter than the ambient temperature surrounding the component.

When an electronic component is suspended in a controlled environmental chamber, heat is transferred from the component to the surrounding ambient by means of radiation and natural convection. Component manufacturers and component testing engineers often use this technique to determine how much power a component can dissipate and what the failure rates are for various ambient temperature conditions.

The surface temperature of a component on a PCB is usually known from test data, or it can be calculated. The equivalent ambient temperature for the same component (the ambient temperature that will result in the same component surface temperature) is determined by obtaining the temperature rise from the component to the surrounding ambient using Eq. 5.86.

$$t_s = t_a + \Delta t_{s-A} \quad (5.86)$$

where t_s = component surface temperature

t_a = ambient temperature

Δt_{s-A} = temperature rise from surface to ambient

The temperature rise from the component surface to the ambient Δt_{s-A} is determined from Eq. 5.79.

$$\Delta t_{s-A} = \frac{Q}{(h_c + h_r)A} \quad (\text{ref. Eq. 5.79})$$

The physical properties of the component are used to determine the surface area, the radiation coefficient, and the natural convection coefficient. The electrical lead wires are ignored in these calculations because they are usually very short in most applications, so that they will transfer very little heat by radiation and natural convection.

Radiation depends upon the actual surface temperature of the component, which is unknown. Therefore, the method of solution is to assume a temperature rise (Δt) that permits the radiation and natural convection coefficients to be determined. A Δt value is then calculated from the coefficients and compared with the assumed value. If the Δt 's do not agree, a correction is made and the process is repeated until the assumed Δt agrees relatively

well with the calculated Δt . A sample problem is used to demonstrate the technique.

5.28 SAMPLE PROBLEM—EQUIVALENT AMBIENT TEMPERATURE OF AN RC07 RESISTOR

A MIL style RC07 $\frac{1}{4}$ watt carbon composition resistor, as shown in Figure 5.32, dissipates 0.125 watt when it is mounted on a PCB that is cooled by conduction. A small thermocouple on the resistor shows that the surface temperature is 212°F (100°C) in the operating environment. The reliability engineer must determine the mean time between failures (MTBF) for this resistor. The only failure rate data are based on ambient temperatures. Determine the equivalent *ambient* temperature that will produce the same failure rate as a component *surface* temperature of 212°F (100°C).

SOLUTION

The lead wires are small compared to the body of the resistor, so that they are ignored. Assume that the resistor body is suspended in a temperature-controlled chamber. The natural convection coefficient can then be determined from Eq. 5.3 and Table 5.1, using a C value of 1.39 for small components in free air.

$$h_c = 0.52C \left(\frac{\Delta t}{L} \right)^{0.25} \quad (\text{ref. Eq. 5.3})$$

$$h_c = 0.52(1.39) \left(\frac{\Delta t}{L} \right)^{0.25} = 0.72 \left(\frac{\Delta t}{L} \right)^{0.25} \quad (5.87)$$

Since the Δt is not known yet, assume a value of 67°F to start, and calculate the natural convection coefficient from Eq. 5.87.

Given $\Delta t = 67^\circ\text{F}$ (assumed to start)

$$L = \frac{0.098 \text{ in}}{12 \text{ in/ft}} = 0.0082 \text{ ft diameter}$$

$$h_c = 0.72 \left(\frac{67}{0.0082} \right)^{0.25} = 6.84 \frac{\text{Btu}}{\text{hr ft}^2 ^\circ\text{F}} \quad (5.88)$$

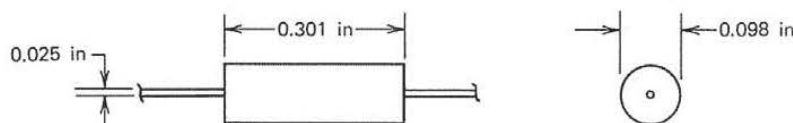


Figure 5.32 MIL style RC 07 carbon composition $\frac{1}{4}$ watt resistor.

The radiation coefficient h_r is determined from Figure 5.29, which is based upon an emissivity of 0.9 and a view factor of 1.0. The view factor of the resistor in the chamber will be 1.0 and the emissivity e of the resistor surface is about 0.9, so that the curves in Figure 5.29 can be used directly with no correction factors. The temperature of the surrounding walls is the same as the ambient temperature t_a , which is determined from Eq. 5.86.

$$\begin{aligned} \text{Given } t_s &= 212^\circ\text{F (surface temperature)} \\ \Delta t_{s-A} &= 67^\circ\text{F (assumed to start)} \\ t_a &= t_s - \Delta t_{s-A} = 212 - 67 = 145^\circ\text{F (ambient)} \end{aligned}$$

From Figure 5.29:

$$h_r = 1.61 \frac{\text{Btu}}{\text{hr ft}^2 ^\circ\text{F}} \quad (5.89)$$

Also,

$$\begin{aligned} Q &= 0.125 \text{ watts} \times \frac{3.413 \text{ Btu/hr}}{\text{watt}} = 0.427 \frac{\text{Btu}}{\text{hr}} \text{ (heat)} \\ A &= \frac{(2)(\pi/4)(0.098)^2 + \pi(0.098)(0.301) \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 0.000748 \text{ ft}^2 \text{ (area)} \end{aligned}$$

Substitute into Eq. 5.79 and solve for Δt .

$$\Delta t = \frac{Q}{(h_c + h_r) A} = \frac{0.427}{(6.84 + 1.61)(0.000748)} = 67.5^\circ\text{F} \quad (5.90)$$

Since the calculated Δt of 67.5°F is close to the assumed value of 67°F , the assumed Δt is valid.

The equivalent ambient temperature that can be used for reliability predictions is determined from Eq. 5.86.

$$t_a = t_s - \Delta t_{s-A} = 212 - 67 = 145^\circ\text{F} \quad (5.91)$$

This means that the RC07 resistor will have a surface temperature of 212°F in an ambient temperature of 145°F when the resistor dissipates 0.125 watt. Reliability failure rate predictions will be based upon operation in a 145°F ambient environment.

5.29 INCREASE IN EFFECTIVE EMITTANCE ON EXTENDED SURFACES

Extended surfaces, such as fins, are often used to improve the heat transfer characteristics of electronic equipment. A substantial improvement is ob-

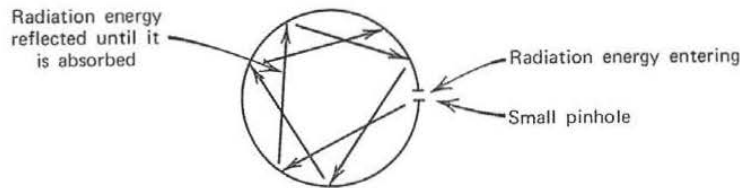


Figure 5.33 Black body represented by a hollow sphere with a pin hole.

tained with convection heat transfer, because the heat removed is directly related to the fin surface area. With radiation, the heat transfer is related to the projected fin area and the emittance of the surface.

A rough surface has a higher emittance than the same surface when it is smooth. Also, a finned surface has a higher emittance than does the same surface without fins. In both cases the emittance increases because the surfaces have many small cavities. These cavities act somewhat like many small partial black bodies [24].

A perfect black body is one that absorbs 100% of all the energy it receives without reflecting any energy. A small hole in a hollow sphere is often used to represent a black body. The energy enters the small opening and strikes the opposite wall, where part of the energy is absorbed and part is reflected. The reflected energy again strikes the opposite wall, where part of the energy is absorbed and again part is reflected. This process continues until all of the energy is absorbed, as shown in Figure 5.33.

Since finned surfaces act somewhat like cavities, a deeper fin should have a higher emittance than a shallow fin. When the emittance of the flat surface is known, the effective emittance of the finned surface can be approximated from Figure 5.34.

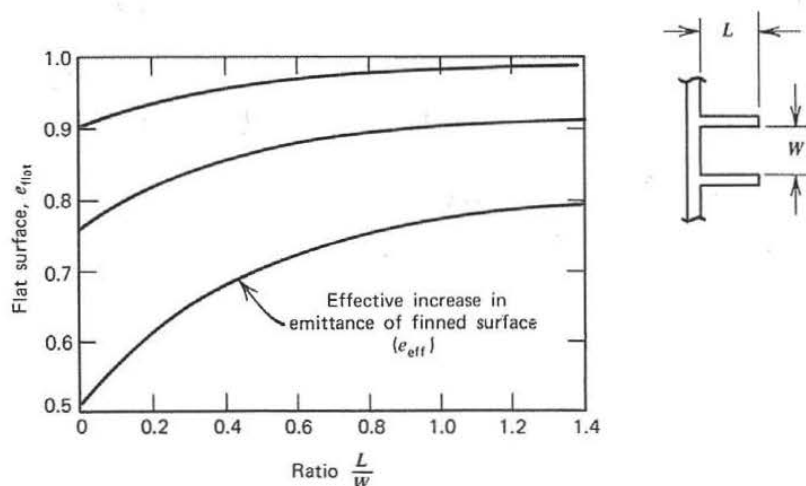


Figure 5.34 Increase in the effective emittance of a finned surface. (From Handbook of Electronic Packaging, by Harper, McGraw Hill Book Co.)

The height of the fin is shown by L and the distance between the fins is shown by W . Dimensions can be in inches or centimeters.

Figure 5.34 shows that the effective emittance of the finned surface can be substantially increased for low emittance surfaces, when the ratio of L/W is greater than 1.0, as shown by Eq. 5.92.

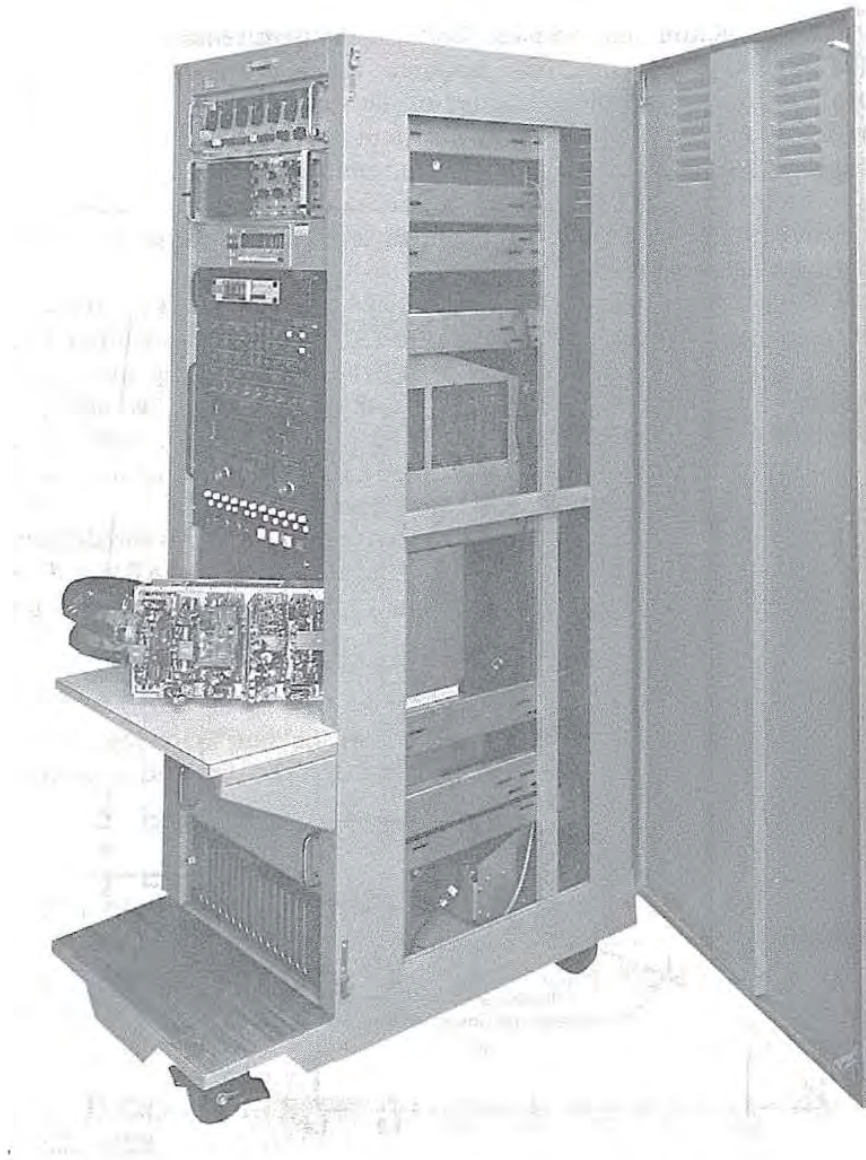


Figure 5.35 Tall single bay cabinet, with a blower at the bottom, for cooling panel and drawer mounted electronics. (Courtesy Litton Systems, Inc.)

$$e_{\text{eff}} = e_{\text{flat}} + \left(\frac{1 - e_{\text{flat}}}{2} \right) \quad (5.92)$$

When e_{flat} is 0.5, then for a L/W ratio of 1.0 the effective emissivity e_{eff} becomes

$$e_{\text{eff}} = 0.5 + \frac{1 - 0.5}{2} = 0.5 + 0.25 = 0.75 \text{ (approximately)}$$

Figure 5.35 shows a tall cabinet that is cooled by means of a blower in the base.

Forced Air Cooling for Electronics

6.1 FORCED COOLING METHODS

Heat transfer by forced convection generally makes use of a fan, blower, or pump to provide high velocity fluid (air or liquid) past a heated surface. The high velocity fluid results in a decreased thermal resistance across the boundary layer from the fluid to the heated surface. This, in turn, increases the amount of heat that is carried away by the fluid [40–42].

Forced air systems can provide heat transfer rates in electronic systems that are 10 times greater than those available with natural convection and radiation. Forced liquid cooling systems can provide heat transfer rates that are 10 times greater than those for forced air cooling systems. The penalty for increased cooling is increased costs, power, noise, and complexity. For liquid cooled systems the size and weight may also increase. Since a simple system is generally a more reliable system, heat transfer by natural convection and radiation should be used wherever possible.

The rewards of forced convection cooling techniques are generally reduced size for air cooled systems and higher component densities, with lower hot spot temperatures. This increases the electronic component reliability but requires added maintenance for the extra fans or pumps.

A liquid cooled system is usually larger and heavier than an air cooled system, because a reservoir is generally required for the liquid. A fan cooled system, on the other hand, normally has a large supply of air readily available with no storage requirements, which reduces the size and weight of the system.

Laminar flow conditions as well as turbulent flow conditions can exist with forced convection in liquids and air. Turbulent flow conditions are much more desirable, because they permit more heat to be removed. However, turbulent flow will usually result in a higher pressure drop through the system, which requires that larger pumps and fans and more power be used to overcome the added resistance.

6.2 COOLING AIR FLOW DIRECTION FOR FANS

When a fan is used for cooling electronic equipment, the air flow direction can be quite important. The fan can be used to draw air through a box or to blow air through a box. A blowing fan system will raise the internal air pressure within the box, which will help to keep dust and dirt out of a box that is not well sealed. A blowing system will also produce slightly more turbulence, which will improve the heat transfer characteristics within the box. However, when an axial flow fan is used in a blowing system, the air may be forced to pass over the hot fan motor, which will tend to heat the air as it enters the electronic box, as shown in Figure 6.1.

An exhaust fan system, which draws air through an electronic box, will reduce the internal air pressure within the box. If the box is located in a dusty or dirty area, the dust and dirt will be pulled into the box through all of the small air gaps if the box is not sealed. In an exhaust system, the cooling air passes through an axial flow fan as the air exits from the box, as shown in Figure 6.2. The cooling air entering the electronic box is therefore cooler.

The position of the fan blades within an axial flow fan housing can be a critical factor in determining how well a fan will perform. This is especially important in high speed fans that have speeds greater than about 8000 rpm.

An examination of most high speed axial flow fans will show that the fan blades are not located at the center of the tubular housing but near one end. When the fan is located adjacent to a restricted area, such as a 90° bend, the fan blades should be positioned so that they are at the downstream end of the housing, for the best performance. Air has weight and kinetic energy, so that the air velocity must be allowed to develop to effectively overcome the flow resistance. When the fan blades are located at the downstream end of the fan housing, the air has a slightly longer flow path. This improves the velocity profile, as shown in Figures 6.3 and 6.4.

It makes no difference if a blowing or an exhaust fan system is used; the velocity profile must have a chance to develop to provide an efficient air delivery system.

The reduced flow efficiency for the fan shown in Figure 6.4 will not be obvious to a casual observer. When the fan is in operation, you can place your hand over the exhaust and feel a large volume of air flowing through the fan. However, if a thin strip of paper is slowly passed across the fan exhaust, it will show that some of the air is being short-circuited. The air

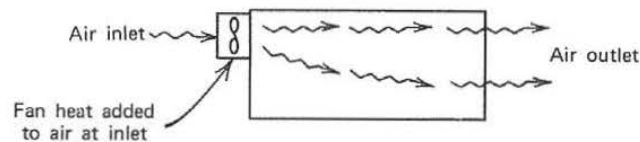


Figure 6.1 Axial flow fan blowing cooling air through a box.

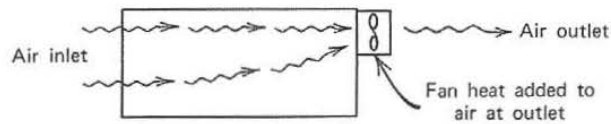


Figure 6.2 Axial flow fan drawing cooling air through a box.

flow at the outer perimeter of the fan will be moving away from the fan, but the air flow at the center will be moving toward the fan, resulting in a short circuit, as shown in Figure 6.5.

Some good fan installations and some poor ones are shown in Figure 6.6. Test data on these types of installations have shown that the cooling air flow rate can be more than doubled just by properly orienting the position of the fan blade within the fan housing when the fan is located adjacent to an area that restricts the free flow of cooling air.

6.3 STATIC PRESSURE AND VELOCITY PRESSURE

Air flow through an electronic box is due to a pressure difference between two points in the box, with the air flowing from the high pressure area to the low pressure area. The flow of air will result in a static pressure and a velocity pressure.

Static pressure is the pressure that is exerted on the walls of the container or electronic box, even when there is no flow of air; it is independent of the air velocity. Static pressure can be positive or negative, depending upon whether it is greater or less than the outside ambient pressure.

Velocity pressure is the pressure that forces the air to move through the electronic box at a certain velocity. The velocity pressure depends upon the velocity of the air and always acts in the direction of the air flow.

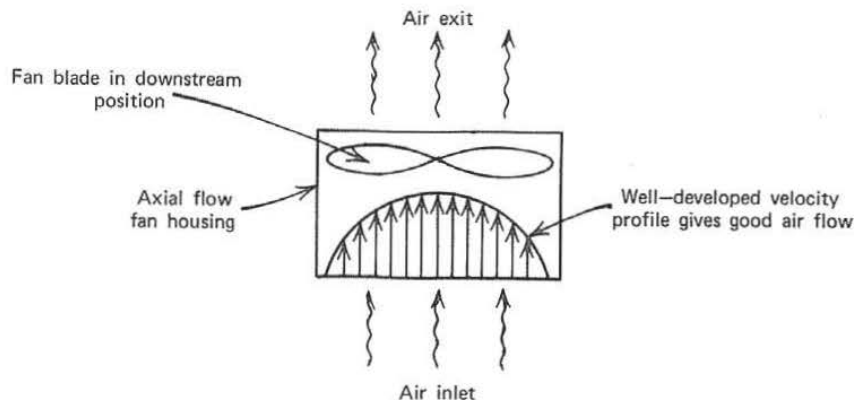


Figure 6.3 Well-developed velocity profile in an axial flow fan.

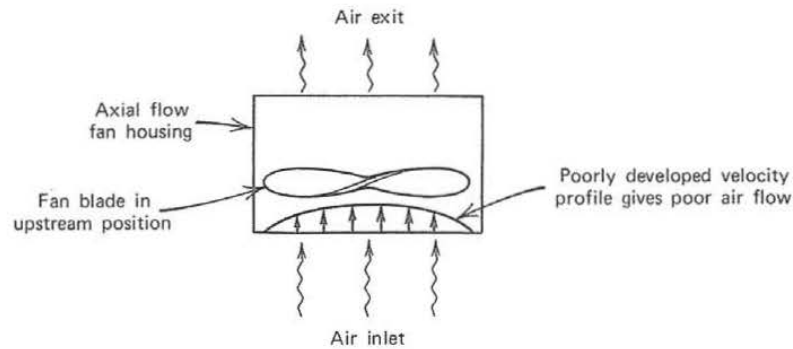


Figure 6.4 Poorly developed velocity profile in an axial flow fan.

The amount of cooling air flowing through an electronic box will usually determine the amount of heat removed from the box. If more air flows through the box, more heat will be removed. As the air flow through the box increases, however, it requires an even greater pressure to force the air through the box.

Static and velocity pressures can be expressed in lb/in^2 and g/cm^2 . However, these values are usually very small, so that it is often more convenient to express these pressures in terms of the height of a column of water. For example, 1.0 lb/in^2 (70.37 g/cm^2) equals 27.7 in of water (70.37 cm of water), or 1.0 in of water (2.54 cm of water) equals 0.036 lb/in^2 (2.54 g/cm^2). The water column height is called the "head" of water and is shown as H_s for the static head and as H_v for the velocity head.

The velocity head (H_v) is a convenient reference that is often used to determine pressure drops through electronic boxes. The velocity head can be related to the air flow velocity with the use of Eq. 6.1 [10, 18, 69].

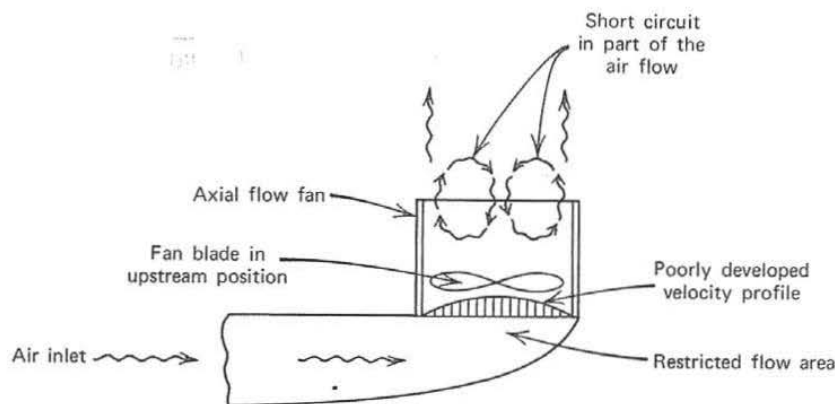


Figure 6.5 Short circuit in part of the cooling air flow due to improper positioning of the fan blades adjacent to a restricted flow area.

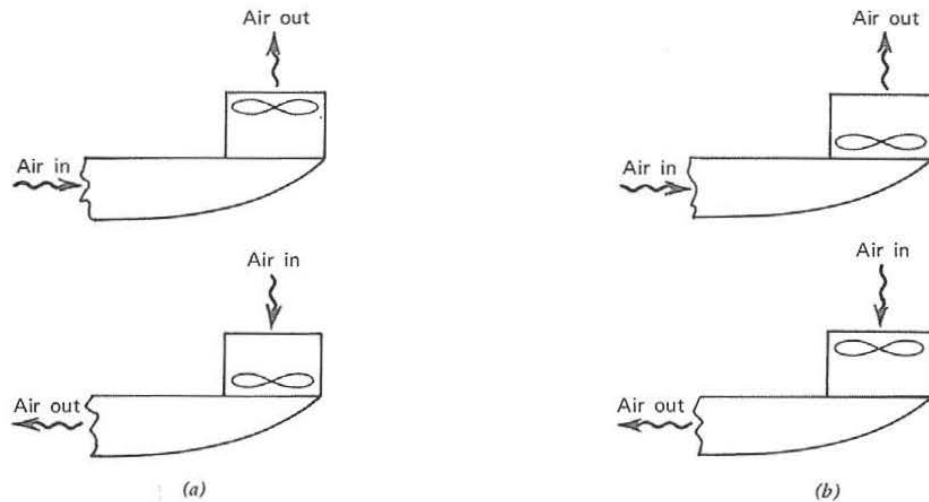


Figure 6.6 Examples of good fan installations and poor fan installations on axial flow fans located adjacent to restricted areas. (a) Good fan installations; (b) poor fan installations.

$$V = \sqrt{2gH} \quad (6.1)$$

where V = velocity = ft/sec
 g = acceleration of gravity = ft/sec²
 H = height = ft

When the air velocity is expressed in feet per minute and the height is expressed as the velocity head H_v in inches of water, Eq. 6.1 can be modified using standard air with a density of 0.0750 lb/ft³ at 69°F and 14.7 psia. For English units, this is shown in Eq. 6.2.

$$V = 60 \frac{\text{sec}}{\text{min}} \sqrt{\frac{2(32.14 \text{ ft/sec}^2)(62.4 \text{ lb/ft}^3 \text{ water})(H_v \text{ in water})}{(12 \text{ in/ft})(0.0750 \text{ lb/ft}^3 \text{ standard air})}}$$

$$V = 4005 \sqrt{H_v \text{ (in water)}} = \frac{\text{ft}}{\text{min}} \quad (6.2)$$

For metric units, this is shown in Eq. 6.2a.

$$V = \sqrt{\frac{2(979.6 \text{ cm/sec}^2)(1 \text{ g/cm}^3 \text{ water})(H_v \text{ cm water})}{0.0012 \text{ g/cm}^3 \text{ air at } 20.5^\circ\text{C}}}$$

$$V = 1277 \sqrt{H_v \text{ (cm water)}} = \frac{\text{cm}}{\text{sec}} \quad (6.2a)$$

A sealed electronic box with a gauge pressure of 0.05 lb/in² (3.518 g/cm²)

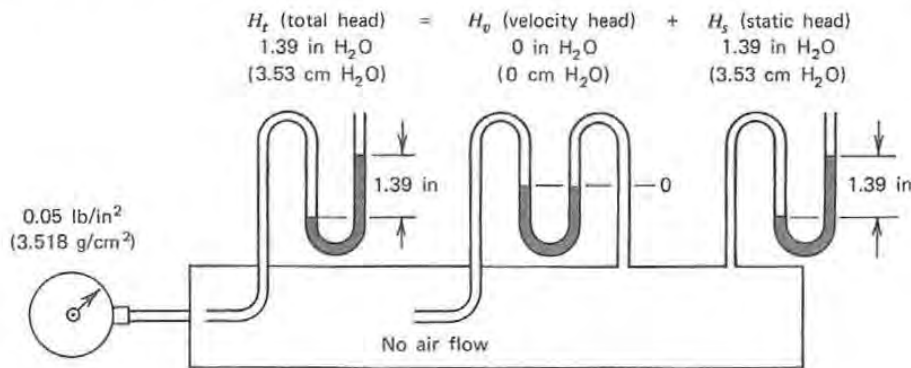


Figure 6.7 A pressurized electronic box with no air flow.

with no air flow will have a pressure equal to the static pressure. If small holes are drilled through the walls and if water filled manometers are used to measure the various pressure heads, the results will appear as shown in Figure 6.7 [40, 41, 69].

When a fan blows air through the electronic box, the pressure within the box will be slightly higher than the outside air pressure. A velocity head will now be developed, as shown in Figure 6.8 [40, 41, 69].

The total head will be the sum of the velocity head and the static head, as shown by Eq. 6.3 [40, 41, 69].

$$H_t = H_v + H_s \quad (6.3)$$

When an exhaust fan is used, and the air is drawn through the box, the pressure within the box will be slightly lower than the outside air pressure, and the pressure head characteristics will appear as shown in Figure 6.9. The total head is still shown by Eq. 6.3 [40, 41, 69].

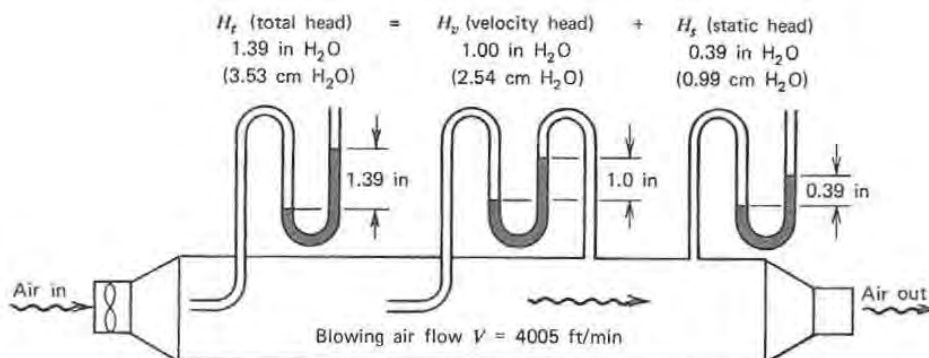


Figure 6.8 Pressure head characteristics when the fan blows air through an electronic box.

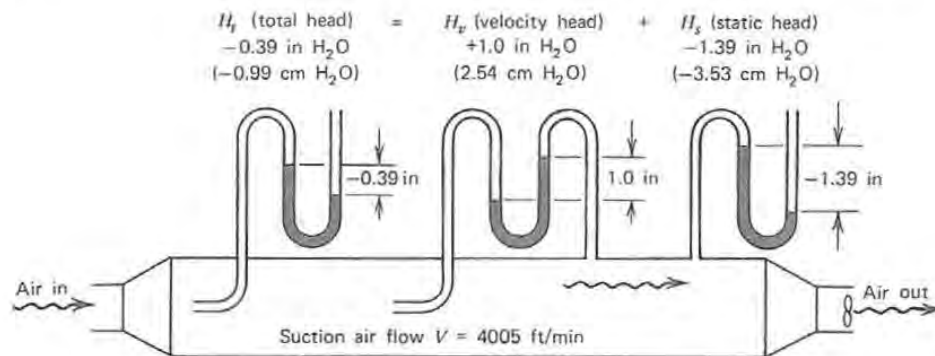


Figure 6.9 Pressure head characteristics when fan draws air through an electronic box.

6.4 LOSSES EXPRESSED IN TERMS OF VELOCITY HEADS

As the cooling air flows through the electronic box, it will experience friction and dynamic losses. There are many ways of expressing these losses. One convenient method is to express the *static* pressure losses in terms of the *velocity* head for the air at some specific point. For example, there will be friction and dynamic losses at the inlets to the fans shown in Figures 6.1 through 6.6. These losses can be expressed as some ratio of the velocity head at the fan inlet. If the loss is judged to be equal to one velocity head, or perhaps two velocity heads for a severely restricted flow area, Eq. 6.2 can be used to determine the actual pressure loss in inches of water, based upon the velocity of the air flow at that section [40, 41, 69].

The loss in a 90° elbow can be expressed as some ratio of the velocity head at that elbow, which is related to the velocity of the air flowing through the elbow. A loss in a sudden expansion is expressed as a ratio of the velocity head just before the expansion. A typical sudden expansion will result in a loss of one velocity head, because the air velocity at the expansion is sharply reduced. Most of the energy required to provide that velocity at the point just before the expansion is lost. That energy was originally supplied by the fan. If more energy is lost, a larger fan is required to supply that energy. If the losses in an air flow system can be reduced, by avoiding such things as sharp turns and sudden expansions, a smaller fan can be used.

Air velocities must be kept low to avoid large losses. High air velocities result in high losses. A system with high losses will require a fan or blower that can supply the required cooling air at a higher pressure. This usually results in a larger and heavier fan or blower, with greater power requirements and higher costs [42].

6.5 SAMPLE PROBLEM—AIR FLOW LOSS AT A FAN ENTRANCE

The axial flow fan shown in Figure 6.1 has an air inlet velocity of 3500 ft/min (1778 cm/sec). It is estimated that the turbulence and contraction in the air at the entrance to the fan will result in a static loss of about 1.5 velocity heads. Determine the static head loss in terms of the height of a column of water.

SOLUTION

The static loss represented by one velocity head can be determined by rewriting Eq. 6.2 slightly, as shown in Eq. 6.4 for English units.

$$H_v = \left(\frac{V}{4005} \right)^2 \quad (6.4)$$

where $V = 3500$ ft/min

$$H_v = \left(\frac{3500}{4005} \right)^2 = 0.764 \text{ in H}_2\text{O for one velocity head} \quad (6.5)$$

Using Eq. 6.2a for metric units yields

$$H_v = \left(\frac{V}{1277} \right)^2$$

$$H_v = \left(\frac{1778}{1277} \right)^2 = 1.94 \text{ cm H}_2\text{O for one velocity head} \quad (6.5a)$$

The static head loss at the fan entrance is expected to be 1.5 velocity heads. The static loss expected in English units is shown in Eq. 6.6.

$$H_{\text{loss}}^{\text{fan}} = 1.5(0.764 \text{ in H}_2\text{O}) = 1.146 \text{ in H}_2\text{O} \quad (6.6)$$

The static loss expected, in metric units, is shown in Eq. 6.6a.

$$H_{\text{loss}}^{\text{fan}} = 1.5(1.94 \text{ cm H}_2\text{O}) = 2.91 \text{ cm H}_2\text{O} \quad (6.6a)$$

The air exit velocity should also be kept low, if possible, to reduce the velocity head lost at the exit. Sometimes a gradual enlargement can be incorporated at the exit. This will help to convert some of the velocity head to a higher static head, which will help to increase the flow of air through the box.

6.6 ESTABLISHING THE FLOW IMPEDANCE CURVE FOR AN ELECTRONIC BOX

Electronic boxes that are cooled with the use of fans must be carefully evaluated to make sure the fan will provide the proper cooling. If the fan is too small for the box, the electronic system may overheat and fail. If the fan is too big for the box, the cooling will be adequate, but the larger fan will be more expensive, heavier, and will draw more power.

Air flowing through the electronic box will encounter resistance as it enters different chambers and is forced to make many turns. This flow resistance is approximately proportional to the square of the velocity, so that it is approximately proportional to the square of the flow rate in cubic foot per minute (cfm). When the static pressure of the air flow through a box is plotted against the cfm air flow rate, the result will be a parabolic curve. This curve can be generated by considering the various flow resistances the moving air will encounter as it flows through the box. The method of analysis is to assume several different cfm flow rates through the box and then to calculate a static pressure drop through the box for each flow rate. The result will be a curve similar to Figure 6.10.

Once the box flow impedance curve has been developed, it is necessary to examine different fan impedance curves to see how well the fans will match the box. A typical fan impedance curve is shown in Figure 6.11.

If the impedance curve for the box is superimposed on the impedance curve for the fan, they will intersect. The point of intersection represents the actual operating point for the system, as shown in Figure 6.12 [42].

The method of pressure drop analysis just outlined can be demonstrated by considering a typical fan cooled electronic box.

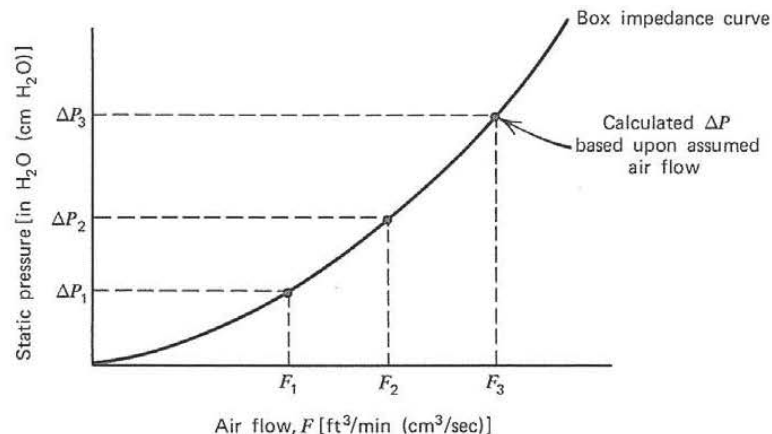


Figure 6.10 Typical air flow impedance curve for an electronic box.

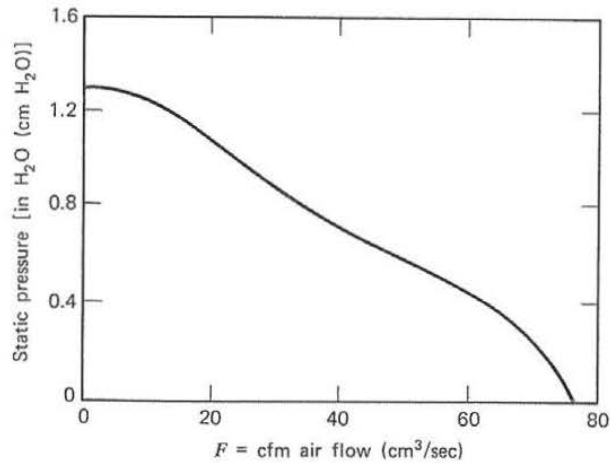


Figure 6.11 Typical flow impedance curve for a fan.

6.7 SAMPLE PROBLEM—FAN COOLED ELECTRONIC BOX

Determine the pressure drop characteristics of the electronic box in Figure 6.13, and determine a fan size for the system. The system must be capable of continuous operation in a 131°F (55°C) ambient at sea level conditions. The maximum allowable hot spot component surface temperature is limited to 212°F (100°C). The system contains seven PCBs, each dissipating 20 watts, for a total power dissipation of 140 watts. This does not include the fan power dissipation.

SOLUTION

The box must be examined in two phases to ensure the integrity of the complete design. In phase 1, the thermal design of the box is examined,

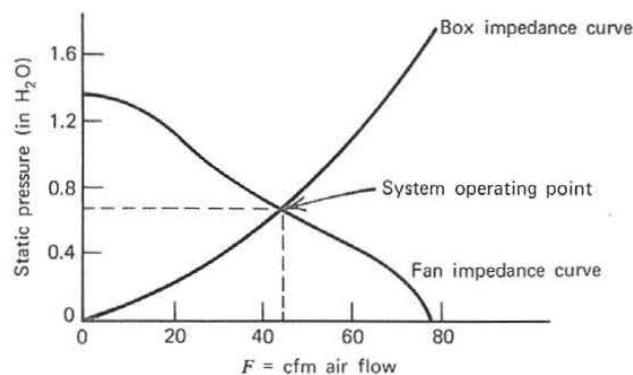


Figure 6.12 Intersection of fan and box impedance curves.

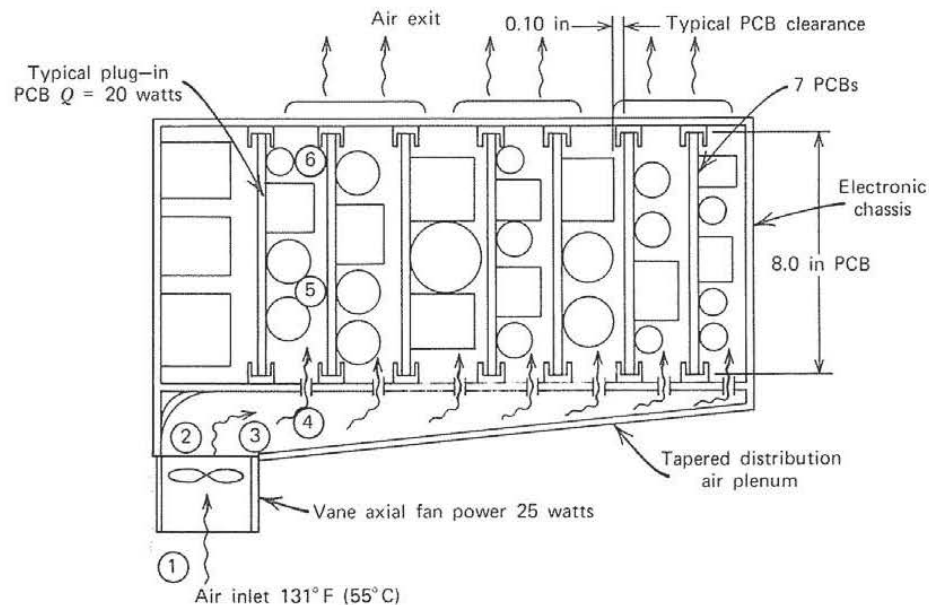


Figure 6.13 Plan view of fan cooled electronic box.

with the proposed fan, to make sure the component hot spot temperature of 212°F (100°C) is not exceeded. In phase 2, the electronic chassis air flow impedance curve is developed and matched with several fans, to make sure there is sufficient cooling air available for the system.

Phase 1. Electronic Box Thermal Design

A preliminary thermal analysis must be made of the system to determine approximately how much cooling air is required. The preliminary analysis can also be used to establish an approximate fan size.

The maximum allowable temperature rise from the air inlet to the component surface hot spot temperature is 212°F – 131°F = 81°F (100°C – 55°C = 45°C). There are two major contributors to this temperature rise.

- 1 Δt due to the heat input from the fan and the electronics to the cooling air.
- 2 Δt due to the thermal resistance across the convection film from the cooling air to the surface of the component.

These temperature rise sources can be examined in more detail to see how much each contributes to the total value.

- 1 Δt due to the heat input from the fan motor to the cooling air.

Heat generated by the fan motor is usually removed by convection from the cooling air. If the fan is bolted to a large heat sink away from the electronic box, a large part of the heat from the fan motor may be removed by direct conduction and radiation from the fan housing to the heat sink.

When the fan is bolted directly to the chassis that houses the electronics, the chassis walls may be quite hot, so relatively little fan heat may be transferred to the chassis. Therefore, it is a good practice to be slightly conservative and to assume that all of the fan power will be picked up by the cooling air as it passes through the fan and into the chassis housing.

Several types of fans can be used for cooling the electronics. Centrifugal fans (sometimes called radial or squirrel cage blowers) can deliver a large quantity of air against a high static pressure. Also, there are probably more centrifugal fans in general use for cooling than any other type of fan. However, centrifugal fans are generally larger and heavier than tube axial and vane axial fans for the same performance in small fan sizes. Axial flow fans will be required to run at higher speeds to get the same performance, so that they will generate more noise than will centrifugal fans.

If a small, high speed axial flow fan is selected, the noise generated may be objectionable to personnel working in areas adjacent to the fan. Simple mufflers can be made of plywood and foam packing material to reduce the noise levels when these fans are operating with room ambient cooling air.

Past experience with air cooled electronic systems has shown that satisfactory thermal performance can be obtained if the cooling air exit temperature from the electronic chassis does not exceed 160°F (71°C). The allowable cooling air temperature rise then becomes

$$\Delta t_A = 160^\circ\text{F} - 131^\circ\text{F} = 29^\circ\text{F}$$

or

$$\Delta t_A = 71^\circ\text{C} - 55^\circ\text{C} = 16^\circ\text{C}$$

The required cooling air weight flow for the system can then be determined from the general weight flow equation [15, 17, 24].

$$W = \frac{Q}{\Delta t C_p} \quad (6.7)$$

Given

<i>Item</i>	<i>English Units</i>	<i>Metric Units</i>
Q = heat	= Btu/hr	= cal/sec
Δt = temperature rise	= °F	= °C
C_p = specific heat	= Btu/lb °F	= cal/g °C
W = air flow	= lb/hr	= g/sec

Considering a lightweight vane axial flow fan similar to the Rotron Axi-max 2, which weighs only about 0.25 lb, the power dissipation for a 15,000 rpm fan will be about 25 watts. The total box power dissipation in English units then becomes:

$$Q = 140 + 25 = 165 \text{ watts} \times 3.413 \frac{\text{Btu/hr}}{\text{watts}} = 563 \frac{\text{Btu}}{\text{hr}} \text{ (heat)}$$

$$\Delta t = 160 - 131 = 29^\circ\text{F} \text{ (cooling air temperature rise)}$$

$$C_p = 0.24 \frac{\text{Btu}}{\text{lb } ^\circ\text{F}} \text{ (specific heat)}$$

The cooling air flow through the box must be

$$W = \frac{563 \text{ Btu/hr}}{(29^\circ\text{F})(60 \text{ min/hr})(0.24 \text{ Btu/lb } ^\circ\text{F})} = 1.35 \frac{\text{lb}}{\text{min}} \quad (6.8)$$

In metric units:

$$Q = 140 + 25 = 165 \text{ watts} \times 0.239 \frac{\text{cal/sec}}{\text{watt}} = 39.4 \text{ cal/sec (heat)}$$

$$\Delta t = 71 - 55 = 16^\circ\text{C}$$

$$C_p = 0.24 \frac{\text{cal}}{\text{g } ^\circ\text{C}} \text{ (specific heat)}$$

$$W = \frac{39.4 \text{ cal/sec}}{(16^\circ\text{C})(0.24 \text{ cal/g } ^\circ\text{C})} = 10.3 \frac{\text{g}}{\text{sec}} \quad (6.8a)$$

- 2 Δt due to the thermal resistance across the convection film from the cooling air to the surface of the component.

This temperature rise can be determined from the convection relation shown by Eq. 5.7.

$$\Delta t = \frac{Q}{h_c A} \quad (\text{ref. Eq. 5.7})$$

The convection coefficient h_c is determined from the geometry of the air duct that is formed by the air space between the plug-in PCBs. Its value is determined from Eq. 6.9 [3, 15, 43].

$$h_c = JC_p G \left(\frac{C_p \mu}{K} \right)^{-2/3} \quad (6.9)$$

Given J = Colburn factor
 C_p = specific heat of fluid (ref. Figure 6.28)
 G = weight velocity flow through duct
 μ = viscosity of fluid (ref. Figure 6.28)
 K = thermal conductivity of fluid (ref. Figure 6.28)

The Colburn J factor can be determined from the Reynolds number (N_R). For laminar flow conditions with N_R between 200 and 1800, the approximate J value for rectangular ducts with aspect ratios greater than about 8 is shown in Eq. 6.10 (compiled from data shown in Kays and London, *Compact Heat Exchangers*) [43].

$$J = \frac{6}{(N_R)^{0.98}} \quad (6.10)$$

For square ducts (aspect ratio = 1.0) the approximate J value for the same laminar flow conditions is as follows:

$$J = \frac{2.7}{(N_R)^{0.95}} \quad (6.10a)$$

For turbulent conditions, where N_R is between 10,000 and 120,000 McAdams [15] shows that

$$J = \frac{0.023}{(N_R)^{0.2}} \quad (6.10b)$$

The air duct formed by the clearance between the components and the PCBs is shown in Figure 6.14.

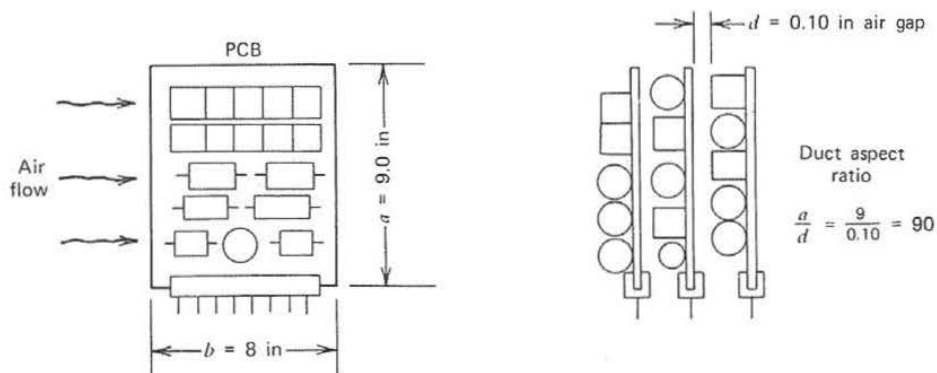


Figure 6.14 Air flow duct formed between printed circuit boards.

The Reynolds number for the air duct formed by the PCBs is shown in Eq. 6.11.

$$N_R = \frac{VD\rho}{\mu} = \frac{GD}{\mu} \quad (6.11)$$

$$\text{Given } D = \frac{4 \times \text{area}}{\text{perimeter}} = \frac{4ad}{2(a+d)} = \frac{2ad}{a+d} \text{ (hydraulic diameter)}$$

In English units:

$$D = \frac{2(9.0)(0.10)}{9.0 + 0.10} = 0.198 \text{ in} = 0.0165 \text{ ft}$$

In metric units:

$$D = \frac{2(22.86)(0.254)}{22.86 + 0.254} = 0.502 \text{ cm}$$

The weight velocity flow G through the duct formed by the PCBs is determined from Eq. 6.12.

$$G = \frac{W}{A} \quad (6.12)$$

$$\text{Given } W = 1.35 \text{ lb/min (10.3 g/sec) (ref. Eq. 6.8)}$$

$$A = \frac{(9.0)(0.10) \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 0.00625 \text{ ft}^2 \text{ (area one duct)}$$

In English units:

$$G = \frac{(1.35 \text{ lb/min})(60 \text{ min/hr})}{0.00625 \text{ ft}^2 \times 7 \text{ ducts}} = 1851 \frac{\text{lb}}{\text{hr ft}^2} \quad (6.13)$$

In metric units:

$$G = \frac{10.3 \text{ g/sec}}{(22.86)(0.254) \text{ cm}^2 \times 7 \text{ ducts}} = 0.253 \frac{\text{g}}{\text{sec cm}^2} \quad (6.13a)$$

$$\mu = 0.050 \frac{\text{lb}}{\text{ft hr}} = 0.000207 \frac{\text{g}}{\text{cm sec}} \text{ [ref. Fig. 6.28; viscosity at } 150^\circ\text{F (65.5}^\circ\text{C)}]$$

Substitute into Eq. 6.11 for the Reynolds number, in English units:

$$N_R = \frac{(1851 \text{ lb/hr ft}^2)(0.0165 \text{ ft})}{0.050 \text{ lb/ft hr}} = 611 \text{ (dimensionless)} \quad (6.14)$$

In metric units:

$$N_R = \frac{(0.253 \text{ g/sec cm}^2)(0.502 \text{ cm})}{0.000207 \text{ g/cm sec}} = 613 \text{ (dimensionless)} \quad (6.14a)$$

$$K = 0.017 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} = 0.000070 \frac{\text{cal}}{\text{sec cm}^2 \text{ } ^\circ\text{C}} \text{ (ref. Fig. 6.28; thermal conductivity at } 150^\circ\text{F)}$$

Substitute into Eq. 6.10 for the J factor.

$$J = \frac{6}{(N_R)^{0.98}} = \frac{6}{(611)^{0.98}} = 0.0111 \text{ (dimensionless)}$$

Substitute into Eq. 6.9 for the forced air convection coefficient, in English units.

$$h_c = \frac{(0.0111)(0.24)(1851 \text{ lb/hr ft}^2)}{[(0.24 \times 0.05)/0.017]^{0.666}} = 6.22 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} \quad (6.15)$$

In metric units:

$$h_c = \frac{(0.0111)(0.24)(0.253 \text{ g/sec cm}^2)}{[(0.24 \times 0.000207)/0.000070]^{0.666}} = 0.000846 \frac{\text{cal}}{\text{sec cm}^2 \text{ } ^\circ\text{C}} \quad (6.15a)$$

Section 5.11 points out that some of the surface area on the backside of flow soldered PCBs is available for heat dissipation when the component lead wires extend through the PCB. When there is copper on the back side of such a PCB, the heat can flow through the PCB to the copper, which acts as a heat spreader. As much as 30% of the back side of a PCB may be available for effective heat transfer. The total heat transfer surface area may then be 1.3 times the area of one face on the PCB.

The forced convection coefficient h_c can be increased by using a smaller air gap between the PCBs. For example, if the air gap is reduced from 0.10 in to 0.08 in, the forced convection coefficient will increase to a value of about 7.8 Btu/hr ft² °F. However, if the electronic system must operate in a severe vibration environment, there must be sufficient sway space between the PCBs, to prevent tall electronic components from striking the sharp lead wires on the back surface of the adjacent PCB.

The temperature rise across the convection film from the air stream to

the component surface can now be determined using Eq. 5.7. The heat from the fan is not included with the heat generated by the PCBs. Each PCB dissipates 20 watts = 68.3 Btu/hr.

$$\begin{aligned} \text{Given } Q &= 68.3 \text{ Btu/hr} = 4.78 \text{ cal/sec (heat)} \\ h_c &= 6.22 \text{ Btu/hr ft}^2 \text{ }^\circ\text{F} = 0.000846 \text{ cal/sec cm}^2 \text{ }^\circ\text{C (convection)} \end{aligned}$$

In English units:

$$\begin{aligned} A &= \frac{(1.3)(8)(9) \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} \\ &= 0.65 \text{ ft}^2 \text{ (PCB area, including the back side of the PCB)} \end{aligned}$$

In metric units:

$$A = (1.3)(20.32)(22.86) = 603.8 \text{ cm}^2 \text{ (PCB area)}$$

The temperature rise, in English units, is

$$\Delta t_B = \frac{68.3}{(6.22)(0.65)} = 16.9^\circ\text{F} \quad (6.16)$$

In metric units:

$$\Delta t_B = \frac{4.78}{(0.000846)(603.8)} = 9.3^\circ\text{C} \quad (6.16a)$$

The component hot spot surface temperature t_s is determined from the cooling air inlet temperature, plus the temperature rises to the component surface, as shown in Eq. 6.17.

$$t_s = t_{\text{in}} + \Delta t_A + \Delta t_B \quad (6.17)$$

In English units, the component surface temperature is

$$t_s = 131 + 29 + 16.9 = 176.9^\circ\text{F} \quad (6.18)$$

In metric units:

$$t_s = 55 + 16 + 9.3 = 80.3^\circ\text{C} \quad (6.18a)$$

Since the component hot spot surface temperature is less than 212°F (100°C), the initial design concept with the vane axial flow fan appears to be satisfactory.

Phase 2. Electronic Chassis Air Flow Impedance Curve

The air flow conditions are examined at six different points in the chassis, where the maximum static pressure losses are expected to occur, as shown in Figure 6.13. These static pressure losses are itemized as follows:

- 1 Air inlet to fan
- 2 90° turn and transition to an oval section
- 3 Contraction and transition to a rectangular section
- 4 Plenum entrance to PCB duct
- 5 Flow through PCB channel duct
- 6 Exhaust from PCB duct and chassis

The static pressure losses are examined in greater detail using the velocity head loss technique discussed in Section 6.4.

- 1 *Air Inlet to Fan.* The air inlet to the axial flow fan is very similar to the plain end duct, as shown in Table 6.2 (Section 6.12). Ducts of this type can be expected to show a static pressure loss of 0.93 velocity head. The loss is conservatively rounded off to 1.0 velocity head, as shown in Eq. 6.19.

$$H_1 = 1.0H_{v1} \quad (6.19)$$

- 2 *90° Turn and Transition from a Round Section at the Fan to an Oval Section.* The radius of the turn is rather sharp, probably less than 1 diameter. Also, the section is changing from round to oval (the section will eventually become rectangular).

$$90^\circ \text{ turn} = 0.7 H_v$$

$$\text{Transition} = \frac{0.2 H_v}{0.9 H_v}$$

Conservatively round off this loss to unity.

$$H_2 = 1.0H_{v2} \quad (6.20)$$

- 3 *Contraction and Transition to Rectangular Section*

$$\text{Contraction} = 0.2H_v$$

$$\text{Transition} = \frac{0.2H_v}{0.4H_v}$$

Conservatively round off to $0.5H_v$.

$$H_3 = 0.5H_{v3} \quad (6.21)$$

- 4 *Plenum Entrance to Printed Circuit Board (PCB) Duct.* The air must make an abrupt change at this point. It must make a 90° turn and enter the PCB duct area. This is a critical area with respect to pressure drop. If the inlet extends slightly into the air stream, as shown in Figure 6.15a, the pressure loss can exceed 3 velocity heads as a result of the turbulence generated ($2.7H_v$ entry + $1.0H_v$ for 90° turn = $3.7H_v$ total loss) [41, 69].

An attempt should be made to keep the entrance flush. It is much better to bevel the edges as shown in Figure 6.15b, where the pressure loss is only about 1.5 velocity heads ($0.5H_v$ entry + $1.0H_v$ turn = $1.5H_v$ total loss).

Conservatively assume a flush edge.

$$\begin{aligned} \text{Entrance} &= 1.0H_v \\ 90^\circ \text{ turn} &= 1.0H_v \\ H_4 &= \frac{1.0H_v + 1.0H_v}{2.0H_{v4}} \end{aligned} \quad (6.22)$$

- 5 *Air Flow through PCB Channel Duct.* In this case the duct is really the space between the backside of one board and the electronic components on the adjacent board. The back side of the PCB has small exposed tips of the electrical lead wires, and the front side of the PCB is populated with electronic components. The loss can be expressed in terms of air flowing through a rectangular duct as shown in Section 6.24 (Eq. 6.82 or 6.83). Using velocity heads for convenience, a high pressure loss is expected for this condition because of the narrow (0.10 in) air gap.

$$H_5 = 1.0H_{v5} \quad (6.23)$$



Figure 6.15 Protruding duct creates turbulence that increases pressure loss at air inlet (a); beveled edge reduces pressure loss at air inlet (b).

- 6 *Exhaust from PCB Duct and Chassis.* This calculation is normally not included in the static pressure drop loss through the box. However, it is always a good idea to see how much energy is being thrown away at the exit. Remember that the fan must supply all of the air-moving energy. If too much is thrown away at the exhaust, the fan has to work harder. When the exhaust loss is too high in a critical design, some attempt should be made to recover part of the loss by slightly enlarging the exhaust area to reduce the exhaust velocity.

As the cooling air exhausts from the box, it will expand suddenly, so that it will lose its velocity completely.

$$H_6 = 1.0H_{v6} \quad (6.24)$$

Note that the air flows across all of the PCBs in a parallel flow path. The pressure drop across this section will all be about the same for all of the PCBs. Therefore, only one air flow path across the circuit board stack has to be examined to obtain the pressure drop.

All of the information is now available to calculate the approximate pressure drop through the box based upon the geometry of the air flow path. Three different cfm air flow conditions will be assumed: 10 cfm (4719 cm³/sec), 20 cfm (9438 cm³/sec), and 30 cfm (14157 cm³/sec). The pressure drops at each of the six points in Figure 6.13 will be determined for each of the three assumed flow conditions, starting with the 10 cfm (4719 cm³/sec) flow condition.

- 1 *Air Inlet to Fan.* Equation 6.4 shows that the velocity head H_v is a function of the air flow velocity in feet per minute. Starting with an assumed air flow of 10 cfm, the cross section flow area through the axial flow fan is required, and is shown in Figure 6.16.

$$A_1 = \frac{(\pi/4)[(1.88)^2 - (1.10)^2] \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 0.0127 \text{ ft}^2 = 11.8 \text{ cm}^2$$

$$V_1 = \frac{F}{A_1} = \frac{10 \text{ ft}^3/\text{min}}{0.0127 \text{ ft}^2} = 787 \frac{\text{ft}}{\text{min}} = 400 \frac{\text{cm}}{\text{sec}}$$

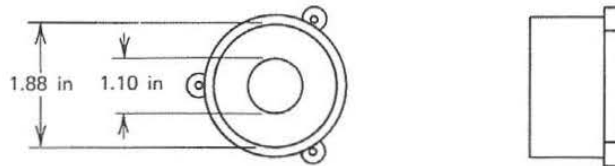


Figure 6.16 Cross section flow area for a small lightweight vane axial flow fan (similar to Rotron Aximax 2).

Substitute into Eq. 6.4 for the velocity head at point 1 in English units.

$$H_{v1} = \left(\frac{V}{4005} \right)^2 = \left(\frac{787}{4005} \right)^2 = 0.0386 \text{ in H}_2\text{O} \quad (6.25)$$

In metric units:

$$H_{v1} = \left(\frac{V}{1277} \right)^2 = \left(\frac{400}{1277} \right)^2 = 0.098 \text{ cm H}_2\text{O} \quad (6.25a)$$

The losses for each flow rate of 10 cfm, 20 cfm, and 30 cfm can be determined from Eq. 6.19 by using a ratio of the 10 cfm flow rate.

$$\left. \begin{array}{l} 10 \text{ cfm: } H_1 = (1.0)(0.0386) = 0.0386 \text{ in H}_2\text{O} = 0.098 \text{ cm H}_2\text{O} \\ 20 \text{ cfm: } H_1 = \left(\frac{20}{10} \right)^2 (0.0386) = 0.1544 \text{ in H}_2\text{O} = 0.392 \text{ cm H}_2\text{O} \\ 30 \text{ cfm: } H_1 = \left(\frac{30}{10} \right)^2 (0.0386) = 0.3474 \text{ in H}_2\text{O} = 0.882 \text{ cm H}_2\text{O} \end{array} \right\} \quad (6.26)$$

2 *90° Turn and Transition from a Round Section at the Fan to an Oval Section.* The cross section area at section 2 is shown in Figure 6.17.

$$A_2 = \frac{(\pi/4)(0.5)^2 + (3.5)(0.5) \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 0.0135 \text{ ft}^2 = 12.56 \text{ cm}^2$$

$$V_2 = \frac{10 \text{ ft}^3/\text{min}}{0.0135 \text{ ft}^2} = 741 \frac{\text{ft}}{\text{min}} = 376 \frac{\text{cm}}{\text{sec}}$$

Substitute into Eq. 6.4 for the velocity head at point 2.

$$H_{v2} = \left(\frac{741}{4005} \right)^2 = 0.034 \text{ in H}_2\text{O} = 0.087 \text{ cm H}_2\text{O}$$

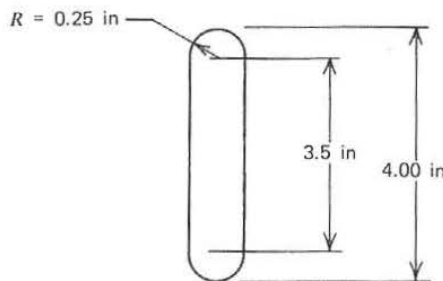


Figure 6.17 Cross section of air flow path at point 2.

The losses for each flow rate at point 2 are determined from Eq. 6.20.

$$\left. \begin{array}{l} 10 \text{ cfm: } H_2 = (1.0)(0.034) = 0.034 \text{ in H}_2\text{O} = 0.086 \text{ cm H}_2\text{O} \\ 20 \text{ cfm: } H_2 = \left(\frac{20}{10}\right)^2 (0.034) = 0.136 \text{ in H}_2\text{O} = 0.345 \text{ cm H}_2\text{O} \\ 30 \text{ cfm: } H_2 = \left(\frac{30}{10}\right)^2 (0.034) = 0.306 \text{ in H}_2\text{O} = 0.777 \text{ cm H}_2\text{O} \end{array} \right\} (6.27)$$

- 3 *Contraction and Transition to Rectangular Section.* The cross section area at section 3 is shown in Figure 6.18.

$$V_3 = \frac{10 \text{ cfm}}{0.0104} = 961 \frac{\text{ft}}{\text{min}} = 488 \frac{\text{cm}}{\text{sec}}$$

$$H_{v3} = \left(\frac{961}{4005}\right)^2 = 0.0575 \text{ in H}_2\text{O} = 0.146 \text{ cm H}_2\text{O}$$

The losses for each flow rate at point 3 are determined from Eq. 6.21.

$$\left. \begin{array}{l} 10 \text{ cfm: } H_3 = (0.5)(0.0575) = 0.0287 \text{ in H}_2\text{O} = 0.0729 \text{ cm H}_2\text{O} \\ 20 \text{ cfm: } H_3 = \left(\frac{20}{10}\right)^2 (0.0287) = 0.115 \text{ in H}_2\text{O} = 0.292 \text{ cm H}_2\text{O} \\ 30 \text{ cfm: } H_3 = \left(\frac{30}{10}\right)^2 (0.0287) = 0.259 \text{ in H}_2\text{O} = 0.657 \text{ cm H}_2\text{O} \end{array} \right\} (6.28)$$

- 4 *Plenum Entrance to PCB Duct.* The cross section at section 4 is shown in Figure 6.19. Since there are seven card slots, approximately $\frac{1}{7}$ of the total flow will pass through each slot if the tapered distribution duct shown in Figure 6.13 does its job properly. When the power dissipation is not the same on each PCB, more air should be distributed to the PCBs that have a higher power dissipation. A mockup of the

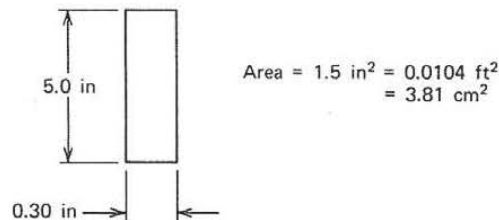


Figure 6.18 Cross section of air flow path at point 3.

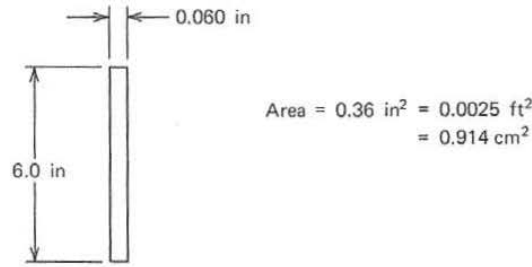


Figure 6.19 Inlet slot for distributing cooling air to PCB duct.

air flow system is usually built and tested, to make sure the air distribution to each PCB is adequate.

$$V_4 = \frac{10 \text{ cfm}}{(0.0025 \text{ ft}^2)(7 \text{ slots})} = 571 \frac{\text{ft}}{\text{min}} = 290 \frac{\text{cm}}{\text{sec}}$$

$$H_{v4} = \left(\frac{571}{4005} \right)^2 = 0.0203 \text{ in H}_2\text{O} = 0.0516 \text{ cm H}_2\text{O}$$

The losses for each flow rate at point 4 are determined from Eq. 6.22.

$$\left. \begin{array}{l} 10 \text{ cfm: } H_4 = (2)(0.0203) = 0.0406 \text{ in H}_2\text{O} = 0.103 \text{ cm H}_2\text{O} \\ 20 \text{ cfm: } H_4 = \left(\frac{20}{10} \right)^2 (0.0406) = 0.162 \text{ in H}_2\text{O} = 0.411 \text{ cm H}_2\text{O} \\ 30 \text{ cfm: } H_4 = \left(\frac{30}{10} \right)^2 (0.0406) = 0.365 \text{ in H}_2\text{O} = 0.927 \text{ cm H}_2\text{O} \end{array} \right\} (6.29)$$

- 5 Air Flow through PCB Channel Duct. The cross section of the duct formed by the PCBs is shown in Figure 6.14.

$$A_5 = (0.10)(9.0) = 0.90 \text{ in}^2 = 0.00625 \text{ ft}^2 = 5.80 \text{ cm}^2$$

$$V_5 = \frac{10 \text{ cfm}}{(0.00625 \text{ ft}^2)(7 \text{ slots})} = 228 \frac{\text{ft}}{\text{min}} = 116 \frac{\text{cm}}{\text{sec}}$$

$$H_{v5} = \left(\frac{228}{4005} \right)^2 = 0.0032 \text{ in H}_2\text{O} = 0.0082 \text{ cm H}_2\text{O}$$

The losses for each flow rate at point 5 are determined from Eq. 6.23.

$$\begin{aligned}
 10 \text{ cfm: } H_5 &= (1.0)(0.0032) = 0.0032 \text{ in H}_2\text{O} = 0.0081 \text{ cm H}_2\text{O} \\
 20 \text{ cfm: } H_5 &= \left(\frac{20}{10}\right)^2 (0.0032) = 0.0128 \text{ in H}_2\text{O} = 0.0325 \text{ cm H}_2\text{O} \\
 30 \text{ cfm: } H_5 &= \left(\frac{30}{10}\right)^2 (0.0032) = 0.0288 \text{ in H}_2\text{O} = 0.0731 \text{ cm H}_2\text{O}
 \end{aligned}
 \quad \left. \vphantom{\begin{aligned} 10 \text{ cfm: } H_5 &= (1.0)(0.0032) = 0.0032 \text{ in H}_2\text{O} = 0.0081 \text{ cm H}_2\text{O} \\ 20 \text{ cfm: } H_5 &= \left(\frac{20}{10}\right)^2 (0.0032) = 0.0128 \text{ in H}_2\text{O} = 0.0325 \text{ cm H}_2\text{O} \\ 30 \text{ cfm: } H_5 &= \left(\frac{30}{10}\right)^2 (0.0032) = 0.0288 \text{ in H}_2\text{O} = 0.0731 \text{ cm H}_2\text{O} \end{aligned}} \right\} (6.30)$$

- 6 *Exhaust from PCB Duct and Chassis.* The velocity of the air at the PCB exit is the same as the velocity at point 5. However, at the exit, one full velocity head is lost, as shown by Eq. 6.24. The losses for each flow rate are as follows:

$$\begin{aligned}
 10 \text{ cfm: } H_6 &= (1.0)(0.0032) = 0.0032 \text{ in H}_2\text{O} = 0.0081 \text{ cm H}_2\text{O} \\
 20 \text{ cfm: } H_6 &= \left(\frac{20}{10}\right)^2 (0.0032) = 0.0128 \text{ in H}_2\text{O} = 0.0325 \text{ cm H}_2\text{O} \\
 30 \text{ cfm: } H_6 &= \left(\frac{30}{10}\right)^2 (0.0032) = 0.0288 \text{ in H}_2\text{O} = 0.0731 \text{ cm H}_2\text{O}
 \end{aligned}
 \quad \left. \vphantom{\begin{aligned} 10 \text{ cfm: } H_6 &= (1.0)(0.0032) = 0.0032 \text{ in H}_2\text{O} = 0.0081 \text{ cm H}_2\text{O} \\ 20 \text{ cfm: } H_6 &= \left(\frac{20}{10}\right)^2 (0.0032) = 0.0128 \text{ in H}_2\text{O} = 0.0325 \text{ cm H}_2\text{O} \\ 30 \text{ cfm: } H_6 &= \left(\frac{30}{10}\right)^2 (0.0032) = 0.0288 \text{ in H}_2\text{O} = 0.0731 \text{ cm H}_2\text{O} \end{aligned}} \right\} (6.31)$$

The exhaust losses are relatively small, so that very little energy is wasted at the exit.

The static pressure loss at each flow is the sum of the individual pressure losses, which are shown in Tables 6.1 and 6.1A.

Table 6.1 is used to plot the chassis air flow impedance curve, which is shown in Figure 6.20. The static pressure curve for two different vane axial flow fans are superimposed on the same curve, using the manufacturers data. Both fans have 400 cycle motors in similar 2 in diameter housings.

Table 6.1 Static Pressure Loss for Different Air Flow Rates through Electronic Box in English Units

Station Point	Pressure Loss (in H ₂ O)		
	10 cfm	20 cfm	30 cfm
1	0.0386	0.1544	0.3470
2	0.0340	0.1360	0.3060
3	0.0287	0.1150	0.2590
4	0.0406	0.1620	0.3650
5	0.0032	0.0128	0.0288
6	0.0032	0.0128	0.0288 ← Exhaust
Total in H ₂ O	0.1483	0.5930	1.3346

Table 6.1A Static Pressure Loss for Different Air Flow Rates through Electronic Box in Metric Units

Station Point	Pressure Loss (cm H ₂ O)		
	4719 $\frac{\text{cm}^3}{\text{sec}}$	9438 $\frac{\text{cm}^3}{\text{sec}}$	14,157 $\frac{\text{cm}^3}{\text{sec}}$
1	0.0980	0.3920	0.8820
2	0.0860	0.3450	0.7770
3	0.0729	0.2920	0.6570
4	0.1030	0.4110	0.9270
5	0.0081	0.0325	0.0731
6	0.0081	0.0325	0.0731 ← Exhaust
Total cm H ₂ O	0.3761	1.5050	3.3892

Fan A is a 3 phase 15,500 rpm, 25 watt motor. Fan B is a single phase 11,000 rpm 18 watt motor which speeds up at high altitudes [44].

If fan A is used, the air flow rate through the chassis will be about 30 cfm with a constant cfm flow rate at all altitudes. If fan B is used, the sea level air flow rate will be 23 cfm. Fan B, however, is a variable speed fan which increases in speed as the altitude is increased (Rotron trade name is Altivar). This type of fan is usually very desirable for cooling electronic boxes at high altitudes where the air density is reduced [44].

The air weight flow through the electronic box can be determined from the average air density (ρ) in the box, using Eq. 6.32, which is obtained from the gas laws [45–47].

$$\rho = \frac{P}{RT} \quad (6.32)$$

where $P = 14.7 \text{ lb/in}^2$ sea level pressure (English units)
 $P = 1034.4 \text{ g/cm}^2$ sea level pressure (metric units)
 $R = 53.3 \text{ ft/}^\circ\text{R}$ gas constant for air (English units)
 $R = 2924 \text{ cm/}^\circ\text{K}$ gas constant for air (metric units)

$$t_{\text{av}} = \frac{131 + 160}{2} = 145.5^\circ\text{F} = 63^\circ\text{C} \text{ (average temperature)}$$

$$T = 460 + 145.5 = 605.5^\circ\text{R} \text{ (absolute air temperature)}$$

$$\rho = \frac{(14.7 \text{ lb/in}^2)(144 \text{ in}^2/\text{ft}^2)}{(53.3 \text{ ft/}^\circ\text{R})(605.5^\circ\text{R})} = 0.0656 \frac{\text{lb}}{\text{ft}^3} \text{ (English units)} \quad (6.33)$$

The weight flow is determined from Eq. 6.34.

$$W = \rho F \quad (6.34)$$

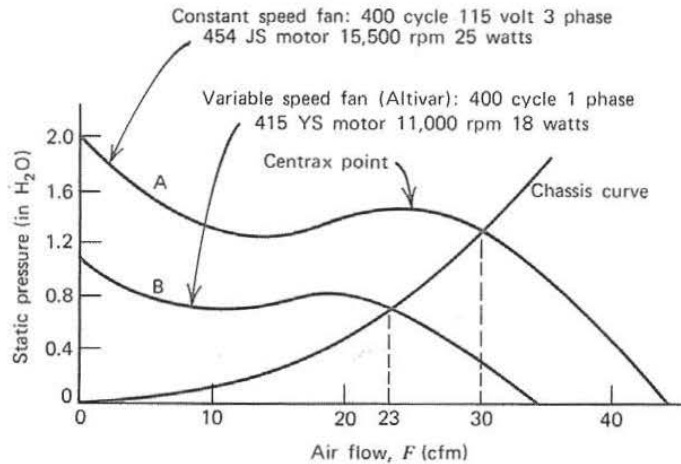


Figure 6.20 Chassis air flow impedance curve superimposed on two different vane axial fan curves with different speeds (both fans have same 2 in diameter outer housing).

Using fan A, where the air flow $F = 30$ cfm, the weight flow is

$$W = \left(0.0656 \frac{\text{lb}}{\text{ft}^3} \right) \left(30 \frac{\text{ft}^3}{\text{min}} \right) = 1.97 \frac{\text{lb}}{\text{min}} \quad (6.35)$$

Using fan B, where the air flow is 23 cfm, the weight flow is:

$$W = \left(0.0656 \frac{\text{lb}}{\text{ft}^3} \right) \left(23 \frac{\text{ft}^3}{\text{min}} \right) = 1.51 \frac{\text{lb}}{\text{min}} \quad (6.36)$$

Equation 6.8 showed that the minimum required cooling air flow rate is 1.35 lb/min. Since fans A and B can both supply more than the minimum required flow rate, either fan will be acceptable.

6.8 HOLLOW CORE PCBs

Cooling air is often contaminated with moisture, which can bridge printed circuits and electrical connector contacts. This can result in a high resistance path across two conductors, which can affect the electrical operation of high impedance circuits. Many of the new specifications for advanced electronic systems will, therefore, not permit cooling air to come into direct contact with the electronic components or circuits. Under these circumstances, the fan cooling air is usually ducted through the walls of the chassis, which have been made into heat exchangers. Plug-in types of printed circuit boards (PCBs) with metal cores of strips, similar to those shown in Figures 4.1 and 4.2, and then used to conduct away the heat.

The length of the heat flow path from the PCB components to the cooling air in the chassis walls, as shown in Figure 4.2, is quite long. This may result in a large temperature rise. If a hollow core can be provided for the cooling air between each circuit board, with the components on the outside, the length of the heat flow path from the component to the cooling air will be sharply reduced. This will reduce the temperature rise to the components, so that they will run cooler and last longer.

Electrical lead wires must not extend into the cooling air ducts in the hollow cores, so that components must be lap soldered to the circuit boards. Existing flow soldering and wave soldering equipment, normally used for high volume production of printed circuits, could not be used to assemble lap soldered components.

The most difficult task associated with plug-in hollow core PCBs is the air seal. A low resistance air flow path must be provided to the PCB, with a seal that can be engaged and disengaged many times with very little air leakage.

Figure 6.21*a*, *b*, and *c* shows some different types of air seals which have been used successfully for plug-in types of hollow PCBs. Figure 6.21*a* relies on tapered PCB sides to provide a seal with a soft gasket interface. Figure 6.21*b* uses an overlap interface, with a gasket that seals the outer surfaces at the extreme edges of the PCB. In the foregoing two designs, the inlet cooling air plenum and the exhaust air plenum are part of the chassis. In Figure 6.21*c*, the inlet cooling air slot and the exhaust air slot are part of the PCB itself. The PCBs are stacked together and clamped using four axial clamp bars, located at the four corners. This stacking arrangement forms a

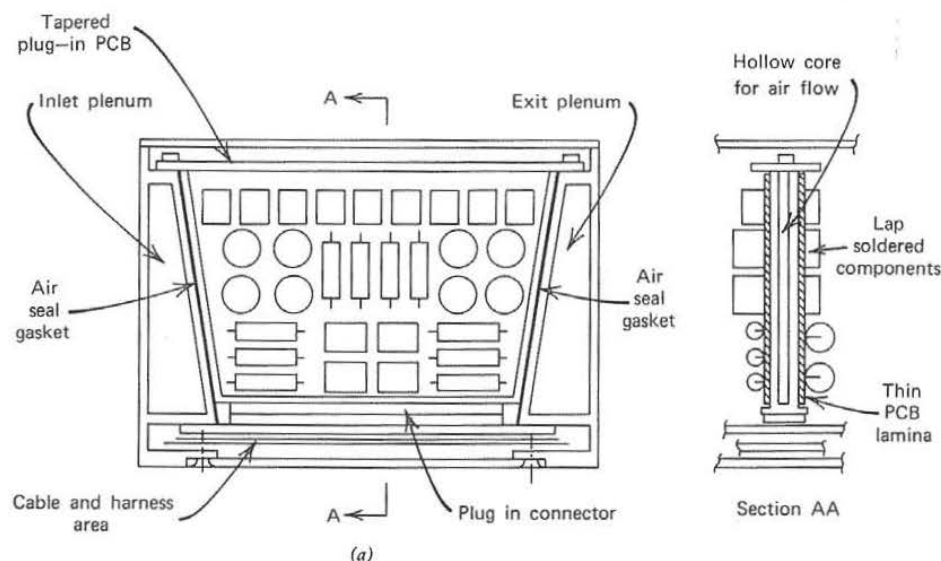


Figure 6.21 Hollow core air cooled circuit boards.

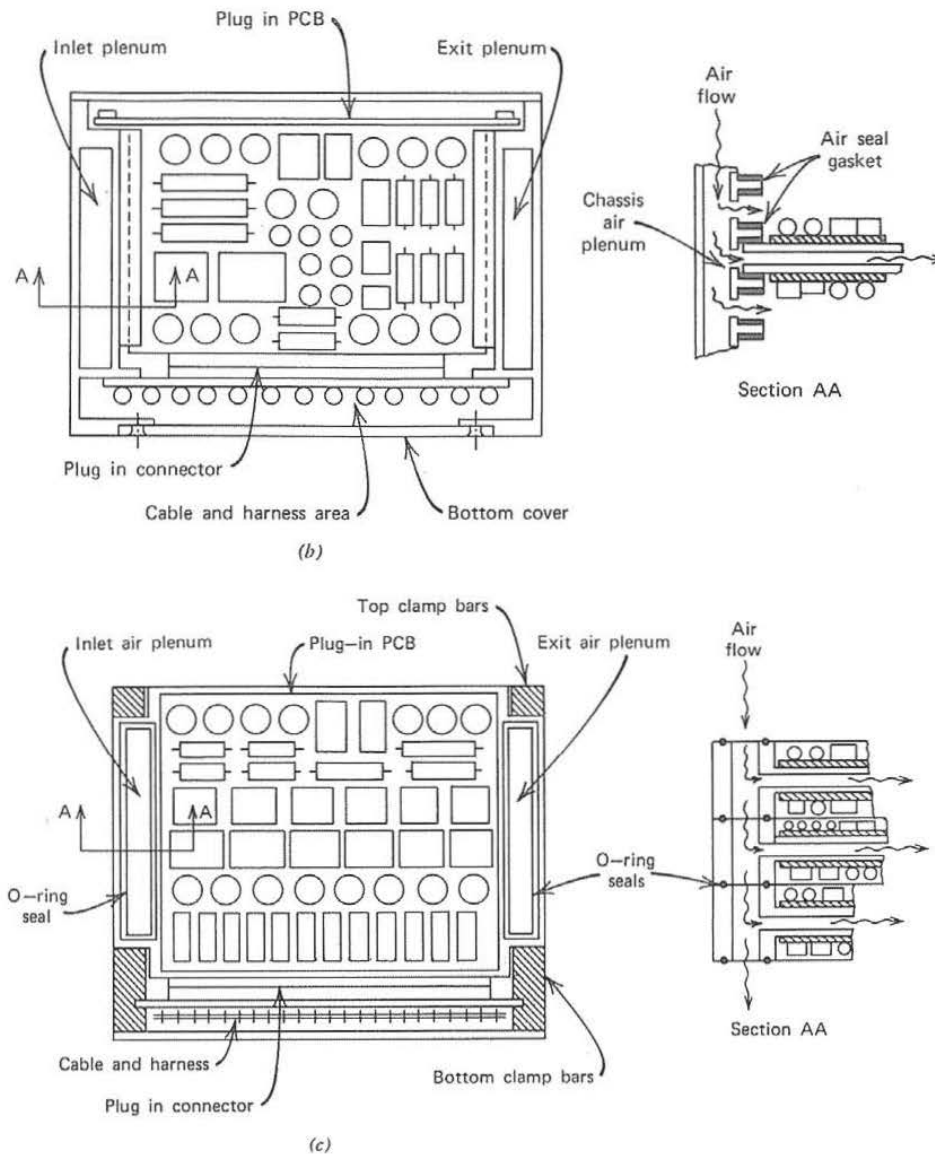


Figure 6.21 (Continued)

cooling air plenum out of the slots on each side of the PCB. An O-ring seal is used around the perimeter of each side slot to reduce the air leakage [48].

Equation 6.9 can still be used to determine the forced convection coefficient h_c for the hollow core PCB. If a slot width of 0.050 in (0.127 cm) is used for the hollow core PCB, the average forced convection coefficient shown in Eq. 6.15 can be doubled to about 12 Btu/hr ft² °F.

Extreme care must be exercised if the hollow core PCBs will be operating in a vibration environment. The thin epoxy fiberglass circuit boards will

have a low resonant frequency, which will result in large vibration displacement amplitudes and rapid fatigue failures [1].

To reduce the failure rate and to provide a long fatigue life, stiffening ribs should be provided in the hollow core air space between the circuit boards. If more stiffening is required, external stiffening ribs can be added to the outside surface of the circuit boards between the electronic components.

6.9 COOLING AIR FANS FOR ELECTRONIC EQUIPMENT

Many different types of fans are available for cooling electronic equipment. These can generally be broken down into four major types: propeller, tube axial, vane axial, and centrifugal blowers. These fans can be single phase, three phase, 60 cycles, 400 cycles, 800 cycle ac/dc, constant speed, variable speed, and flow rates from 1 cfm to several thousand cfm [44].

Propeller fans can be obtained in a large variety of sizes, with different blade shapes and arrangements. Large propeller fans may be belt driven from an electric motor, while small fans may have the propeller fan attached directly to the electric motor, as shown in Figure 6.22. This type of fan can normally provide a static pressure of about 0.30 in (0.76 cm) of water at 0 cfm, or no delivery. Propeller fans are well suited for flushing operations through large racks or cabinets, which require large volumes or air at relatively low static pressures.

Tube axial fans have their motors mounted in the fan hub. This results in a fan with an axial depth that is much shorter than a propeller fan but which has a greater static pressure capability than a propeller fan of the same diameter. This type of fan, which is shown in Figure 6.23, is normally used where low noise, long life, and low cost are important. Typical applications are in areas that might use propeller fans, but where the depth is the limiting factor and the air flow requirements exceed those of a flushing operation [44].

Figure 6.24 shows a small vane axial fan which operates at a high rotational speed and provides static pressures of more than 4 in (10.16 cm) of water at zero delivery. Fans of this type are usually used for cooling electronic assemblies with high heat densities, which must operate in severe thermal and vibration environments. They generally operate at a high frequency of 400 Hz, and are available with variable speed [44] motors, which



Figure 6.22 Typical small propeller fan.

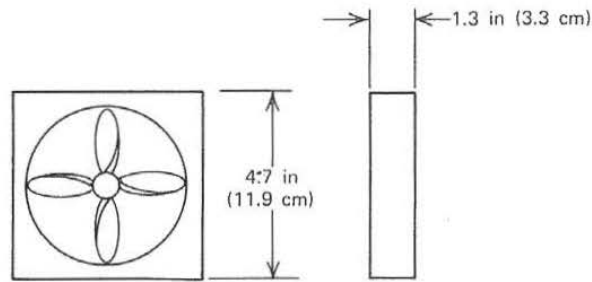


Figure 6.23 Typical flat tube axial fan that has a rated life of more than 10 years of continuous duty at 20°C.

speed up at high altitudes to provide extra cooling with low density air. These high speed fans generate a high level of acoustic noise, so that mufflers are recommended for fans that will be used in areas adjacent to working personnel.

The pressure flow characteristics of these axial flow fans, with high blade angles, produces a region of discontinuity which corresponds to a stalling condition on the blade airfoil. It is generally not advisable to operate fans in this area (which is sometimes called the centrax point) or at lower flow volumes, as shown in Figure 6.25. Operation should take place in the area to the right of the centrax point.

When the system operates to the left of the centrax point, pressure surges can occur. For example, the static pressure at points A and B in Figure 6.25 are the same, but the cfm flow conditions are not. Therefore, the flow conditions through the box can switch back and forth between points A and B. If the higher flow at point B is required, the reduced flow at point A may produce overheating in the electronics.

If the box impedance curve falls to the left of the centrax point, it is usually possible to bleed air from the fan with shims or holes. This will move the box impedance curve to the right of the centrax point.

The fan will then be delivering an excess cfm flow rate, so that some air will be thrown away. Since the fan will be operating to the right side of the

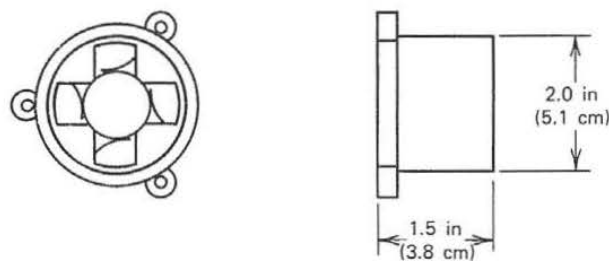


Figure 6.24 High speed, small size vane axial fan.

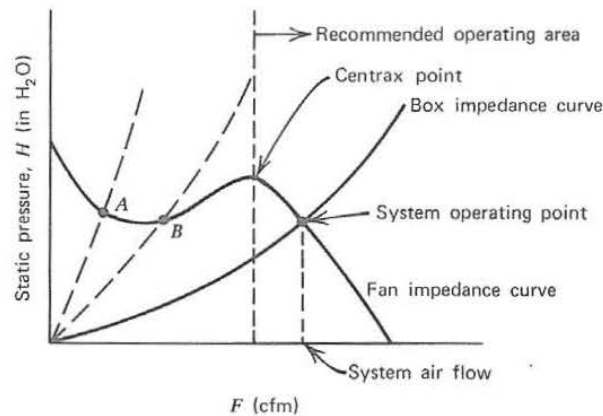


Figure 6.25 Recommended operating area for axial flow fans.

centrax point, its operation will be stable and not be subjected to pressure surges. Tests should be run to ensure that the proper amount of air is still flowing through the box for cooling.

High static pressures and high air flow rates can be obtained with low rotational speeds using centrifugal blowers (sometimes called squirrel cage blowers). Three types of blades are commonly used: backward curved, radial or paddle blade, and forward curved. A typical centrifugal blower is shown in Figure 6.26.

The highest pressures for a given propeller diameter and speed are usually developed by the forward curved fans. However, a relatively small increase in volume flow may cause a considerable increase in the power required.

Backward curved fans made with air foil section blades have produced efficiencies well above 80%. These fans are often referred to as nonoverloading fans, because they work well over a large range of volume flow.

6.10 AIR FILTERS

Electronic equipment may be required to operate around heavy machinery or in dusty areas where the air can pick up oil, lint, and other impurities.

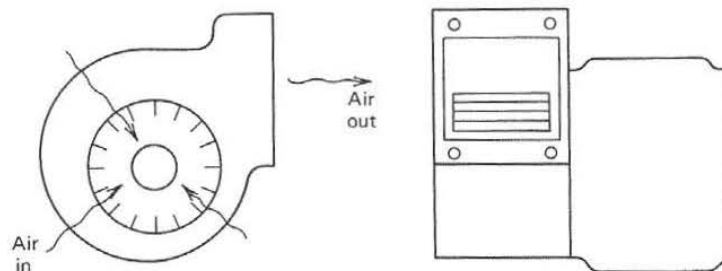


Figure 6.26 Centrifugal (squirrel cage) fan.

If these contaminants are permitted to enter the cooling air passages, they might plug up the openings and reduce the air flow, causing overheating of the electronics. To prevent this from happening, air filters are often used.

The pressure drop across an air filter can be very large if the air velocity through the filter is too high. To keep the pressure drop to a minimum, the air velocity through the filter should be less than about 250 ft/min (127 cm/sec). A lower air velocity will reduce the pressure drop through the filter even more, but will require a large filter surface area. This is desirable if there is sufficient room available for the larger filter.

Some filter manufacturers recommend a minimum air flow velocity of about 200 ft/min (102 cm/sec) through their filters. This velocity is supposed to make the suspended particles strike the filter with sufficient force to capture the particle. A lower velocity through their filter might permit some particles to change their direction so that they pass through larger openings in a filter that is not too dense.

6.11 CUTOFF SWITCHES

When cooling air is required to provide a safe operating temperature for electronic equipment, some means should be provided to turn the equipment off when the cooling air flow is interrupted, to prevent damage to the equipment. The required air flow may not be supplied because of a fan failure, power failure, clogged ducts, or clogged filters.

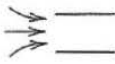
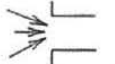
An air flow interlock switch is convenient for turning off the power in electronic systems that are not required to operate in severe vibration, shock, or acceleration environments. This device has an air flow sensor that extends into the air stream. If the air flow falls below a certain critical level, the sensor shuts off the main power to prevent the electronics from overheating.

Electronic equipment that must operate in severe vibration, shock, or acceleration environments can use miniature thermal cutoff switches. These switches should be mounted adjacent to the hottest components to prevent them from overheating. They can be purchased to open at virtually any temperature, and the cutoff point is accurate to about $\pm 2^{\circ}\text{C}$.

6.12 STATIC PRESSURE LOSS TABLES AND CHARTS

Static pressure losses for various flow relations and geometry changes are shown in terms of the velocity head for different structures and conditions in Tables 6.2, 6.3, and 6.4 [40, 41, 69].

Table 6.2 Entry Losses for Different Shapes

Type of Opening	Description	Number of Velocity Heads (H_p) Lost
	Plain duct end	0.93
	Flanged duct end	0.49
	Well-rounded entry	0.04
	Protruding entry	2.70

6.13 HIGH ALTITUDE CONDITIONS

Electronic equipment is often required to operate at high altitudes, without conditioned cooling air. For fan cooled systems, the ambient air within the electronic equipment bay is normally used as the air source.

At high altitudes, the air density is very low. The weight flow of cooling air may be so sharply reduced that the equipment can be damaged as a result of overheating. If the system is protected with a thermal switch, it will simply shut the equipment off.

Many types of fans are available for cooling electronic systems. Some fans will speed up as the air density is reduced, so that they will deliver more cfm and more weight flow at the higher altitudes [44, 49].

Constant speed fans, either single phase or three phase, will not change their speeds very much. They will run at approximately the same speed, and deliver approximately the same cfm, regardless of the altitude. The weight flow the fan delivers will change as the density or altitude changes. A constant speed fan will, therefore, deliver a different weight flow for different altitudes. As the weight flow through the electronic box changes, the temperatures within the box will also change.

As the altitude (or pressure) changes, the cfm flow through the box will remain the same if a constant speed fan is used. When the altitude is increased, the air density is decreased and therefore the weight flow is decreased. There is less resistance to the flow through the box, so the box flow impedance curve goes down, as shown in Figure 6.27 [42, 44, 49].

With the reduced density at high altitudes, the ability of the fan to draw against a high static pressure is reduced. The static pressure curve for the

Table 6.3 Turn Losses in Elbows.

Round Elbows

A diagram of a round elbow. It shows a 90-degree bend in a pipe. The radius of the bend is labeled R with a dimension line. The diameter of the pipe is labeled D with a dimension line across the straight section.

Number of Diameters, R	Number of Velocity Heads (H_v) Lost
2.75	0.20
2.50	0.22
2.25	0.24
2.00	0.27
1.75	0.32
1.50	0.39
1.25	0.55

A diagram of a square elbow. It shows a 90-degree bend in a square duct. The radius of the bend is labeled R with a dimension line. The diameter (width) of the duct is labeled D with a dimension line across the straight section.

A diagram of a rectangular elbow. It shows a 90-degree bend in a rectangular duct. The width of the duct is labeled W with a dimension line. The height of the duct is labeled D with a dimension line.

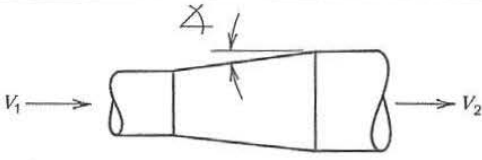
Square and Rectangular Elbows
Number of Velocity Heads (H_v) Lost

Number of Diameters, R	Aspect Ratio, W/D					
	0.25	0.5	1.0	2.0	3.0	4.0
0.0 (Mitre)	1.50	1.32	1.15	1.04	0.92	0.86
0.5	1.36	1.21	1.05	0.95	0.84	0.79
1.0	0.45	0.28	0.21	0.21	0.20	0.19
1.5	0.28	0.18	0.13	0.13	0.12	0.12
2.0	0.24	0.15	0.11	0.11	0.10	0.10
3.0	0.24	0.15	0.11	0.11	0.10	0.10

fan therefore goes down, as shown in Figure 6.27. The rate of reduction for the box impedance curve is the same as the rate of reduction for the fan static pressure curve, so the two curves cross at the same cfm flow point. As a result, the cfm flow through the box remains approximately constant as the altitude changes.

The static pressure drop through the box at the altitude condition can be determined from the static pressure drop through the box at sea level

Table 6.4 Expansion and Contraction Losses.

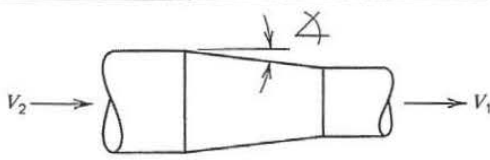


$$H_L = a \left(\frac{V_1 - V_2}{4005} \right)^2$$

$$V = \text{velocity} = \frac{\text{ft}}{\text{min}}$$

Expansion Loss

Taper Angle (deg) α	Loss Constant a
3 $\frac{1}{2}$	0.22
5	0.28
10	0.44
15	0.58
20	0.72
25	0.87
30	1.00
Over 30	1.00



Contraction Loss

Taper Angle (deg) α	Loss Constant a
5	0.05
10	0.06
15	0.08
20	0.10
25	0.11
30	0.13
45	0.20
60	0.30

conditions if the altitude and sea level densities are known. The relation is linear as shown in Eq. 6.37 [42, 44, 49].

$$\Delta P_{\text{alt}} = \Delta P_{\text{sl}} \left(\frac{\rho_{\text{alt}}}{\rho_{\text{sl}}} \right) \quad (6.37)$$

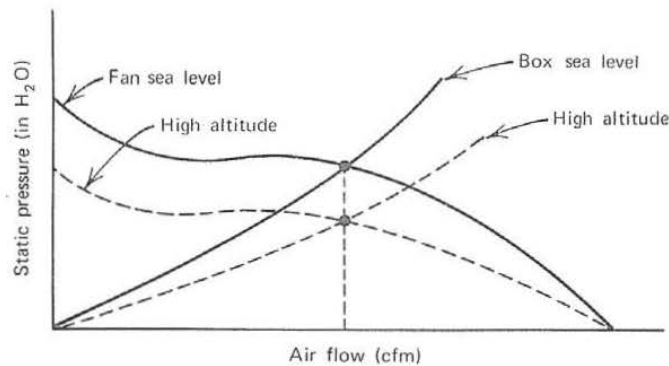


Figure 6.27 Effects of density (or altitude) on fan performance.

6.14 SAMPLE PROBLEM—FAN COOLED BOX AT 30,000 FEET

Determine if the fan cooled box shown in Figure 6.13 is capable of operating at an altitude of 30,000 ft, using a constant speed fan delivering 30 cfm, as shown in Figure 6.20. The total power dissipation of the box is 165 watts, or 563 Btu/hr. This includes the power dissipation of 25 watts for the fan. The ambient air temperature in the equipment bay is expected to be 104°F (40.0°C) according to MIL-E-5400, the electronic equipment specification for class 1X equipment.

SOLUTION

The average density of the cooling air passing through the box is not known because the exit air temperature is not known. Therefore, assume a temperature rise and determine the average air density. Compute the temperature rise based upon the power dissipation and compare it with the assumed value. Correct it if necessary, and repeat the process until reasonably good correlation is obtained. Start by assuming a temperature rise of 66°F (36.6°C). The average air temperature is then

$$t_{av} = \frac{t_{in} + t_{out}}{2} = \frac{104 + 170}{2} = 137^{\circ}\text{F}$$

The air pressure at 30,000 ft is obtained from Table 6.5 as 629.6 lb/ft². Now the air density is obtained from Eq. 6.32.

$$\rho = \frac{P}{RT} = \frac{629.6}{(53.3)(460 + 137)} = 0.0198 \frac{\text{lb}}{\text{ft}^3} \quad (6.38)$$

Table 6.5 Changes of Temperature and Pressure with Altitude [37, 38]

Temperature and Pressure as Functions of Geometric Altitude: English Units		
Altitude (ft)	Temperature (°R)	Pressure (P , lbf ft ⁻²)
-16,500	577.58	3.6588 + 3
-15,000	572.22	3.5462
-10,000	554.37	3.0020
- 5,000	536.52	2.5277
0	518.69	2.1162
5,000	500.86	1.7609
10,000	483.04	1.4556
15,000	465.23	1.1948
20,000	447.43	9.7327 + 2
25,000	429.64	7.8633
30,000	411.86	6.2966 + 2
35,000	394.08	4.9934
40,000	389.99	3.9312
45,000	389.99	3.0945
50,000	389.99	2.4361
55,000	389.99	1.9180
60,000	389.99	1.5103
65,000	389.99	1.1893
70,000	389.99	9.3672 + 1
75,000	389.99	7.3784
80,000	389.99	5.8125 + 1
85,000	394.32	4.5827
90,000	402.48	3.6292
95,000	410.64	2.8878
100,000	418.79	2.3085
105,000	426.94	1.8536
110,000	435.09	1.4947
115,000	443.23	1.2102
120,000	451.37	9.8372 + 0
125,000	459.50	8.0267
130,000	467.63	6.5735 + 0
135,000	475.76	5.4025
140,000	483.88	4.4552
145,000	492.00	3.6862
150,000	500.11	3.0597
155,000	508.22	2.5475
160,000	508.79	2.1247
165,000	508.79	1.7723
170,000	508.79	1.4784
175,000	508.79	1.2334
180,000	497.49	1.0272 + 0

Table 6.5 Continued

Temperature and Pressure as Functions of Geometric Altitude: English Units		
Altitude (ft)	Temperature (°R)	Pressure (<i>P</i> , lbf ft ⁻²)
185,000	485.36	8.5160 – 1
190,000	473.24	7.0278
195,000	461.12	5.7713
200,000	449.00	4.7151
210,000	424.79	3.0955
220,000	400.60	1.9835
230,000	376.44	1.2368
240,000	352.30	7.4774 – 2
250,000	328.20	4.3640

The average air weight flow through the box is determined from Eq. 6.34.

$$W = \rho F = \left(0.0198 \frac{\text{lb}}{\text{ft}^3} \right) \left(30 \frac{\text{ft}^3}{\text{min}} \right) = 0.594 \frac{\text{lb}}{\text{min}} \quad (6.39)$$

The temperature rise of the cooling air as it passes through the box is determined from Eq. 6.7.

$$\Delta t_A = \frac{Q}{WC_p} = \frac{563 \text{ Btu/hr}}{(0.594 \text{ lb/min})(60 \text{ min/hr})(0.24 \text{ Btu/lb } ^\circ\text{F})} = 65.8^\circ\text{F} \quad (6.40)$$

The assumed temperature rise of 66°F agrees well with the calculated value of 65.8°F, so that the assumed value is accurate.

The temperature rise across the convection air film to the surface of the electronic components for the altitude condition is still determined from Eq. 5.7, using Eqs. 6.9 through 6.18 for the required information.

Substitute Eq. 6.39 into Eq. 6.12.

$$G = \frac{W}{A} = \frac{(0.594 \text{ lb/min})(60 \text{ min/hr})}{(0.00625 \text{ ft}^2)(7 \text{ ducts})} = 814 \frac{\text{lb}}{\text{hr ft}^2} \quad (6.41)$$

Substitute into Eq. 6.11 for the Reynolds number:

$$N_R = \frac{(814 \text{ lb/hr ft}^2)(0.0165 \text{ ft})}{0.050 \text{ lb/ft hr}} = 269 \text{ (dimensionless)} \quad (6.42)$$

Substitute into Eq. 6.10 for the J factor:

$$J = \frac{6}{(N_R)^{0.98}} = \frac{6}{(269)^{0.98}} = 0.0249 \quad (6.43)$$

Substitute into Eq. 6.9 for the forced air convection coefficient at 30,000 ft (physical properties of air are obtained from Figure 6.28 for an average air temperature of 137°F).

$$h_c = \frac{(0.0249)(0.24)(814)}{[(0.24)(0.05)/0.017]^{0.666}} = 6.13 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} \quad (6.44)$$

Notice that although there has been a relatively large reduction in the Reynolds number, the forced convection coefficient at the altitude condition, Eq. 6.44, has not changed much from the forced convection coefficient at sea level, as shown in Eq. 6.15. This is quite typical for laminar flow conditions. The forced convection coefficient there depends much more on the hydraulic diameter than on the Reynolds number. A smaller hydraulic diameter increases the value of the coefficient (see Figure 6.38).

The temperature rise across the convection film to the components is determined from Eq. 5.7.

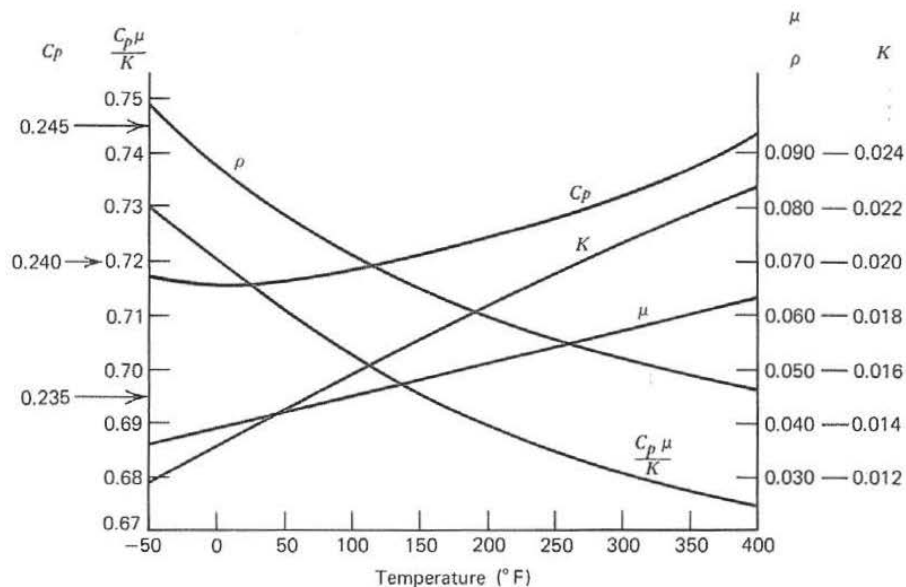


Figure 6.28 Physical properties of dry air at 1 atm (English units).

Given $Q = 20 \text{ watts} = 68.3 \text{ Btu/hr}$ (PCB heat)

$$h_c = 6.13 \text{ Btu/hr ft}^2 \text{ }^\circ\text{F} \text{ (ref. Eq. 6.44)}$$

$$A = \frac{(1.3)(8)(9)}{144} = 0.65 \text{ ft}^2 \text{ [effective area 1 PCB (including back side)]}$$

$$\Delta t_B = \frac{Q}{h_c A} = \frac{68.3}{(6.13)(0.65)} = 17.1^\circ\text{F} \quad (6.45)$$

The component hot spot temperature is determined from Eq. 6.17.

Given $t_{in} = 104^\circ\text{F}$

$$\Delta t_A = 65.8^\circ\text{F} \text{ (ref. Eq. 6.40)}$$

$$\Delta t_B = 17.1^\circ\text{F} \text{ (ref. Eq. 6.45)}$$

$$t_s = 104 + 65.8 + 17.1 = 186.9^\circ\text{F} \quad (6.46)$$

Since this temperature is less than 212°F (100°C), the design is satisfactory for the altitude condition.

If a slightly cooler operating box is desired, the cooling air fan can be used as an exhaust fan. The 25 watts of heat from the fan will not be added to the inlet cooling air. This will reduce the cooling air temperature about 10°F and will also reduce the component hot spot temperature about 10°F , as shown in Eq. 6.47.

$$\Delta T = \frac{Q}{WC_p} = \frac{(25 \text{ watts})(3.413 \text{ Btu/hr/watt})}{(0.594 \text{ lb/min})(60 \text{ min/hr})(0.24 \text{ Btu/lb } ^\circ\text{F})} = 10^\circ\text{F} \quad (6.47)$$

The pressure drop through the box at the 30,000 ft altitude condition can be determined from Eq. 6.37.

Given $\Delta P_{sl} = 1.30 \text{ in H}_2\text{O}$ (ref. Figure 6.20 sea level for constant speed fan)

$$\rho_{alt} = 0.0198 \text{ lb/ft}^3 \text{ 30,000 ft (ref. Eq. 6.38)}$$

$$\rho_{sl} = 0.0704 \text{ lb/ft}^3 \text{ sea level and } 104^\circ\text{F}$$

$$\Delta P_{alt} = 1.30 \left(\frac{0.0198}{0.0704} \right) = 0.366 \text{ in H}_2\text{O} \quad (6.48)$$

6.15 OTHER CONVECTION COEFFICIENTS

Convection coefficients can become quite high for turbulent air flow conditions through ducts. These ducts may be air passages that are formed

between plug-in PCBs or ducts that deliver air between two electronic boxes. The convection coefficient is shown in Eq. 6.49.

$$h = 0.0198 \left(\frac{K}{D} \right) (N_R)^{0.8} \quad (6.49)$$

Equation 6.49 can be used to find the forced convection coefficient for a pressurized enclosed duct. For example, consider a flat rectangular duct with a cross section $a = 3.0$ in, $b = 0.50$ in, and many feet long. Air at 20 psig and 150°F flows at 200 cfm. Find the convection heat transfer coefficient. The physical properties of air are needed. These are shown in Figure 6.28.

$$D = \frac{2ab}{a + b} \text{ (hydraulic diameter)}$$

$$D = \frac{2(3.0)(0.50)}{3.0 + 0.50} = 0.857 \text{ in} = 0.0714 \text{ ft}$$

$$\rho = \frac{P}{RT} \text{ (air density; ref. Eq. 6.32)}$$

$$\rho = \frac{(20.0 + 14.7)(144 \text{ in}^2/\text{ft}^2)}{(53.3)(460 + 150)^\circ\text{R}} = 0.154 \frac{\text{lb}}{\text{ft}^3}$$

$$A = \frac{(3.0)(0.5) \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 0.0104 \text{ ft}^2 \text{ (cross section area)}$$

$$v = \frac{\text{cfm}}{A} = \frac{(200 \text{ ft}^3/\text{min})(60 \text{ min}/\text{hr})}{0.0104 \text{ ft}^2} = 1.15 \times 10^6 \frac{\text{ft}}{\text{hr}} \text{ (velocity)}$$

$$\mu = 0.055 \text{ lb/ft hr (viscosity at 34.7 psia, 150°F; ref. Figure 6.28)}$$

$$K = 0.0175 \text{ Btu/hr ft } ^\circ\text{F (conductivity at 150°F; ref. Figure 6.28)}$$

The Reynolds number is

$$N_R = \frac{VD\rho}{\mu} = \frac{(1.15 \times 10^6)(0.0714)(0.154)}{0.055} = 230,000 \text{ (dimensionless)}$$

Since N_R is well above 2300, the flow is obviously turbulent. Substitute into Eq. 6.49.

$$h = (0.0198) \left(\frac{0.0175}{0.0714} \right) (230,000)^{0.8} = 94.5 \frac{\text{Btu}}{\text{hr ft}^2 ^\circ\text{F}} \quad (6.50)$$

For the flow of gasses parallel to plane surfaces, the general heat transfer convection coefficient can be determined from Eq. 6.51.

$$h = 0.055 \left(\frac{K}{L} \right) (N_R)^{0.75} \quad (6.51)$$

The length of the surface is shown by L . This is to be limited to 2 ft, even if the length is greater than 2 ft.

When the air is blowing across a cylinder or wire, the convection coefficient can take the form shown in Eq. 6.52 [7, 8].

$$h = b \left(\frac{K}{D} \right) (N_R)^m \quad (6.52)$$

The constants b and m , which are a function of the Reynolds number, are shown in Table 6.6 for different conditions [7, 8].

The characteristic dimension D to be used in the equation for the square and diamond cross section is that of a circular cross section that has the same perimeter dimension.

For air flow over spheres, the convection heat transfer coefficient can be determined from Eq. 6.53.

$$h = 0.33 \left(\frac{K}{D} \right) (N_R)^{0.6} \quad (6.53)$$

6.16 SAMPLE PROBLEM—COOLING A TO-5 TRANSISTOR

A 2N2905 transistor (TO-5 case size) dissipates 0.25 watt under steady state power conditions in a 160°F (71°C) ambient environment. The transistor is

Table 6.6 Values to Be Used with Eq. 6.52.

Shape	Reynolds Number	b	m
	0.4–4.0	0.891	0.330
	4.0–40	0.821	0.385
	40–4,000	0.615	0.466
	4,000–40,000	0.174	0.618
	40,000–400,000	0.0239	0.805
	5,000–10,000	0.0921	0.675
	5,000–100,000	0.222	0.585

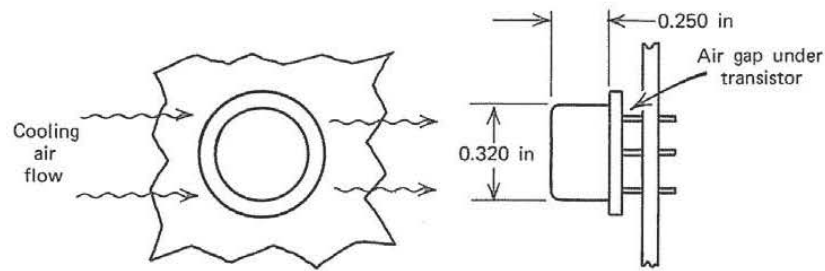


Figure 6.29 Cooling air flow over a transistor.

mounted above the circuit board as shown in Figure 6.29. Determine the case temperature when the air velocity over the transistor is 250 ft/min.

SOLUTION

The Reynolds number (N_R) must be computed first, using the following information. (Physical properties of air are determined from Figure 6.28.)

$$V = 250 \text{ ft/min} \times 60 \text{ min/hr} = 15,000 \text{ ft/hr (air velocity)}$$

$$D = \frac{0.320 \text{ in}}{12 \text{ in/ft}} = 0.0267 \text{ ft (hydraulic diameter)}$$

$$\rho = \frac{P}{RT} = \frac{(14.7 \text{ lb/in}^2)(144 \text{ in}^2/\text{ft}^2)}{(53.3 \text{ ft}^2/\text{R})(460 + 160)^\circ\text{R}} = 0.064 \frac{\text{lb}}{\text{ft}^3} \text{ (air density)}$$

$$\mu = 0.050 \text{ lb/ft hr (viscosity at } 160^\circ\text{F)}$$

$$K = 0.017 \text{ Btu/hr ft } ^\circ\text{F (thermal conductivity at } 160^\circ\text{F)}$$

Substitute into Eq. 6.11 for the Reynolds number:

$$N_R = \frac{(15000)(0.0267)(0.064)}{0.050} = 512 \text{ (dimensionless)} \quad (6.54)$$

The Reynolds number is between 40 and 4000, so that the values shown in Table 6.6 are $b = 0.615$ and $m = 0.466$. Substituting these values into Eq. 6.52 results in Eq. 6.55.

$$h = 0.615 \left(\frac{K}{D} \right) (N_R)^{0.466} \quad (6.55)$$

$$h = 0.615 \left(\frac{0.017}{0.0267} \right) (512)^{0.466} = 7.17 \frac{\text{Btu}}{\text{hr ft}^2 ^\circ\text{F}} \quad (6.56)$$

The temperature rise across the boundary layer, from the ambient air to the transistor surface, is determined from Eq. 5.7.

$$\Delta t = \frac{Q}{hA} \text{ (ref. Eq. 5.7)}$$

$$\text{Given } Q = 0.25 \text{ watt} \times 3.413 \text{ Btu/hr/watt} = 0.85 \text{ Btu/hr}$$

$$h = 7.16 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F} \text{ (ref. Eq. 6.56)}$$

A = surface area of transistor

$$= \frac{2(\pi/4)(0.32)^2 + \pi(0.32)(0.25) \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 0.00286 \text{ ft}^2$$

$$\Delta t = \frac{0.85 \text{ Btu/hr}}{(7.16 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F})(0.00286 \text{ ft}^2)} = 41.5^\circ\text{F} \quad (6.57)$$

The case temperature of the transistor becomes

$$t_{\text{case}} = 160 + 41.5 = 201.5^\circ\text{F} \quad (6.58)$$

The junction temperature of the transistor is determined using the thermal impedance from the junction to the case (Θ_{jc}) which is usually obtained from the component manufacturer. When the value of $\Theta_{jc} = 59.4^\circ\text{F/watt}$ (33°C/watt) the temperature rise from the case to the junction is:

$$\Delta t_{jc} = 59.4 \frac{^\circ\text{F}}{\text{watt}} \times 0.25 \text{ watt} = 14.8^\circ\text{F} \quad (6.59)$$

The junction temperature of the transistor can then be obtained from Eq. 6.60.

$$t_j = t_{\text{case}} + \Delta t_{jc} \quad (6.60)$$

$$t_j = 201.5 + 14.8 = 216.3^\circ\text{F} \quad (6.61)$$

6.17 CONDITIONED COOLING AIR FROM AN EXTERNAL SOURCE

Conditioned cooling air is often supplied to the electronic equipment from an external source, such as the compressor section of an aircraft engine. For example, in the F-16 airplane, the electronic equipment cooling air is obtained from the seventeenth stage of the engine compressor. The air temperature and pressure are both very high, and the air may have a high moisture content. This air is therefore throttled, cooled, and dried before it is passed through the electronic equipment.

The amount of conditioned air required to cool an electronic system will vary with the temperature of the cooling air. Less air is required when the cooling air temperature is low, and more air is required when the cooling air temperature is high. Extensive test data and analysis have shown that an efficient thermal design can be achieved if the temperature of the cooling air is not allowed to exceed 160°F (71°C) as it exits from the electronic chassis. This condition should result in a maximum component surface temperature of about 212°F (100°C) when the component mounting techniques shown in Chapter 4 are used.

Equation 6.7 can be used to establish the cooling air flow curve for an electronic box when the power dissipation of the box is known. The technique is demonstrated with a sample problem.

6.18 SAMPLE PROBLEM—GENERATING A COOLING AIR FLOW CURVE

An electronic box, with a power dissipation of 330 watts, has conditioned cooling air supplied by the airplane engine. Generate the flow curve required to maintain a box cooling air exit temperature of 160°F (71°C) when the cooling air inlet temperature varies from -65°F (-54°C) to +131°F (+55°C). The conversion tables in Section 1.11 are used to convert to English and metric units.

SOLUTION

$$Q = 330 \text{ watts} = 1126.3 \text{ Btu/hr} = 78.87 \text{ cal/sec (heat dissipation)}$$

$$C_p = 0.24 \text{ Btu/lb } ^\circ\text{F} = 0.24 \text{ cal/g } ^\circ\text{C (specific heat)}$$

$$\Delta t = 160^\circ\text{F} - t_{\text{in}} \text{ (English)} \quad (71^\circ\text{C} - t_{\text{in}} \text{ metric})$$

Start with an air inlet temperature of -65°F for English units, which corresponds to -54°C for metric units.

$$\Delta t = 160^\circ\text{F} - (-65)^\circ\text{F} = 225^\circ\text{F (English units)}$$

$$\Delta t = 71^\circ\text{C} - (-54)^\circ\text{C} = 125^\circ\text{C (metric units)}$$

Substituting into Eq. 6.7, the cooling air weight flow is determined using English units.

$$W = \frac{1126.3 \text{ Btu/hr}}{(225^\circ\text{F})(0.24 \text{ Btu/lb } ^\circ\text{F})(60 \text{ min/hr})} = 0.347 \frac{\text{lb}}{\text{min}} \quad (6.62)$$

Using Eq. 6.7, the weight flow for metric units is

$$W = \frac{78.87 \text{ cal/sec}}{(125^\circ\text{C})(0.24 \text{ cal/g } ^\circ\text{C})} = 2.63 \frac{\text{g}}{\text{sec}} \quad (6.62a)$$

At an air inlet temperature of 80°F (26.6°C)

$$\Delta t = 160^\circ\text{F} - 80^\circ\text{F} = 80^\circ\text{F} \text{ (English units)}$$

$$\Delta t = 71^\circ\text{C} - 26.6^\circ\text{C} = 44.4^\circ\text{C} \text{ (metric units)}$$

Substituting into Eq. 6.7 for the weight flow, we obtain

$$W = \frac{1126.3 \text{ Btu/hr}}{(80^\circ\text{F})(0.24)(60 \text{ min/hr})} = 0.978 \frac{\text{lb}}{\text{min}} \text{ (English units)} \quad (6.63)$$

$$W = \frac{78.87 \text{ cal/sec}}{(44.4^\circ\text{C})(0.24)} = 7.40 \frac{\text{g}}{\text{sec}} \text{ (metric units)} \quad (6.63a)$$

At an air inlet temperature of 131°F (55°C):

$$\Delta t = 160^\circ\text{F} - 131^\circ\text{F} = 29^\circ\text{F} \text{ (English units)}$$

$$\Delta t = 71^\circ\text{C} - 55^\circ\text{C} = 16^\circ\text{C} \text{ (metric units)}$$

Substituting into Eq. 6.7 yields

$$W = \frac{1126.3 \text{ Btu/hr}}{(29^\circ\text{F})(0.24)(60 \text{ min/hr})} = 2.70 \frac{\text{lb}}{\text{min}} \text{ (English units)} \quad (6.64)$$

$$W = \frac{78.87 \text{ cal/sec}}{(16^\circ\text{C})(0.24)} = 20.54 \frac{\text{g}}{\text{sec}} \text{ (metric units)} \quad (6.64a)$$

When the cooling air flow rate is plotted against the cooling air temperature, the cooling air flow curve will be produced, as shown in Figure 6.30.

6.19 STATIC PRESSURE LOSSES FOR VARIOUS ALTITUDE CONDITIONS

Electronic systems are often required to operate at different altitude conditions. As the altitude varies, it is desirable to maintain a constant cooling air weight flow rate, similar to the curve shown in Figure 6.30. At high altitudes the air density is lower, so that the cooling air flow velocity must be increased to provide the same air flow weight. As the air velocity increases, however, the static pressure drop through the electronic system

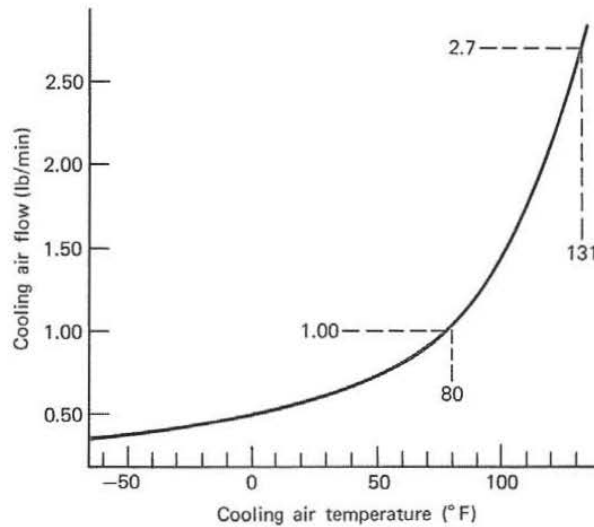


Figure 6.30 Typical cooling air weight flow curve.

also increases. It is therefore important to know how these pressure losses will change with the altitude, to ensure an adequate flow of cooling air to the electronics under all conditions.

The complexity of the static pressure drop relations for various altitude conditions can be simplified by considering the density of the cooling air at the altitude condition and comparing it with the density of air at some standard condition. The standard selected is usually based on a temperature of 59°F (15°C) and a pressure of 14.7 lb/in² (1034.4 g/cm²), where the air density is 0.0765 lb/ft³ (0.00122 g/cm³).

It is convenient to use a density ratio that is called sigma and is normally shown by the Greek letter sigma (σ), which is defined as follows:

$$\sigma = \frac{\text{air density at any condition}}{\text{air density at standard conditions}} = \frac{\rho}{\rho_{\text{std}}}$$

$$\sigma = \frac{\rho}{0.0765 \text{ lb/ft}^3} \text{ (for English units)} \quad (6.65)$$

$$\sigma = \frac{\rho}{0.00122 \text{ g/cm}^3} \text{ (for metric units)} \quad (6.65a)$$

It is desirable to work with the average density of the cooling air for high altitude conditions. At high altitudes, the outside air pressure and density are very low, so that the cooling air velocity must be very high to provide a reasonable weight flow. High air velocities mean high flow losses, so that a high pressure must be used to push the cooling air through the box. Under these circumstances, the inlet air density can be many times greater than

the outlet air density. If these density changes are ignored at high altitude conditions, it can lead to large pressure drop errors. The average air density within the box is therefore used, to improve the accuracy of the calculations.

The density ratio (σ) can be written in another way by using the density relation obtained from the gas law relation shown in Eq. 6.32. Then substituting into Eq. 6.65, we obtain

$$\sigma = \frac{P_1/R_1T_1}{P_2/R_2T_2} = \frac{\frac{P_1}{(53.3 \text{ ft/}^\circ\text{R}) T_1}}{\frac{14.7 \text{ lb/in}^2}{(53.3 \text{ ft/}^\circ\text{R})(460 + 59)^\circ\text{R}}} = 35.3 \frac{P_1}{T_1} \text{ (English units)}$$

$$\sigma = \frac{\frac{P_1}{(2924 \text{ cm/}^\circ\text{K}) T_1}}{\frac{1034.4 \text{ g/cm}^2}{(2924 \text{ cm/}^\circ\text{K})(273 + 15)^\circ\text{K}}} = 0.279 \frac{P_1}{T_1} \text{ (metric units)}$$

The inlet and the outlet air pressures should be averaged to obtain the average value for the electronic box in English units and metric units. The resulting equation is convenient for working with electronic boxes at high altitude conditions.

$$\sigma = \frac{35.3}{2} \left(\frac{P_{\text{in}}}{T_{\text{in}}} + \frac{P_{\text{out}}}{T_{\text{out}}} \right) \text{ (English units)} \quad (6.66)$$

where P = pressure = lb/in²
 T = absolute temperature (460 + °F) = °R
 $P_{\text{in}} = P_{\text{out}} + \Delta P$ = lb/in² (inlet pressure)
 P_{out} = outside pressure = lb/in²

$$\sigma = \frac{0.279}{2} \left(\frac{P_{\text{in}}}{T_{\text{in}}} + \frac{P_{\text{out}}}{T_{\text{out}}} \right) \text{ (metric units)} \quad (6.66a)$$

where P = pressure = g/cm²
 T = absolute temperature (273 + °C) = °K
 $P_{\text{in}} = P_{\text{out}} + \Delta P$ = g/cm² (inlet pressure)
 P_{out} = outside pressure = g/cm²

The value of σ is usually multiplied by the static pressure drop ΔP and plotted against the cooling air weight flow on a log-log plot. This results in a straight line, as shown in Figure 6.31.

For fully turbulent flow conditions, the slope of the $\sigma \Delta P$ line on a log-log plot is 2.0. For laminar flow conditions, the slope will usually vary from about 1.2 to about 1.7, depending upon the severity of the restrictions in the cooling air flow path.

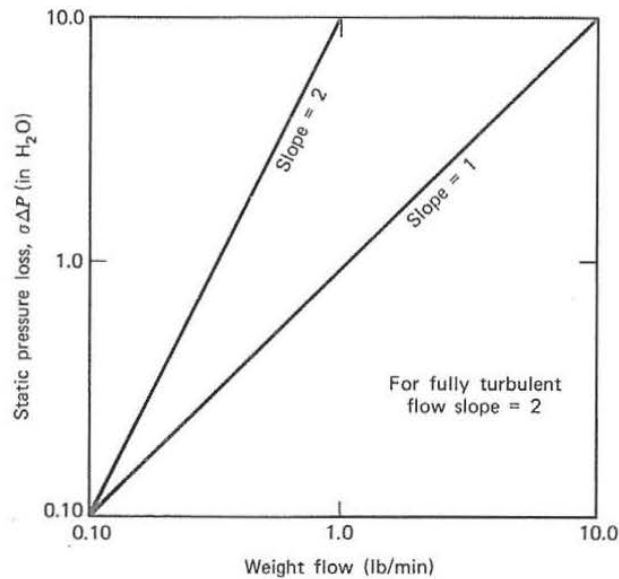


Figure 6.31 Typical $\sigma \Delta P$ pressure loss curves for electronic boxes.

Once the $\sigma \Delta P$ curve has been established for a particular electronic box, it is very easy to determine the pressure drop through the box for any altitude and for any cooling air flow condition.

6.20 SAMPLE PROBLEM—STATIC PRESSURE DROP AT 65,000 FEET

An electronic box with a power dissipation of 330 watts has conditioned cooling air supplied (by the airplane turbine) with a weight flow rate as shown in Figure 6.30. This flow rate will provide a cooling air exit temperature of 160°F (71°C) for any inlet air temperature shown in the curve. Test data show that the static pressure drop through the box is 2.0 in of water (5.08 cm of water) when the air inlet temperature is 80°F (26.6°C) and the cooling air flow rate is 1.0 lb/min (7.56 g/sec).

- 1 Determine the static pressure drop through the box at an altitude of 65,000 ft (19,812 m) with an air flow rate of 1.0 lb/min at 80°F. The outside air pressure at this altitude is 0.83 lb/in² (58.4 g/cm²).
- 2 Determine the static pressure drop through the box at sea level and at an altitude of 65,000 ft when the air inlet temperature is 131°F (55°C) and the cooling air flow rate is 2.7 lb/min (20.43 g/sec).

SOLUTION

For the solution of part 1 of the problem, the value of σ is first computed for the sea level condition. Equations 6.65 and 6.65a will give about the same results as Eqs. 6.66 and 6.66a because the 2.0 in pressure drop only represents a $2 \times 0.036 = 0.072$ lb/in² pressure drop. This is small compared to the atmospheric pressure of 14.7 lb/in², so that P_{in} is virtually the same as P_{out} in Eq. 6.66.

The average air density for the 80°F (26.6°C) sea level condition is computed from Eq. 6.32, using English units first. The average temperature in the box must be used.

$$t_{av} = \frac{t_{in} + t_{out}}{2} = \frac{80 + 160}{2} = 120^\circ\text{F (English units)}$$

$$t_{av} = \frac{26.6 + 71}{2} = 48.8^\circ\text{C (metric units)}$$

In English units:

$$\rho = \frac{(14.7 \text{ lb/in}^2)(144 \text{ in}^2/\text{ft}^2)}{(53.3 \text{ ft/}^\circ\text{R})(460 + 120)^\circ\text{R}} = 0.0685 \frac{\text{lb}}{\text{ft}^3} \quad (6.67)$$

In metric units:

$$\rho = \frac{1034.4 \text{ g/cm}^2}{(2924 \text{ cm/}^\circ\text{K})(273 + 48.8)^\circ\text{K}} = 0.001099 \frac{\text{g}}{\text{cm}^3} \quad (6.67a)$$

Substitute into Eq. 6.65 for the sea level value of σ using English units.

$$\sigma = \frac{\rho}{0.0765} = \frac{0.0685}{0.0765} = 0.895 \text{ (dimensionless)} \quad (6.68)$$

Substitute into Eq. 6.65a for the sea level value of σ using metric units.

$$\sigma = \frac{\rho}{0.001224} = \frac{0.001099}{0.001224} = 0.897 \text{ (dimensionless)} \quad (6.68a)$$

The value of σ is multiplied by the static pressure drop (ΔP) head of water, shown previously as 2.0 in H₂O (5.08 cm H₂O), to obtain the $\sigma \Delta P$ values of static pressure drop:

$$\sigma \Delta P = (0.895)(2.0 \text{ in H}_2\text{O}) = 1.79 \text{ in H}_2\text{O (English units)} \quad (6.69)$$

$$\sigma \Delta P = (0.897)(5.08 \text{ cm H}_2\text{O}) = 4.55 \text{ cm H}_2\text{O (metric units)} \quad (6.69a)$$

The values shown above are valid only for a cooling air flow rate of 1.0 lb/min (7.56 g/sec) and an air inlet temperature of 80°F (26.6°C). However, this $\sigma \Delta P$ value is exactly the same *at any altitude condition, as long as the air flow rate and temperature stay the same.*

The static pressure drop at an altitude of 65,000 ft, with the 1.0 lb/min flow rate, can be determined if the value of σ can be obtained for the high altitude condition. Using Eqs. 6.66 and 6.66a, the following information is required.

<i>Item</i>	<i>English Units</i>	<i>Metric Units</i>
P_{out}	0.83 lb/in ²	58.4 g/cm ²
T_{out}	460 + 160 = 620°R	273 + 71 = 344°K
T_{in}	460 + 80 = 540°R	273 + 26.6 = 299.6°K
P_{in}	0.83 lb/in ² + ΔP	58.4 g/cm ² + ΔP

The static pressure drop ΔP through the electronic box is not known for the high altitude condition, but its value is required. Therefore, a value is assumed and the calculation is made. The assumed value is then compared with the calculated value. If they are not close, another guess is made and the calculation is repeated. Close agreement can usually be obtained with two or three iterations. Considering English units first, assume a static pressure drop, ΔP , of 22 in H₂O.

$$\Delta P = 22 \text{ in H}_2\text{O} \times 0.036 \frac{\text{lb/in}^2}{\text{in H}_2\text{O}} = 0.792 \text{ lb/in}^2$$

$$P_{in} = 0.83 \text{ lb/in}^2 + 0.792 \text{ lb/in}^2 = 1.622 \text{ lb/in}^2$$

Substitute into Eq. 6.66 for the σ value at 65,000 ft with a flow rate of 1.0 lb/min at 80°F.

$$\sigma_{alt} = \frac{35.3}{2} \left(\frac{1.622}{540} + \frac{0.83}{620} \right) = 0.0766 \quad (6.70)$$

For metric units, assume a static pressure drop ΔP of 56 cm H₂O.

$$\Delta P = 56 \text{ cm H}_2\text{O} = 56 \text{ g/cm}^2$$

$$P_{in} = 58.4 \text{ g/cm}^2 + 56 \text{ g/cm}^2 = 114.4 \text{ g/cm}^2$$

Substitute into Eq. 6.66a for the σ value at 19,812 m with a flow rate of 7.56 g/sec at 26.6°C.

$$\sigma_{alt} = \frac{0.279}{2} \left(\frac{114.4}{299.6} + \frac{58.4}{344} \right) = 0.0769 \quad (6.70a)$$

The static pressure drop at the altitude condition is determined from Eq. 6.71.

$$\Delta P_{\text{alt}} = \frac{\sigma \Delta P}{\sigma_{\text{alt}}} \quad (6.71)$$

For English units, substitute Eqs. 6.69 and 6.70 into Eq. 6.71.

$$\Delta P_{\text{alt}} = \frac{1.79}{0.0766} = 23.4 \text{ in H}_2\text{O} \quad (6.72)$$

For metric units, substitute Eqs. 6.69a and 6.70a into Eq. 6.71.

$$\Delta P_{\text{alt}} = \frac{4.556}{0.0769} = 59.2 \text{ cm H}_2\text{O} \quad (6.72a)$$

A pressure drop of 22 in H₂O (and 56 cm H₂O) was assumed for the box at an altitude of 65,000 ft (19,812 m), with a flow rate of 1.0 lb/min (7.56 g/sec) at 80°F (26.6°C). The calculated value turned out to be 23.4 in H₂O (and 59.2 cm H₂O) on the first try. A second iteration would show that the correct value is closer to 23 in H₂O (58.4 cm H₂O).

For the solution to part 2 of the problem, the slope of the $\sigma \Delta P$ curve on a log-log plot similar to Figure 6.31 must be known. For fully turbulent flow systems the $\sigma \Delta P$ slope is 2.0. This is the theoretical maximum that can be obtained with an air flow system. In actual systems, however, this slope is seldom obtained. The actual slope for a typical box will generally have a range from about 1.2 for a box with very few widely spaced PCBs to about 1.8 for a box with heat exchanger side walls containing 20 wavy fins per inch.

When the electronic box has a $\sigma \Delta P$ slope of 1.4, for example, the pressure drop at any cooling air weight flow condition can be determined from the general definition of the slope, as shown in Figure 6.32 and Eq. 6.73.

$$\text{Slope } S = \frac{\Delta Y}{\Delta X} = \frac{\log Y_2 - \log Y_1}{\log X_2 - \log X_1} \quad (6.73)$$

Common logarithms to the base 10 or natural logarithms to the base e can be used for Eq. 6.73. Considering common logarithms to the base 10 with English units, the $\sigma \Delta P$ static pressure drop through the box for a flow rate of 2.7 lb/min is computed.

$$1.4 = \frac{\log \sigma \Delta P_2 - \log 1.79}{\log 2.7 - \log 1.0} = \frac{\log \sigma \Delta P_2 - 0.2528}{0.4313 - 0}$$

$$\log \sigma \Delta P_2 = 1.4(0.4313) + 0.2528 = 0.8566$$

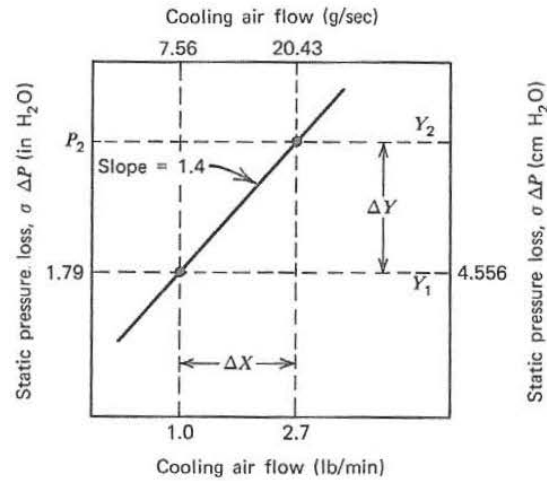


Figure 6.32 $\sigma \Delta P$ cooling curve with a slope of 1.4.

Take the antilog of both sides:

$$\sigma \Delta P_2 = 7.188 \text{ in H}_2\text{O} \quad (6.74)$$

Considering common logarithms to the base 10 with metric units, the $\sigma \Delta P$ static pressure drop through the box for a flow rate of 20.43 g/sec is computed.

$$1.4 = \frac{\log \sigma \Delta P_2 - \log 4.556}{\log 20.43 - \log 7.56} = \frac{\log \sigma \Delta P_2 - 0.6586}{1.3102 - 0.8785}$$

$$\log \sigma \Delta P_2 = 1.4 (0.4317) + 0.6586 = 1.2629$$

Take the antilog of both sides:

$$\sigma \Delta P_2 = 18.32 \text{ cm H}_2\text{O} \quad (6.74a)$$

Before the static pressure drop through the box at sea level conditions, with a flow rate of 2.7 lb/min, can be computed from the $\sigma \Delta P$ values, the average air temperature and average air density must be computed. In English units:

$$t_{av} = \frac{t_{in} + t_{out}}{2} = \frac{131 + 160}{2} = 145.5^\circ\text{F}$$

$$\rho = \frac{(14.7)(144) \text{ lb/ft}^2}{(53.3 \text{ ft/}^\circ\text{R})(460 + 145.5)^\circ\text{R}} = 0.0656 \frac{\text{lb}}{\text{ft}^3} \quad (6.75)$$

In metric units:

$$t_{av} = \frac{55 + 71}{2} = 63^{\circ}\text{C}$$

$$\rho = \frac{1034.4 \text{ g/cm}^3}{(2924 \text{ cm}^3/\text{K})(273 + 63)^{\circ}\text{K}} = 0.001053 \frac{\text{g}}{\text{cm}^3} \quad (6.75a)$$

Substitute Eq. 6.75 into Eq. 6.65 for English units:

$$\sigma = \frac{\rho}{0.0765} = \frac{0.0656}{0.0765} = 0.857 \text{ (dimensionless)} \quad (6.76)$$

Substitute Eq. 6.75a into Eq. 6.65a for metric units:

$$\sigma = \frac{\rho}{0.001224} = \frac{0.00105}{0.001224} = 0.857 \text{ (dimensionless)} \quad (6.76a)$$

The sea level static pressure drop through the box with a flow rate of 2.7 lb/min (20.43 g/sec) is now computed from Eqs. 6.74 and 6.75 in English units.

$$\Delta P_{sl} = \frac{\sigma \Delta P_2}{\sigma} = \frac{7.188}{0.857} = 8.39 \text{ in H}_2\text{O} \quad (6.77)$$

In metric units, with Eqs. 6.74a and 6.75a:

$$\Delta P_{sl} = \frac{\sigma \Delta P_2}{\sigma} = \frac{18.32}{0.857} = 21.38 \text{ cm H}_2\text{O} \quad (6.77a)$$

The static pressure drop through the box at an altitude of 65,000 ft, with a flow rate of 2.7 lb/min, can be calculated as before. Assume a pressure drop of 62 in H₂O to start.

$$\Delta P = 62 \text{ in H}_2\text{O} \times 0.036 \text{ (lb/in}^2\text{)/in H}_2\text{O} = 2.232 \text{ lb/in}^2$$

$$P_{in} = 0.83 \text{ lb/in}^2 + 2.232 \text{ lb/in}^2 = 3.062 \text{ lb/in}^2$$

$$P_{out} = 0.83 \text{ lb/in}^2$$

$$T_{in} = 460 + 131 = 591^{\circ}\text{R}$$

$$T_{out} = 460 + 160 = 620^{\circ}\text{R}$$

$$\sigma_{alt} = \frac{35.3}{2} \left(\frac{3.062}{591} + \frac{0.83}{620} \right) = 0.115$$

The $\sigma \Delta P$ value at 65,000 ft must be the same as the value shown by Eq.

6.74 for the 2.7 lb/min flow rate. The static pressure drop through the box at 65,000 ft is, therefore,

$$\Delta P_{\text{alt}} = \frac{\sigma \Delta P}{\sigma_{\text{alt}}} = \frac{7.188}{0.115} = 62.5 \text{ in H}_2\text{O} \quad (6.78)$$

A static pressure drop of 62 in H₂O was assumed for the 65,000 ft altitude condition for a flow rate of 2.7 lb/min. The calculated value is 62.5 in H₂O, which is close enough for these calculations and shows that the assumed value is accurate.

The force that pushes the cooling air through the electronic box is the pressure differential across the box. As this pressure difference increases, the velocity of the cooling air through the box will increase until the critical velocity is reached. Once the critical velocity is reached, any further increase in the pressure difference across the box will not produce any further increase in cooling air velocity.

Critical flow velocity will be reached when the ratio of the downstream pressure (or outside pressure) to the upstream pressure (or inside pressure) falls below the critical pressure ratio (P_{cr}) of 0.53, as shown below.

$$P_{\text{cr}} = \frac{P_{\text{out}}}{P_{\text{in}}} = 0.53 \text{ (critical pressure ratio)}$$

In the previous sample problem, this pressure ratio (R_p) at the 65,000 ft altitude is:

$$R_p = \frac{P_{\text{out}}}{P_{\text{in}}} = \frac{0.83}{3.062} = 0.27$$

Since this value is less than 0.53, it shows that the critical velocity has been reached. Therefore, the 2.7 lb/min cooling air flow rate at the 65,000 ft altitude condition can never be obtained.

6.21 TOTAL PRESSURE DROP FOR VARIOUS ALTITUDE CONDITIONS

Section 6.3 and Figures 6.7 through 6.9 show the basic relations between static pressures and velocity pressures for electronic boxes. Equation 6.3 shows that when the cooling air is forced through an electronic system, the total pressure is simply the sum of the velocity pressure and the static pressure.

$$H_t = H_v + H_s \quad (\text{ref. Eq. 6.3})$$

Cooling requirements for electronic equipment will sometimes specify the allowable pressure drop through the system in terms of the *static* pressure and sometimes in terms of the *total* pressure. Since the two pressures are different, it is necessary to understand the method for converting from one to another.

The static pressure drop through an electronic box is usually determined first. Section 6.4 shows how the static pressure losses are related to the velocity of the cooling air as it is being forced through the box.

After the static pressure losses have been determined for a box, it is necessary to compute the velocity pressure (or velocity head) for that box if the total pressure (or total head H_t) losses are required.

The velocity head H_v should be based upon the velocity of the cooling air in the most restricted flow section through the box, where the air velocity is the highest. These values can be computed from Eq. 6.2 or 6.4 for English units and Eq. 6.2a for metric units.

6.22 SAMPLE PROBLEM—TOTAL PRESSURE LOSS THROUGH AN ELECTRONIC BOX

An air flow analysis of an electronic box shows that the static pressure drop (H_s) through the box is 1.75 in of water. The most severe flow restriction through the box is at the inlet, where the cooling air velocity is 2200 ft/min. Determine the total pressure drop through the box.

SOLUTION

The velocity head at the air inlet is determined from Eq. 6.4.

$$H_v = \left(\frac{V}{4005} \right)^2 = \left(\frac{2200}{4005} \right)^2 = 0.302 \text{ in H}_2\text{O} \quad (6.79)$$

The total pressure drop through the box is determined from Eq. 6.3.

$$H_t = H_v + H_s = 0.302 + 1.75 = 2.052 \text{ in H}_2\text{O} \quad (6.80)$$

6.23 FINNED COLD PLATES AND HEAT EXCHANGERS

Cold plates and heat exchangers are often fabricated of aluminum, with many thin plate fins spaced close together. When the fins are spaced close together, there is a large increase in the forced convection heat transfer coefficient (h) and a large increase in the surface area (A) available for heat transfer. The product of $h_c \times A$ therefore increases very rapidly, which

improves the thermal efficiency of the design. This results in a lower component hot spot temperature and improves the reliability of the electronic equipment [50].

There is a practical limit to the number of thin plate fins that can be used. In a dip brazed box, the maximum number of straight fins is about 22 fins per inch. More fins will often result in plugged passages because it is difficult to clean out the dip brazing salts, which may become trapped between the fins. If eutectic bonding or fluxless brazing is used, as many as 30 fins per inch can be used without plugging passages. Increasing the number of fins, however, will increase the pressure drop through the box, which can sharply reduce the cooling air flow rate and may result in overheating of the electronics.

Extensive test data [3] and analysis have shown that the forced convection heat transfer coefficient for multiple plate fin types of cold plates and heat exchangers can be determined with Eq. 6.9 [15, 43].

$$h_c = JC_p G \left(\frac{C_p \mu}{K} \right)^{-2/3} \quad (\text{ref. Eq. 6.9})$$

The heat transfer coefficient J can be determined from the Reynolds number (N_R). For laminar flow conditions, with N_R between about 400 and 1500, the approximate J value for finned cold plates and heat exchangers is shown in Eq. 6.81, compiled from data given in [43].

$$J = \frac{0.72}{(N_R)^{0.7}} \quad (6.81)$$

The typical cross section through a multiple fin cold plate or heat exchangers is shown in Figure 6.33.

The cold plate or heat exchanger structure is often used for the side walls of electronic boxes. Printed circuit boards (PCBs) are then supported between the side walls. Copper or aluminum sheets are often laminated to the PCBs to provide a good thermal conduction heat flow path along the PCBs, as shown in Figure 4.1. The heat can then flow along the PCBs to the air cooled side walls of the chassis, which contain multiple fin heat exchangers, as shown in Figure 4.2.

Since the cooling medium used is air, it is important to bring the cooling air as close to the electronic components as possible. This will reduce the temperature rise and improve the thermal efficiency.

In many new packaging applications the cooling air is ducted directly to the circuit boards, which have hollow cores, as shown in Figure 6.21. For improved heat transfer efficiency, multiple fin heat exchangers are often added to the hollow core PCBs. The convection heat transfer coefficient, shown by Eqs. 6.9 and 6.81, can be used to determine the thermal charac-

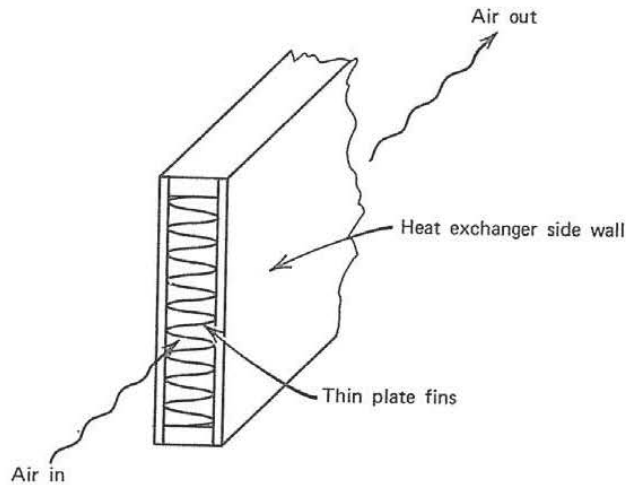


Figure 6.33 Typical cold plate or heat exchanger.

teristics of systems which have hollow core air passages through the center of the PCBs.

6.24 PRESSURE LOSSES IN MULTIPLE FIN HEAT EXCHANGERS

Multiple fin heat exchangers and cold plates require a substantial amount of energy to overcome the friction that resists the flow of cooling air through these units. When the fin density is high or when the surface finish on the fins is very rough, even more energy is required to force the air through the long, narrow tubes that are formed by the heat exchanger fins.

It is convenient to express the friction losses in terms of the height of a column of water H_L in inches (or centimeters) because the magnitudes of the numbers expressed in pounds per square inch (or grams per square centimeter) are so small.

Friction losses increase very rapidly as the velocity of the cooling air flow increases. However, the losses appear to be linear with respect to the length of the flow path through the heat exchanger.

The static head loss due to friction resulting from the flow through the multiple fin heat exchanger tubes only is shown in Eq. 6.82. Losses due to entrance resistance, turns, expansions, and contractions are not included in this equation. These other losses must be evaluated separately using techniques similar to those shown in Section 6.7, and then added to Eq. 6.82 to obtain the full static head loss for the electronic box. This is known as the Darcy Equation, and it is valid for English or metric units as long as a

consistent set of units is used. This equation can be used for laminar and turbulent flow conditions [3, 15, 17, 43, 51].

$$H_L = 4f \left(\frac{L}{D} \right) \left(\frac{V^2}{2g} \right) \quad (6.82)$$

where V = flow velocity = ft/sec or cm/sec
 L = length of air flow path = in, ft, cm
 D = hydraulic diameter of one duct = in, ft, cm
 g = acceleration of gravity
 $g = 32.2 \text{ ft/sec}^2 = 386 \text{ in/sec}^2 = 980 \text{ cm/sec}^2$
 f = friction factors, dimensionless
 $f = 16/N_R$ = Fanning friction factor for laminar flow in round tubes (ref. Table 6.7)*

Equation 6.82 will result in the static pressure loss in terms of the height of a column of air. Sometimes this equation is written with the density (ρ) included, as shown in Eq. 6.83. When the density is included, it changes the units of the losses from a height dimension to a pressure dimension, lb/in² or lb/ft² for English units and g/cm² for metric units.

$$\Delta P_L = 4f \left(\frac{L}{D} \right) \left(\frac{\rho V^2}{2g} \right) \quad (6.83)$$

Sometimes it is more convenient to obtain the static head loss in terms of $\sigma \Delta P$ directly in inches of water, as shown by Eq. 6.84. English units must be used with the constant of 0.226; then the weight flow $W = \text{lb/min}$ and the cross section area $A = \text{in}^2$.

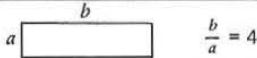
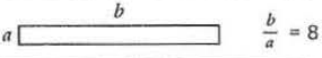
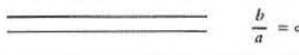
$$\sigma \Delta P = 0.226(4f) \left(\frac{L}{D} \right) \left(\frac{W}{A} \right)^2 \quad (6.84)$$

where $\sigma \Delta P$ = static pressure loss = in H₂O
 f = Fanning friction factor = dimensionless (ref. Table 6.7)
 L = length of heat exchanger = in
 D = hydraulic diameter, one duct = in
 W = cooling air weight flow, one duct = lb/min
 A = cross section flow area, one duct = in²

The friction factor f is usually determined from test data for laminar and turbulent flow conditions and for different surface roughness. In the laminar flow region, for smooth wall surfaces, the friction factor can generally be

* Do not become confused with the Hagen-Poiseuille equation for laminar flow in round tubes, shown in Eqs. 10.3 and 10.4. In the Hagen-Poiseuille equation, the laminar flow friction factor $f = 64/N_R$.

Table 6.7 Laminar Flow Friction Factors (f) for Fully Developed Velocity and Temperature Profiles in Smooth Pipes with Different Cross Sections.

Geometry	Friction Factor
	$f = \frac{13.3}{N_R}$
	$f = \frac{14.2}{N_R}$
	$f = \frac{16}{N_R}$
	$f = \frac{18.3}{N_R}$
	$f = \frac{20.5}{N_R}$
	$f = \frac{24}{N_R}$

related directly to the Reynolds number (N_R). Table 6.7 shows some typical friction factors for different cross sections that have smooth walls [43]. From Kays & London, *Compact Heat Exchangers*, published by McGraw-Hill Book Co.

6.25 FIN EFFICIENCY FACTOR

Fins are used extensively with forced convection cooling systems to increase the amount of heat that is removed. The fins extend into the air stream to expose a large amount of surface area, which improves the heat transfer. One of the factors involved in the transfer is the temperature difference between the cooling air and the surface of the fin. More heat can be transferred when there is a large temperature difference.

Sometimes the fin is relatively long, so that there is a temperature gradient along the fin. The base of the fin may be substantially hotter than the tip of the fin, as shown in Figure 6.34.

The tip of the fin should be nearly as hot as the base of the fin, in order to transfer the maximum amount of heat. This requires a large fin thickness (δ), a short length (d), and a high thermal conductivity (K), to reduce the thermal gradient along the fin [53].

To compensate for the difference in the amount of heat that is transferred from a fin with a nonuniform temperature, a fin efficiency factor (η) is used. This fin efficiency is defined as the ratio of heat that is actually transferred from the fin to the heat that would be transferred at the maximum uniform

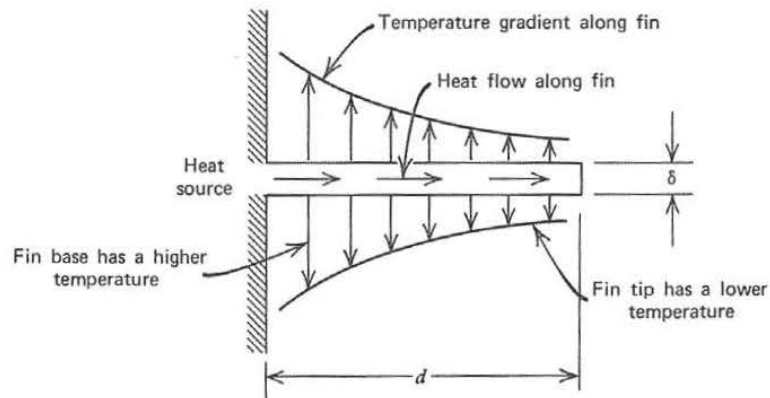


Figure 6.34 Heat flow from a cooling fin.

temperature condition, where the tip of the fin is at the same temperature as the base. The fin efficiency factor is shown in Eq. 5.21 [3, 15, 17].

$$\eta = \frac{\tanh md}{md} \quad (\text{ref. Eq. 5.21})$$

where $\tanh$ = hyperbolic tangent

$$m = \sqrt{2h_c/K\delta}$$

h_c = convection coefficient

K = fin thermal conductivity

δ = fin thickness

d = effective height of fin from base

Sometimes fins extend between two surfaces, which are both heat sources. When this condition exists, the height of the fin from the base is half of the total height, as shown in Figure 6.35.

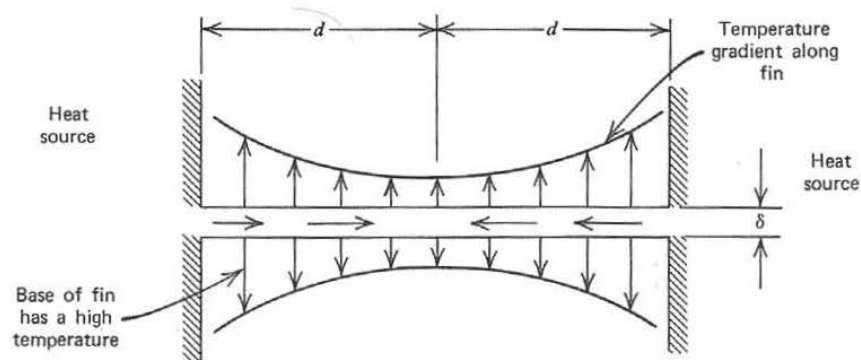


Figure 6.35 Cooling fin with two heat sources on opposite ends.

Figure 6.36 shows an electronic box with multiple fin air cooled heat exchangers in the side panels.

The fin efficiency for a single fin, as shown in Figure 6.35, can be determined from Figure 5.7 when the values of m and d are known.

The use of the heat exchanger convection coefficient equation, pressure drop equations, and fin efficiency equation can be demonstrated with the use of a sample problem.

6.26 SAMPLE PROBLEM—HOLLOW CORE PCB WITH A FINNED HEAT EXCHANGER

An electronic system has many discrete, hybrid, and LSI components that will dissipate a large amount of heat. The size of the electronic box will permit the use of several 4.0 in (10.16 cm) $\times$ 6.2 in (15.75 cm) plug-in PCBs. The system must be capable of operating in a class 1X environment, using forced convection cooling at sea level conditions, where the surrounding ambient air temperature is 131°F (55°C). A maximum steady state power

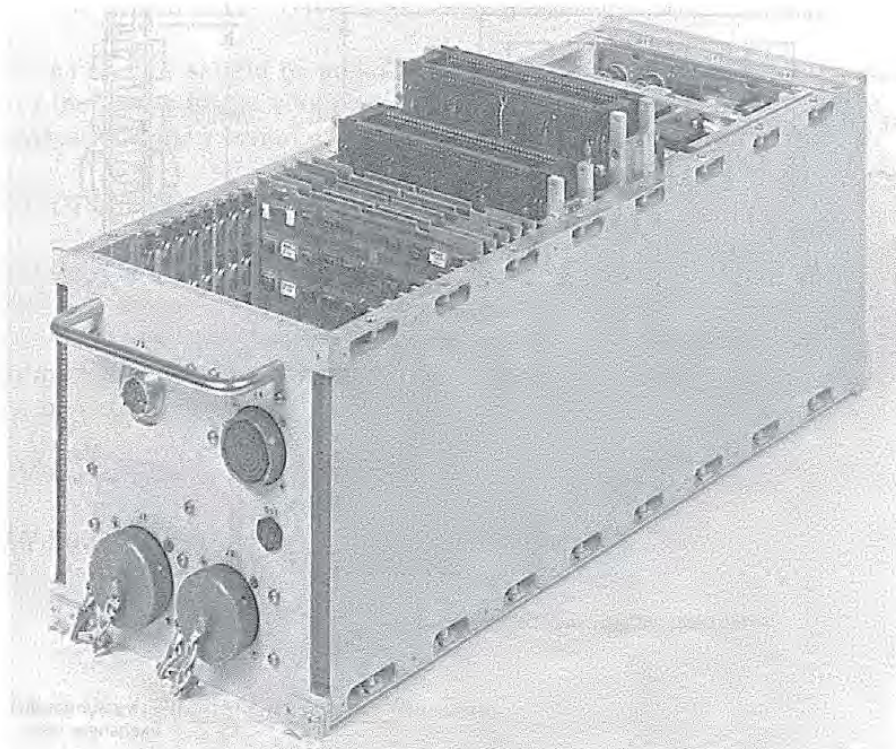


Figure 6.36 Electronic box with plug-in modules that are cooled by conduction of heat to air cooled epoxy bonded heat exchangers in the side walls. (Courtesy Litton Stystems, Inc.)

dissipation of 50 watts is expected for each PCB under certain conditions, with a relatively uniform heat distribution across the component mounting surface. The maximum allowable component mounting surface temperature is 212°F (100°C). Because of the high power dissipation, a hollow core construction is proposed for the PCBs.

Determine the following:

- 1 The minimum cooling air weight flow required through each PCB.
- 2 The heat transfer convection coefficient for a multiple fin heat exchanger at the center of the PCB.
- 3 The maximum component case or surface mounting temperature.
- 4 The static pressure drop through the PCB heat exchanger.

A sketch of the proposed PCB is shown in Figure 6.37.

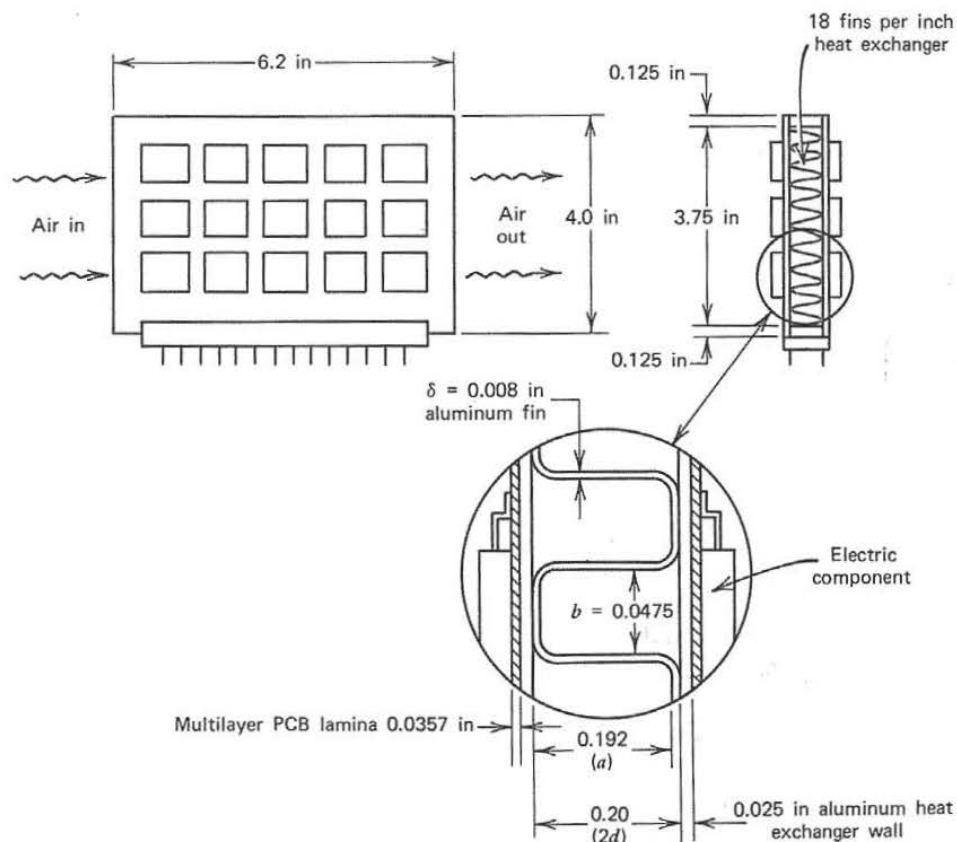


Figure 6.37 Hollow core PCB with a dip brazed aluminum heat exchanger at the center and electronic components mounted on both sides.

SOLUTION FOR PART 1

An efficient cooling system can be designed by providing an exit air temperature of 160°F (71.1°C). The information required is as follows (see Section 6.17):

$$\begin{aligned} Q &= 50 \text{ watts} = 170.6 \text{ Btu/hr} = 11.95 \text{ cal/sec (power dissipation)} \\ C_p &= 0.24 \text{ Btu/lb } ^\circ\text{F} = 0.24 \text{ cal/g } ^\circ\text{C (specific heat)} \\ \Delta t &= 160^\circ\text{F} - 131^\circ\text{F} = 29^\circ\text{F} \quad \text{or} \quad 71.1^\circ\text{C} - 55^\circ\text{C} = 16.1^\circ\text{C} \end{aligned} \quad (6.85)$$

Substitute into Eq. 6.7 to determine the minimum required cooling air weight flow through each PCB in English units.

$$W_{\min} = \frac{170.6 \text{ Btu/hr}}{(29^\circ\text{F})(0.24 \text{ Btu/lb } ^\circ\text{F})(60 \text{ min/hr})} = 0.408 \frac{\text{lb}}{\text{min}} \quad (6.86)$$

In metric units:

$$W_{\min} = \frac{11.95 \text{ cal/sec}}{(16.1^\circ\text{C})(0.24 \text{ cal/g } ^\circ\text{C})} = 3.09 \frac{\text{g}}{\text{sec}} \quad (6.86a)$$

An extra 10% should be added to the foregoing cooling air flow rates to cover the possibility of cooling air leakage. A good seal is often difficult to maintain for a long period of time in a severe service environment.

SOLUTION FOR PART 2

The forced convection coefficient for the PCB heat exchanger shown in Figure 6.37 is determined from Eq. 6.9, and the J factor is determined from Eq. 6.81. The Reynolds number (N_R) is also required, and it can be obtained with the use of Eq. 6.11. But first, the cooling air flow through one air duct must be determined.

$$\text{number of ducts} = 18 \text{ fins/in} \times 3.75 \text{ inch} = 67.5 \text{ ducts}$$

Air flow through one duct, in English units:

$$W = \frac{0.408 \text{ lb/min}}{67.5 \text{ ducts}} = 0.00604 \frac{\text{lb}}{\text{min}} \text{ per duct}$$

Air flow through one duct, in metric units:

$$W = \frac{3.09 \text{ g/sec}}{67.5 \text{ ducts}} = 0.0458 \frac{\text{g}}{\text{sec}} \text{ per duct}$$

Hydraulic diameter of one duct ($a = 0.192$ in, $b = 0.0475$ in):

$$D = \frac{4 \times \text{area}}{\text{perimeter}} = \frac{4ab}{2a + 2b} = \frac{2ab}{a + b} \quad (6.87)$$

$$D = \frac{2(0.192)(0.0475)}{0.192 + 0.0475} = 0.0761 \text{ in} = 0.00635 \text{ ft} = 0.193 \text{ cm}$$

Cross section area of one duct:

$$A = ab = (0.192)(0.0475) = 0.00912 \text{ in}^2 = 0.0000633 \text{ ft}^2 = 0.0588 \text{ cm}^2$$

The weight velocity flow through one duct is determined from Eq. 6.12.

$$G = \frac{0.00604 \text{ lb/min} \times 60 \text{ min/hr}}{0.0000633 \text{ ft}^2} = 5725 \frac{\text{lb}}{\text{hr ft}^2} \text{ (English units)}$$

$$G = \frac{0.0458 \text{ g/sec}}{0.0588 \text{ cm}^2} = 0.779 \frac{\text{g}}{\text{sec cm}^2} \text{ (metric units)}$$

The average cooling air temperature is:

$$t_{av} = \frac{t_{in} + t_{out}}{2} = \frac{131^\circ\text{F} + 160^\circ\text{F}}{2} = 145.5^\circ\text{F} = 63.0^\circ\text{C}$$

The physical properties of air (ref. Figure 6.28; conversion between English units and metric units is obtained from Section 1.11) [7, 8, 37, 38, 52].

$$K = 0.017 \text{ Btu/hr ft } ^\circ\text{F} = 0.0000702 \text{ cal/sec cm } ^\circ\text{C} \text{ (conductivity)}$$

$$C_p = 0.24 \text{ Btu/lb } ^\circ\text{F} = 0.24 \text{ cal/g } ^\circ\text{C} \text{ (specific heat)}$$

$$\mu = 0.050 \text{ lb/ft hr} = 0.000206 \text{ g/cm sec} \text{ (viscosity)}$$

$$\rho = 0.0656 \text{ lb/ft}^3 = 0.00105 \text{ g/cm}^3 \text{ (density)}$$

Substitute into Eq. 6.11 for the Reynolds number. In English units:

$$N_R = \frac{(5725 \text{ lb/hr ft}^2)(0.00635 \text{ ft})}{0.050 \text{ lb/ft hr}} = 727 \text{ (dimensionless)} \quad (6.88)$$

In metric units:

$$N_R = \frac{(0.779 \text{ g/sec cm}^2)(0.193 \text{ cm})}{0.000206 \text{ g/cm sec}} = 729 \text{ (dimensionless)} \quad (6.88a)$$

Substitute into Eq. 6.81 for the Colburn J factor.

$$J = \frac{0.72}{(N_R)^{0.7}} = \frac{0.72}{(727)^{0.7}} = 0.00715 \text{ (dimensionless)}$$

Substitute into Eq. 6.9 for the heat exchanger forced convection coefficient, in English units.

$$h_c = \frac{(0.00715)(0.24)(5725)}{[(0.24)(0.050)/0.017]^{0.666}} = 12.4 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} \quad (6.89)$$

In metric units:

$$h_c = \frac{(0.00715)(0.24)(0.779)}{[(0.24)(0.000206)/0.0000702]^{0.666}} = 0.00168 \frac{\text{cal}}{\text{sec cm}^2 \text{ } ^\circ\text{C}} \quad (6.89a)$$

The forced convection coefficient will be slightly higher at the cooling air entrance section of the heat exchanger because of the extra turbulence that is developed. This small increase is conservatively ignored here.

The fin efficiency factor shown in Eq. 5.21 must be considered because there will be a temperature gradient along the heat exchanger fins. Since there are electronic components mounted on both sides of the PCB, there will be a heat input to the heat exchanger fins from two different surfaces, as shown in Figure 6.35.

Given $\delta = 0.008 \text{ in} = 0.000666 \text{ ft} = 0.0203 \text{ cm}$ (fin thickness)
 $K = 99 \text{ Btu/hr ft } ^\circ\text{F} = 0.409 \text{ cal/sec cm } ^\circ\text{C}$ 6061 (aluminum fin)
 $d = 0.20 \text{ in}/2 = 0.10 \text{ in} = 0.00833 \text{ ft} = 0.254 \text{ cm}$ (note, heat in from both sides)
 $h_c = 12.4 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F} = 0.00168 \text{ cal/sec cm}^2 \text{ } ^\circ\text{C}$ (convection coeff.)

Substitute into Eq. 5.21 for English units, with dimensions in feet.

$$m = \sqrt{\frac{2(12.4)}{99(0.000666)}} = 19.4$$

$$md = 19.4(0.00833) = 0.1616$$

In metric units:

$$m = \sqrt{\frac{2(0.00168)}{(0.409)(0.0203)}} = 0.636$$

$$md = (0.636)(0.254) = 0.1615$$

Substitute into Eq. 5.21 for fin efficiency.

$$\eta = \frac{\tanh 0.1615}{0.1615} = \frac{0.16011}{0.1615} = 99.1\%$$

The corrected value of the forced convection coefficient in English units is

$$h_c = 12.4(0.991) = 12.3 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} \quad (6.90)$$

In metric units:

$$h_c = 0.00168(0.991) = 0.00166 \frac{\text{cal}}{\text{sec cm}^2 \text{ } ^\circ\text{C}} \quad (6.90a)$$

The value of the forced convection coefficient (h_c) is sharply influenced by the hydraulic diameter in the laminar flow regions. With low values of cooling air weightflow velocities, the convection coefficient for a given hydraulic diameter does not show much change. As the hydraulic diameter decreases, the forced convection coefficient increases rapidly, as shown in Figure 6.38. As the fluid flow increases and the turbulent region is approached, the convection coefficient increases very rapidly [70].

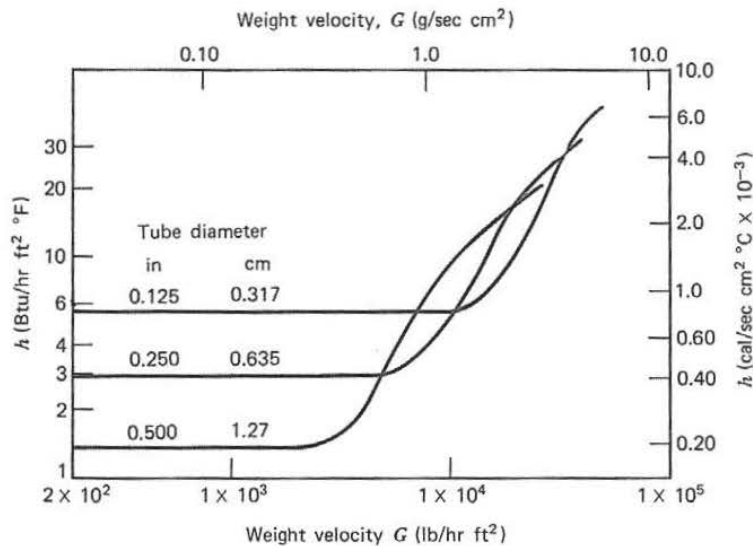


Figure 6.38 Variation of forced convection coefficient with hydraulic diameter and flow rate. (Ref: General Electric Heat Transfer Data Book 1975)

SOLUTION FOR PART 3

The maximum component case temperatures will occur on the components that are mounted closest to the cooling air exhaust end of the heat exchanger.

A large number of interconnections are required for the circuits, so multilayer circuit laminars will be required. Interconnections between layers are made with plated throughholes, which are filled with solder or epoxy to improve the mechanical strength of the interconnection.

The thickness of the multilayer circuit lamina from the component to the heat exchanger will be about 0.0357 in (0.0907 cm). This thickness must be closely controlled because the component heat will flow through the multilayer circuit lamina to the air cooled heat exchanger at the center of the PCB. A thicker circuit lamina will have a higher thermal resistance, which will increase the temperature rise across the lamina.

Plated throughholes will tend to reduce the thermal resistance through the circuit lamina. An enlarged view through the circuit lamina, showing the heat flow path, is shown in Figure 6.39.

Some of the component heat may be lost by external convection and radiation, but this is conservatively ignored because the PCBs face each other and are close enough to restrict natural convection. It is therefore assumed that all the heat is transferred from the components to the cooling air by conduction through the PCB.

The heat flow path from the component surface (or case) to the cooling

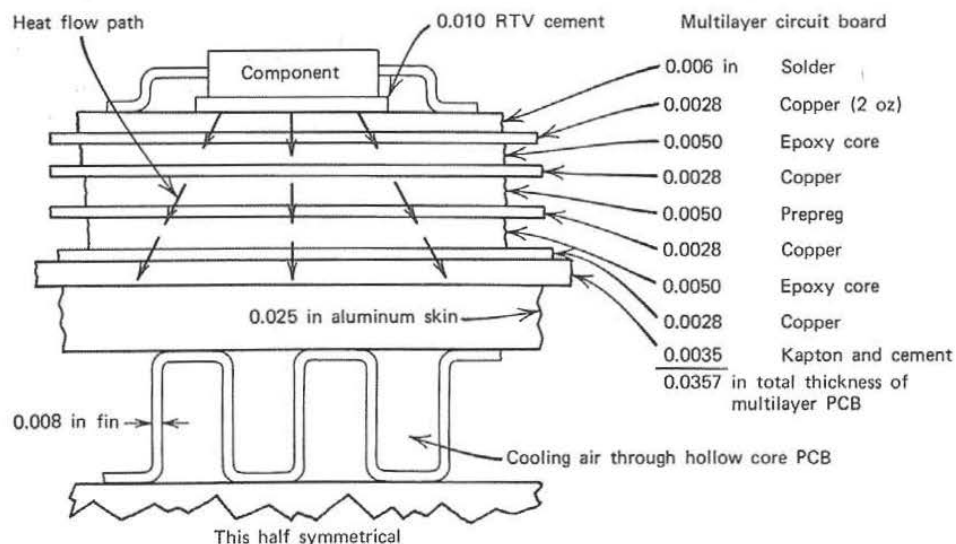


Figure 6.39 Enlarged section through a hollow PCB with a heat exchanger core and multilayer circuit laminas on both sides.

air in the hollow core heat exchanger is evaluated by determining the temperature rise along four segments of the heat flow path as follows:

- Δt_1 = temperature rise across component interface, from case to circuit lamina
- Δt_2 = temperature rise through the multilayer circuit lamina to the aluminum heat exchanger skin
- Δt_3 = temperature rise across the convection air film, from the aluminum heat exchanger surface to the center of cooling air stream
- Δt_4 = temperature rise of the cooling air stream from the entrance of the heat exchanger to the exit

Each section along the heat flow path is examined in more detail.

Δt_1 : temperature rise across component interface from case to circuit lamina

The electronic types being examined are hybrid and large scale integrated (LSI) components. These usually come in relatively large case sizes, approximately 1.0 in (2.54 cm) $\times$ 1.0 in (2.54 cm). These components will dissipate a large amount of heat, since the total power dissipation on the PCB is 50 watts. When the power density on such a component is more than about 4 watts/in², a good thermally conductive cement should be used to prevent an excessive temperature rise across the component interface as shown in Figure 6.40. Room temperature vulcanizing (RTV) cement is often used to permit the component to be removed if it has to be replaced. Epoxy cements are available with higher thermal conductivities, but components bonded with this material are very difficult to replace without damaging the PCB.

Test data show that some heat is conducted from the body of the hybrid component to the electrical lead wires. This heat flows along the lead wires to the circuit lamina solder pads, then through the circuit lamina into the heat exchanger.

The heat loss through the lead wires is usually ignored because the strain relief produces a long lead length with a high resistance, and there is another

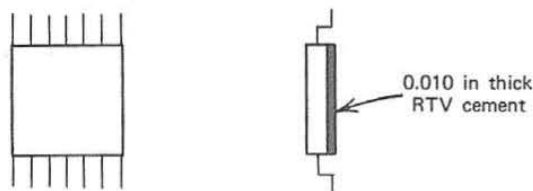


Figure 6.40 Hybrid component fastened to PCB with RTV cement.

high resistance through the small cross section area through the circuit lamina.

Considering a maximum power dissipation of 4 watts for one hybrid, the temperature rise across an RTV interface can be determined from Eq. 3.1. See Section 1.11 for conversions between English units and metric units.

$$\begin{aligned} \text{Given } Q &= 4.0 \text{ watts} = 13.65 \text{ Btu/hr} = 0.956 \text{ cal/sec (heat flow)} \\ L &= 0.010 \text{ in} = 0.000833 \text{ ft} = 0.0254 \text{ cm (RTV thickness)} \\ K &= 0.22 \text{ Btu/hr ft } ^\circ\text{F} = 0.000908 \text{ cal/sec cm } ^\circ\text{C (RTV conductivity)} \\ A &= \frac{(1.0)(1.0) \text{ in}^2}{144 \text{ in}^2/\text{ft}^2} = 0.00694 \text{ ft}^2 = 6.45 \text{ cm}^2 \text{ (RTV area)} \end{aligned}$$

In English units, with dimensions in feet:

$$\Delta t_1 = \frac{(13.65)(0.000833)}{(0.22)(0.00694)} = 7.4^\circ\text{F} \quad (6.91)$$

In metric units:

$$\Delta t_1 = \frac{(0.956)(0.0254)}{(0.000908)(6.45)} = 4.1^\circ\text{C} \quad (6.91a)$$

Δt_2 : temperature rise through the multilayer circuit lamina to the aluminum heat exchanger skin

The multilayer circuit lamina consists of 0.005 in thick epoxy fiberglass sheets, with etched copper circuits on each side. Two of these 0.005 in sheets are bonded together with a prepreg epoxy fiberglass cement using high temperatures and pressures to force the prepreg to flow between the copper runs, eliminating any air bubbles. Holes for interconnecting circuits are drilled through the bonded assembly, which is then plated with a thin copper layer. The plating forms an electrically conducting tube through the drilled holes and also builds up the thickness of the copper runs on both sides of the bonded lamina assembly. The original copper thickness on the lamina is usually 0.0014 in or about 1 ounce. The plating process usually adds about another ounce, which brings the total thickness of the copper runs on the outer surfaces to 0.0028 in.

The copper-plated throughholes (PTH) tend to increase the effective thermal conductivity through the epoxy fiberglass laminas, because copper has a very high thermal conductivity. Typical values are about 20%, depending upon the number of PTHs used.

In addition, the copper circuit layers within the bonded lamina will tend to spread the heat flowing through the lamina if there are many copper circuit runs. This tends to increase the effective cross section area along

the heat flow path through the circuit lamina about 15%, depending upon the number of copper circuit runs.

The temperature rise through the circuit lamina is also determined from Eq. 3.1.

Given $Q = 4 \text{ watts} = 13.65 \text{ Btu/hr} = 0.956 \text{ cal/sec (heat)}$
 $L = 0.0357 \text{ in} = 0.00297 \text{ ft} = 0.0907 \text{ cm (thickness of circuit lamina)}$
 $K = 0.20 \text{ Btu/hr ft } ^\circ\text{F} = 0.000826 \text{ cal/sec cm } ^\circ\text{C (conductivity with no PTHs)}$
 $K = 0.24 \text{ Btu/hr ft } ^\circ\text{F} = 0.000991 \text{ cal/sec cm } ^\circ\text{C (conductivity with PTHs)}$
 $A = (1.0)(1.0) = 1.0 \text{ in}^2 = 0.00694 \text{ ft}^2 = 6.452 \text{ cm}^2 \text{ (area with no heat spreading)}$
 $A = 1.15 \text{ in}^2 = 0.00798 \text{ ft}^2 = 7.419 \text{ cm}^2 \text{ (area with heat spreading)}$

Conservatively ignore the improved conductivity of the PTHs and the spreading effects of the copper runs. In English units:

$$\Delta t_2 = \frac{(13.65)(0.00297)}{(0.20)(0.00604)} = 29.2^\circ\text{F} \quad (6.92)$$

In metric units:

$$\Delta t_2 = \frac{(0.956)(0.0907)}{(0.000826)(6.452)} = 16.3^\circ\text{C} \quad (6.92a)$$

Δt_3 : temperature rise across the convection air film from the aluminum heat exchanger surface to the center of the cooling air stream

Equation 5.7 is used to determine this temperature rise. However, first the surface area of the heat exchanger must be calculated. If the total power dissipation is considered, the total heat exchanger surface area must be used. It is convenient to consider the solid height of the fins to compute the side wall area, as shown in Figure 6.41.

$$\text{fin solid height} = (18 \text{ fins/in})(3.75 \text{ in})(0.008 \text{ in}) = 0.54 \text{ inch}$$

$$\text{fin area} = (67.5 \text{ fins})(0.192 \text{ in})(6.2 \text{ in})(2 \text{ surfaces})$$

$$\text{fin area} = 160.7 \text{ in}^2 = 1.116 \text{ ft}^2 = 1036.8 \text{ cm}^2$$

$$\text{sidewall area} = (3.21 \text{ in})(6.2 \text{ in})(2 \text{ surfaces})$$

$$\text{sidewall area} = 39.8 \text{ in}^2 = 0.276 \text{ ft}^2 = 256.8 \text{ cm}^2$$

$$\text{total surface area } A = 1.392 \text{ ft}^2 = 1293.6 \text{ cm}^2$$

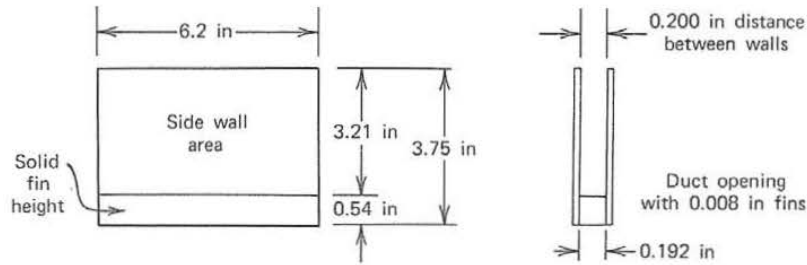


Figure 6.41 Heat exchanger fins shown as a solid height.

Heat:

$$Q = 50 \text{ watts} = 170.65 \frac{\text{Btu}}{\text{hr}} = 11.95 \frac{\text{cal}}{\text{sec}} \text{ (total power)}$$

$$h_c = 12.3 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} = 0.00166 \frac{\text{cal}}{\text{sec cm}^2 \text{ } ^\circ\text{C}} \text{ (Ref. Eqs. 6.90 and 6.90a)}$$

For English units, with dimensions in feet:

$$\Delta t_3 = \frac{Q}{h_c A} = \frac{170.65}{(12.3)(1.392)} = 10.0^\circ\text{F} \quad (6.93)$$

In metric units:

$$\Delta t_3 = \frac{11.95}{(0.00166)(1293.6)} = 5.6^\circ\text{C} \quad (6.93a)$$

Δt_4 : temperature rise of the cooling air from the entrance of the heat exchanger to the exit

As the cooling air flows through the PCB heat exchanger, it will pick up the heat dissipated from the electronic components. The temperature rise of the cooling air is determined by considering the total PCB power dissipation with Eq. 6.7.

Given $Q = 50 \text{ watts} = 170.65 \text{ Btu/hr} = 11.95 \text{ cal/sec}$ (total power)

$W = 0.408 \text{ lb/min} = 3.09 \text{ g/sec}$ (ref. Eqs. 6.86 and 6.86a)

$C_p = 0.24 \text{ Btu/lb } ^\circ\text{F} = 0.24 \text{ cal/g } ^\circ\text{C}$ (specific heat)

In English units:

$$\Delta t_4 = \frac{Q}{WC_p} = \frac{170.65 \text{ Btu/hr}}{(0.408 \text{ lb/min})(60 \text{ min/hr})(0.24 \text{ Btu/lb } ^\circ\text{F})} = 29.0^\circ\text{F} \quad (6.94)$$

In metric units:

$$\Delta t_4 = \frac{11.95 \text{ cal/sec}}{(3.09 \text{ g/sec})(0.24 \text{ cal/g } ^\circ\text{C})} = 16.1^\circ\text{C} \quad (6.94a)$$

The maximum component case temperature is obtained by adding all of the temperature rises Δt_1 through Δt_4 to the inlet air temperature of 131°F (55°C).

$$t_{\text{case}} = t_{\text{in}} + \Delta t_1 + \Delta t_2 + \Delta t_3 + \Delta t_4 \quad (6.95)$$

For English units, substitute Eqs. 6.91 through 6.94 into Eq. 6.95.

$$t_{\text{case}} = 131 + 7.4 + 29.2 + 10.0 + 29.0 = 206.6^\circ\text{F} \quad (6.96)$$

For metric units, substitute Eqs. 6.91a through 6.94a into Eq. 6.95.

$$t_{\text{case}} = 55 + 4.1 + 16.3 + 5.6 + 16.1 = 97.1^\circ\text{C} \quad (6.96a)$$

SOLUTION PART 4

The static pressure drop through the PCB heat exchanger only is desired from Eq. 6.82, in terms of the height of a column of water.

The friction factor is obtained from Table 6.7, considering the cross section of one tube of the heat exchanger to be a rectangular duct with an aspect ratio of 4. Using the Reynolds number computed in Eqs. 6.88 and 6.88a, the friction in English units and metric units is shown in Eq. 6.97.

$$f = \frac{18.3}{N_R} = \frac{18.3}{727} = 0.0252 \text{ (dimensionless)} \quad (6.97)$$

The cooling air velocity (V) through the heat exchanger is shown in Eq. 6.98. It is obtained from the physical properties of the air, which were previously determined for the Reynolds number shown in Eqs. 6.88 and 6.88a. The area (A) of one duct was shown in Eq. 6.87.

$$V = \frac{W}{\rho A} \quad (6.98)$$

In English units:

$$\begin{aligned} V &= \frac{0.00604 \text{ lb/min}}{(0.0656 \text{ lb/ft}^3)(0.0000633 \text{ ft}^2)(60 \text{ sec/min})} \\ &= 24.2 \frac{\text{ft}}{\text{sec}} = 290.4 \frac{\text{in}}{\text{sec}} \quad (6.99) \end{aligned}$$

In metric units:

$$V = \frac{0.0458 \text{ g/sec}}{(0.00105 \text{ g/cm}^3)(0.0588 \text{ cm}^2)} = 741 \frac{\text{cm}}{\text{sec}} \quad (6.99a)$$

Also:

$$\begin{aligned} L &= 6.2 \text{ in} = 0.516 \text{ ft} = 15.7 \text{ cm (length)} \\ D &= 0.0761 \text{ in} = 0.00635 \text{ ft} \\ &= 0.193 \text{ cm (ref. Eq. 6.87) (hydraulic diameter)} \\ g &= 32.16 \text{ ft/sec}^2 = 386 \text{ in/sec}^2 = 980 \text{ cm/sec}^2 \text{ (gravity)} \end{aligned}$$

A dimensional analysis shows that Eq. 6.82 will produce the static pressure loss in terms of the height of a column of air. The pressure loss is desired in terms of the height of a column of water. Therefore, the ratio of the air density (ρ_{air}) and the water density ($\rho_{\text{H}_2\text{O}}$) is used for the conversion, as shown in Eq. 6.100.

$$H_L = (4f) \left(\frac{L}{D} \right) \left(\frac{V^2}{2g} \right) \left(\frac{\rho_{\text{air}}}{\rho_{\text{H}_2\text{O}}} \right) \quad (6.100)$$

For English units, with dimensions in inches:

$$\begin{aligned} H_L &= (4)(0.0252) \left(\frac{6.2}{0.0761} \right) \frac{(290.4)^2}{(2)(386)} \frac{(0.0656 \text{ lb/ft}^3 \text{ air})}{(62.4 \text{ lb/ft}^3 \text{ H}_2\text{O})} \\ &= 0.94 \text{ in H}_2\text{O} \end{aligned} \quad (6.101)$$

In metric units:

$$\begin{aligned} H_L &= (4)(0.0252) \left(\frac{15.7}{0.193} \right) \frac{(741)^2}{(2)(980)} \frac{(0.00105 \text{ g/cm}^3 \text{ air})}{(1.0 \text{ g/cm}^3 \text{ H}_2\text{O})} \\ &= 2.39 \text{ cm H}_2\text{O} \end{aligned} \quad (6.101a)$$

The foregoing static pressure losses are only for the heat exchanger. If the complete static pressure loss through an electronic box is desired, other flow losses, such as entrance effects, turns, expansions contraction, and exhaust effects, must be added to obtain the final picture, as discussed in Section 6.7.

If the total pressure is required for the heat exchanger section, the velocity head must be added to the static head as shown in Eq. 6.3. The velocity head relation for English units is shown in Eqs. 6.2 and 6.4, where the

velocity must be in ft/min. The velocity head in English units becomes:

$$H_v = \left[\frac{(24.2 \text{ ft/sec})(60 \text{ sec/min})}{4005} \right]^2 = 0.13 \text{ in H}_2\text{O} \quad (6.102)$$

The velocity head in metric units is obtained from Eq. 6.5a as follows:

$$H_v = \left(\frac{741}{1277} \right)^2 = 0.33 \text{ cm H}_2\text{O} \quad (6.102a)$$

The total pressure loss through the heat exchanger, in English units, becomes

$$H_t = H_v + H_L = 0.13 + 0.94 = 1.07 \text{ in H}_2\text{O} \quad (6.103)$$

In metric units:

$$H_t = 0.33 + 2.39 = 2.72 \text{ cm H}_2\text{O} \quad (6.103a)$$

Equation 6.84 could also have been used to determine the static pressure loss in terms of $\sigma \Delta P$.

$$\sigma \Delta P = 0.226(4f) \left(\frac{L}{D} \right) \left(\frac{W}{A} \right)^2 \text{ (ref. Eq. 6.84)}$$

$$\text{where } W = \frac{0.408 \text{ lb/min}}{67.5 \text{ ducts}} = 0.00604 \text{ lb/min (ref. Eq. 6.86)}$$

$$L = 6.2 \text{ in} = \text{length of duct}$$

$$D = 0.0761 \text{ in} = \text{hydraulic diameter}$$

$$A = (0.192)(0.0475) = 0.00912 \text{ in}^2 \text{ (ref. Eq. 6.87)}$$

$$f = 0.0252 \text{ (ref. Eq. 6.97)}$$

$$\sigma \Delta P = 0.226(4)(0.0252) \left(\frac{6.2}{0.0761} \right) \left(\frac{0.00604}{0.00912} \right)^2$$

$$\sigma \Delta P = 0.813 \text{ in H}_2\text{O}$$

$$\sigma = \frac{\rho}{\rho_{\text{std}}} = \frac{0.0656}{0.0765} = 0.858 \text{ (ref. Eq. 6.65 and Eq. 6.98)}$$

$$\Delta P = \frac{\sigma \Delta P}{\sigma} = \frac{0.813}{0.858} = 0.94 \text{ in H}_2\text{O}$$

This checks with Eq. 6.101.

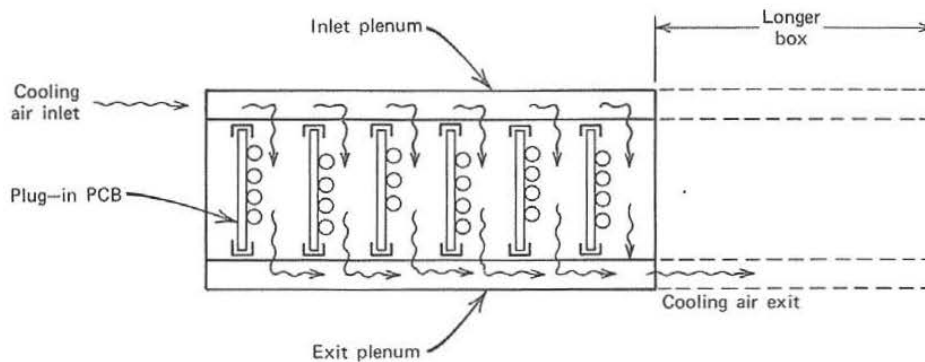


Figure 6.42 Parallel air flow system with an inlet and an exit cooling air plenum.

6.27 UNDESIRABLE AIR FLOW REVERSALS

Under certain sets of conditions it is possible for the air in a forced convection system to reverse its flow direction. If this happens, the cooling efficiency can be sharply reduced and overheating of the electronics may result.

This type of situation can develop in a parallel flow cooling system which incorporates a cooling air distribution plenum as shown in Figure 6.42.

The purpose of an inlet air distribution plenum is to provide a uniform air flow to the PCBs in the system. Ideally, the static pressure will be high and the velocity pressure will be low. A high static pressure will distribute the air equally to all of the PCBs. A low velocity pressure means a low velocity, which results in small entrance and turn losses.

If the box length is increased and the number of PCBs in the system are increased, or if the power dissipation in the system is substantially increased, additional cooling air will be required. If the cross section flow area of the air inlet plenum is not increased, the velocity of the cooling air

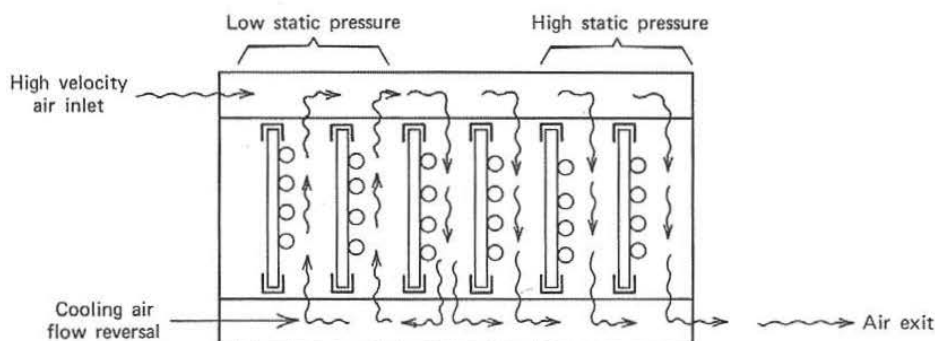


Figure 6.43 Air flow reversals may occur if the cooling air inlet velocity is too high.

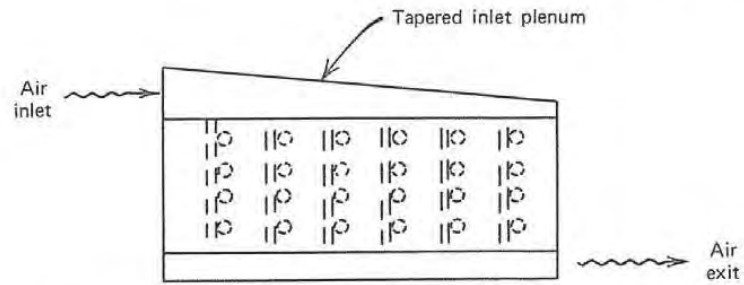


Figure 6.44 Tapered air inlet plenum on an electronic box.

flow must be increased to provide the additional air. As the velocity is increased, the static pressure will decrease. A high air velocity can result in such a low static pressure at the air inlet that it is well below the outside atmospheric pressure. When this happens, a flow reversal may occur. The air will then be drawn in from the exit plenum, as shown in Figure 6.43.

To prevent flow reversal, the inlet air plenum cross section area should

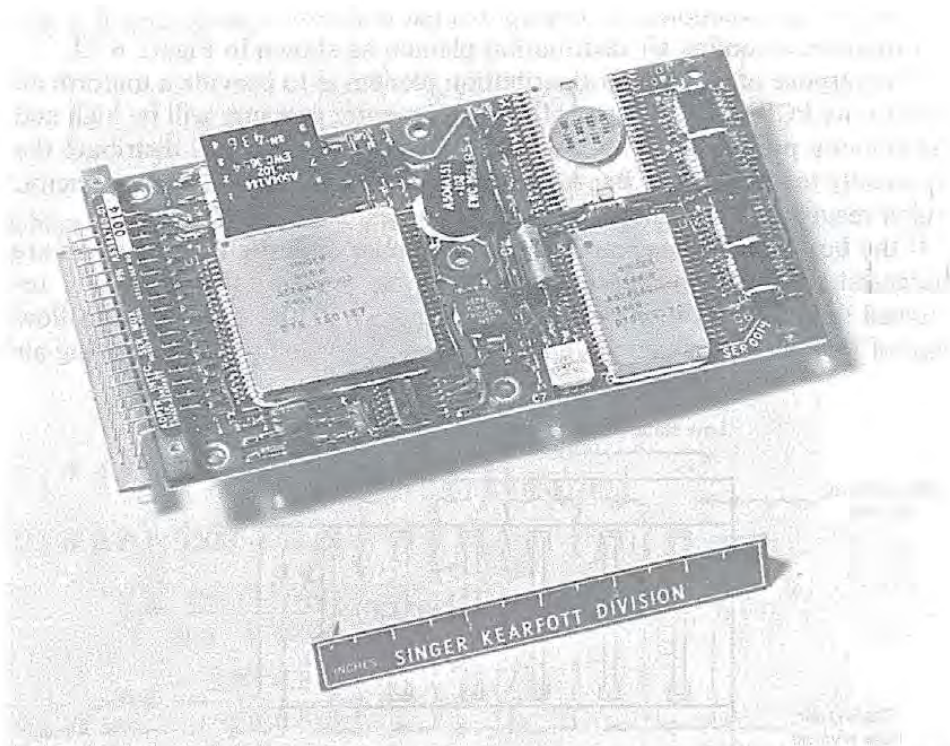


Figure 6.45 Electronic components are cemented to an aluminum core board with RTV cement to improve the conduction heat transfer. (Courtesy Kearfott Division, The Singer Company.)

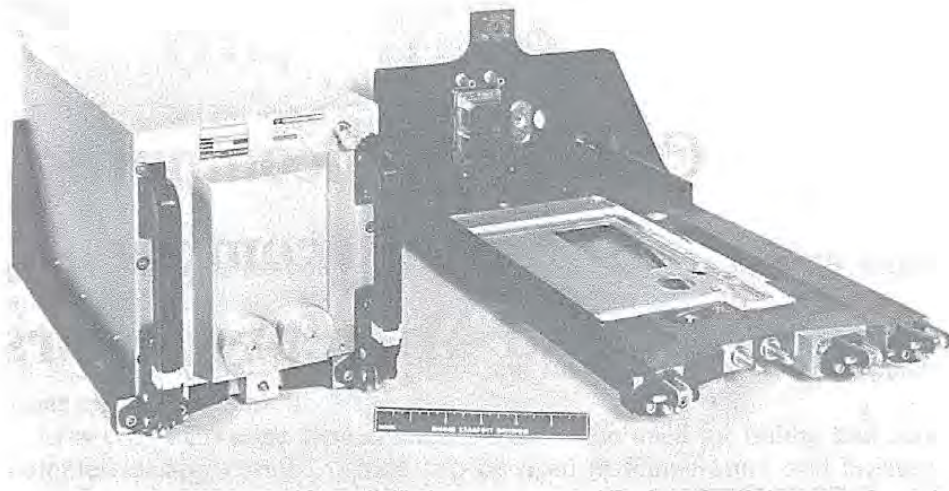


Figure 6.46 Inertial navigation assembly with a mounting rack that has a cooling air plenum in its base. (Courtesy Kearfott Division, The Singer Company.)

be greater than the sum of all the individual PCB inlet ports used to supply cooling air to the PCBs.

A tapered inlet plenum can also be used for the cooling air, as shown in Figure 6.44. The taper should be gradual so that the cross section area of the plenum at any point is greater than the total cross section area of the downstream air inlet ports to the PCBs.

Figure 6.45 shows a circuit board with many discrete and hybrid components held in position with RTV cement.

Figure 6.46 shows a mounting rack and an inertial navigation system that is packaged in a $\frac{3}{4}$ ATR short chassis.