Solutions Manual

for

Heat and Mass Transfer: Fundamentals & Applications

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Chapter 15
COOLING OF ELECTRONIC EQUIPMENT

Introduction and History

15-1C The invention of vacuum diode started the electronic age. The invention of the transistor marked the beginning of a revolution in that age since the transistors performed the functions of the vacuum tubes with greater reliability while occupying negligible space and consuming negligible power compared to the vacuum tubes.

15-2C Integrated circuits are semiconductor devices in which several components such as diodes, transistors, resistors and capacitors are housed together. The initials MSI, LSI, and VLSI stand for medium scale integration, large scale integration, and very large scale integration, respectively.

15-3C The electrical resistance R is a measure of resistance against current flow, and the friction between the electrons and the material causes heating. The amount of the heat generated can be determined from Ohm's law, $W = I^2 R$.

15-4C The electrical energy consumed by the TV is eventually converted to heat, and the blanket wrapped around the TV prevents the heat from escaping. Then the temperature of the TV set will have to start rising as a result of heat build up. The TV set will have to burn up if operated this way for a long time. However, for short time periods, the temperature rise will not reach destructive levels.

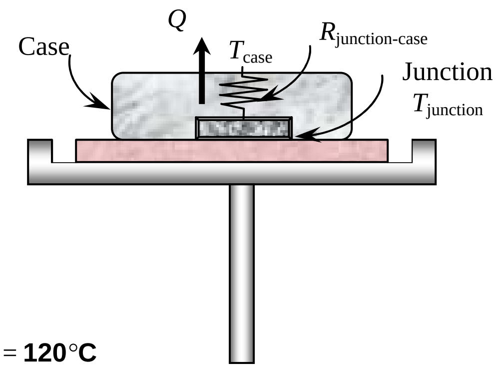
15-5C Since the heat generated in the incandescent light bulb which is completely wrapped can not escape, the temperature of the light bulb will increase, and will possibly start a fire by igniting the towel.

15-6C When the air flow to the radiator is blocked, the hot water coming off the engine cannot be cooled, and thus the engine will overheat and fail, and possible catch fire.

15-7C A car is much more likely to break since it has more moving parts than a TV.

15-8C Diffusion in semi-conductor materials, chemical reactions and creep in the bending materials cause electronic components to fail under prolonged use at high temperatures.

15-9 The case temperature of a power transistor and the junction-to-case resistance are given. The junction temperature is to be determined.



Assumptions Steady operating conditions exist.

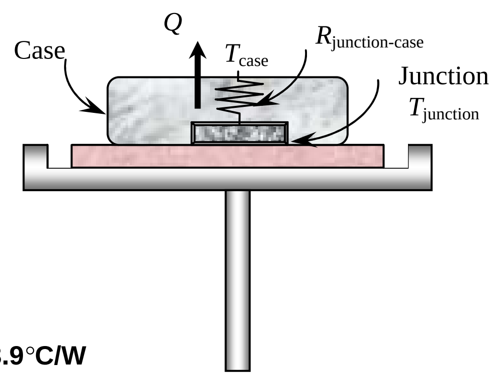
Analysis The rate of heat transfer between the junction and the case in steady operation is

$$\dot{Q} = \left(\frac{\Delta T}{R}\right)_{junction-case} = \frac{T_{junction} - T_{case}}{R_{junction-case}}$$

Then the junction temperature is determined to be

$$T_{junction} = T_{case} + \dot{Q}R_{junction-case} = 60°\text{C} + (12\text{ W})(5°\text{C/W}) = \mathbf{120°C}$$

15-10 The power dissipated by an electronic component as well as the junction and case temperatures are measured. The junction-to-case resistance is to be determined.



Assumptions Steady operating conditions exist.

Analysis The rate of heat transfer from the component is

$$\dot{W}_e = \dot{Q} = \mathbf{V}I = (12\text{ V})(0.15\text{ A}) = 1.8\text{ W}$$

Then the junction-to-case thermal resistance of this component becomes

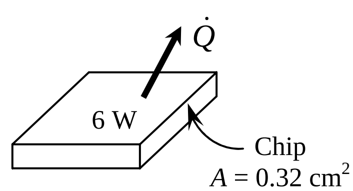
$$R_{junction-case} = \frac{T_{junction} - T_{case}}{\dot{Q}} = \frac{(80-55)°\text{C}}{1.8\text{ W}} = \mathbf{13.9°C/W}$$

15-11 A logic chip dissipates 6 W power. The amount of heat this chip dissipates during a 10-h period and the heat flux on the surface of the chip are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer from the surface is uniform.

Analysis (*a*) The amount of heat this chip dissipates during an eight-hour workday is

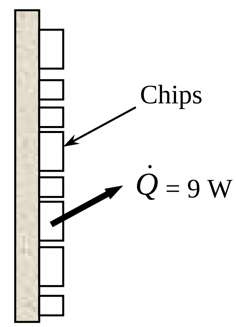
$$Q = \dot{Q}\Delta t = (0.006\text{ kW})(8\text{ h}) = \mathbf{0.048\ kWh}$$



(*b*) The heat flux on the surface of the chip is

$$\dot{q} = \frac{\dot{Q}}{A} = \frac{6\text{ W}}{0.32\text{ cm}^2} = \mathbf{18.8\ W/cm^2}$$

15-12 A circuit board houses 90 closely spaced logic chips, each dissipating 0.1 W. The amount of heat this chip dissipates in 10 h and the heat flux on the surface of the circuit board are to be determined.



Assumptions **1** Steady operating conditions exist. **2** The heat transfer from the back surface of the board is negligible.

Analysis (*a*) The rate of heat transfer and the amount of heat this circuit board dissipates during a ten-hour period are

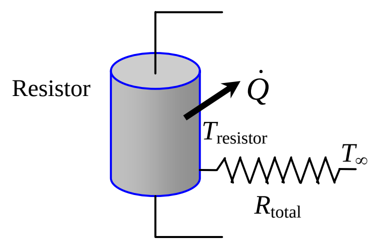
$$\dot{Q}_{total} = (90)(0.1\,\text{W}) = 9\text{ W}$$

$$Q_{total} = \dot{Q}_{total}\Delta t = (0.009\,\text{kW})(10\,\text{h}) = \mathbf{0.09\,kWh}$$

(*b*) The average heat flux on the surface of the circuit board is

$$\dot{q} = \frac{\dot{Q}_{total}}{A_s} = \frac{9\text{ W}}{(15\,\text{cm})(20\,\text{cm})} = \mathbf{0.03\ W/cm^2}$$

15-13E The total thermal resistance and the temperature of a resistor are given. The power at which it can operate safely in a particular environment is to be determined.



Assumptions Steady operating conditions exist.

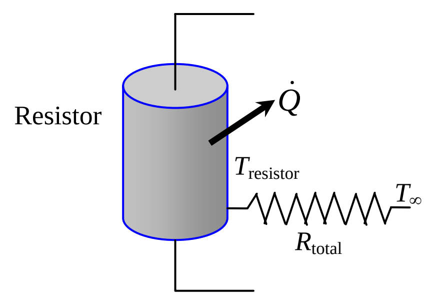
Analysis The power at which this resistor can be operate safely is determined from

$$\dot{Q} = \frac{T_{resistor} - T_{ambient}}{R_{total}} = \frac{(360-120)°\text{F}}{130°\text{F/W}} = \mathbf{1.85\ W}$$

15-14 The surface-to-ambient thermal resistance and the surface temperature of a resistor are given. The power at which it can operate safely in a particular environment is to be determined.

Assumptions Steady operating conditions exist.

Analysis The power at which this resistor can operate safely is determined from



$$\dot{Q} = \frac{T_{resistor} - T_{ambient}}{R_{total}} = \frac{(150-30)°\text{C}}{300°\text{C/W}} = \mathbf{0.4\ W}$$

At specified conditions, the resistor dissipates

$$\dot{Q} = \frac{\mathbf{V}^2}{R} = \frac{(7.5\,\text{V})^2}{(100\,\Omega)} = 0.5625\text{ W}$$

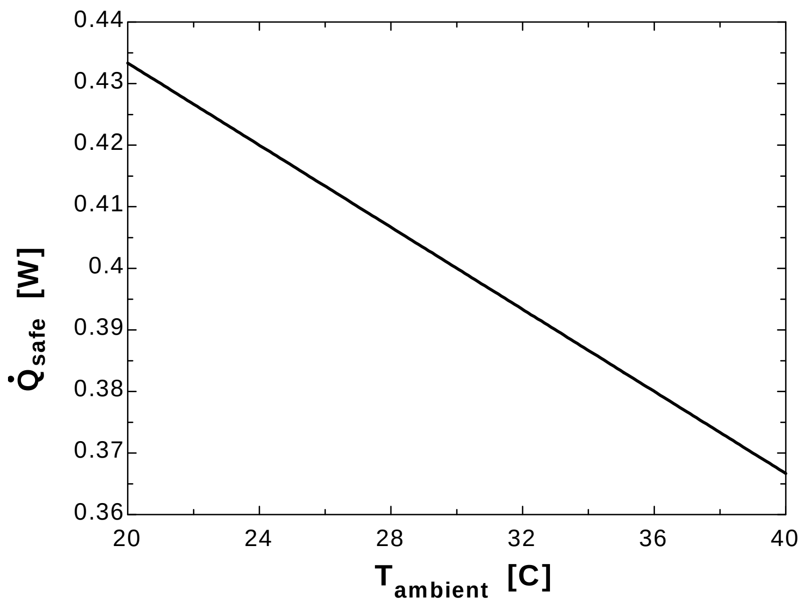
of power. Therefore, the current operation is not safe.

15-15 EES Prob. 15-14 is reconsidered. The power at which the resistor can operate safely as a function of the ambient temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
R_electric=100 [ohm]
R_thermal=300 [C/W]
V=7.5 [volt]
T_resistor=150 [C]
T_ambient=30 [C]

"ANALYSIS"
Q_dot_safe=(T_resistor-T_ambient)/R_thermal
```

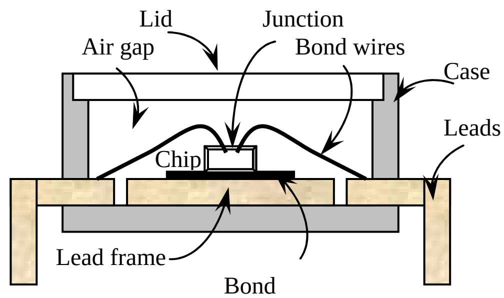
$T_{ambient}$ [C]	Q_{safe} [W]
20	0.4333
21	0.43
22	0.4267
23	0.4233
24	0.42
25	0.4167
26	0.4133
27	0.41
28	0.4067
29	0.4033
30	0.4
31	0.3967
32	0.3933
33	0.39
34	0.3867
35	0.3833
36	0.38
37	0.3767
38	0.3733
39	0.37
40	0.3667



Manufacturing of Electronic Equipment

15-16C The thermal expansion coefficient of the plastic is about 20 times that of silicon. Therefore, bonding the silicon directly to the plastic case will result in such large thermal stresses that the reliability would be seriously jeopardized. To avoid this problem, a lead frame made of a copper alloy with a thermal expansion coefficient close to that of silicon is used as the bonding surface.

15-17C The schematic of chip carrier is given in the figure. Heat generated at the junction is transferred through the chip to the led frame, then through the case to the leads. From the leads heat is transferred to the ambient or to the medium the leads are connected to.



15-18C The cavity of the chip carrier is filled with a gas which is a poor conductor of heat. Also, the case is often made of materials which are also poor conductors of heat. This results in a relatively large thermal resistance between the chip and the case, called the junction-to-case thermal resistance. It depends on the geometry and the size of the chip carrier as well as the material properties of the bonding material and the case.

15-19C A hybrid chip carrier houses several chips, individual electronic components, and ordinary circuit elements connected to each other. The result is improved performance due to the shortening of the wiring lengths, and enhanced reliability. Lower cost would be an added benefit of multi-chip packages if they are produced in sufficiently large quantities.

15-20C A printed circuit board (PCB) is a properly wired plane board on which various electronic components such as the ICs, diodes, transistors, resistors, and capacitors are mounted to perform a certain task. The board of a PCB is made of polymers and glass epoxy materials. The thermal resistance between a device on the board and edge of the board is called as device-to-PCB edge thermal resistance. This resistance is usually high (about 20 to 60 $°C/W$) because of the low thickness of the board and the low thermal conductivity of the board material.

15-21C The three types of circuit boards are the single-sided, double-sided, and multi-layer boards. The single-sided PCBs have circuitry lines on one side of the board only, and are suitable for low density electronic devices (10-20 components). The double-sided PCBs have circuitry on both sides, and are best suited for intermediate density devices. Multi-layer PCBs contain several layers of circuitry, and they are suitable for high density devices. They are equivalent to several PCBs sandwiched together.

15-22C The desirable characteristics of the materials used in the fabrication of circuit boards are: (1) being an effective electrical insulator to prevent electrical breakdown, (2) being a good heat conductor to conduct the heat generated away, (3) having high material strength to withstand the forces and to maintain dimensional stability, (4) having a thermal expansion coefficient which closely matches to that of copper to prevent cracking in the copper cladding during thermal cycling, (5) having a high resistance to moisture absorption since moisture can effect both mechanical and electrical properties and degrade performance, (6) stability in properties at temperature levels encountered in electronic applications, (7) ready availability and manufacturability, and, of course (8) low cost.

15-23C An electronic enclosure (a case or a cabinet) house the circuit boards and the necessary peripheral equipment and connectors. It protects them from the detrimental effects of the environment, and may provide a cooling path. An electronic enclosure can simply be made of sheet metals such as thin gauge aluminum or steel.

Cooling Load of Electronic Equipment and Thermal Environment

15-24C The heating load of an electronic box which consumes 120 W of power is simply 120 W because of the conservation of energy principle.

15-25C Superconductor materials will generate hardly any heat and as a result, more components can be packed into a smaller volume, resulting in enhanced speed and reliability without having to resort to some exotic cooling techniques.

15-26C The actual power dissipated by a device can be considerably less than its rated power, depending on its duty cycle (the fraction of time it is on). A 5 W power transistor, for example, will dissipate an average of 2 W of power if it is active only 40 percent of the time. Then we can treat this transistor as a 2-W device when designing a cooling system. This may allow the selection of a simpler and cheaper cooling mechanism.

15-27C The cyclic variation of temperature of an electronic device during operation is called the temperature cycling. The thermal stresses caused by temperature cycling undermines the reliability of electronic devices. The failure rate of electronic devices subjected to deliberate temperature cycling of more than $20\,^\circ\mathrm{C}$ is observed to increase by eight-fold.

15-28C The ultimate heat sink for a TV is the room air with a temperature range of about 10 to 30°C. For an airplane it is the ambient air with a temperature range of about -50°C to 50°C. The ultimate heat sink for a ship is the sea water with a temperature range of 0°C to 30°C.

15-29C The ultimate heat sink for a DVD payer is the room air with a temperature range of about 10 to 30°C. For a spacecraft it is the ambient air or space with a temperature range of about -273°C to 50°C. The ultimate heat sink for a communication system on top of a mountain is the ambient air with a temperature range of about -20°C to 50°C.

Electronics Cooling in Different Applications

15-30C The electronics of short-range missiles do not need any cooling because of their short cruising times. The missiles reach their destinations before the electronics reach unsafe temperatures. The long-range missiles must be cooled because of their long cruise times (several hours). The electronics in this case are cooled by passing the liquid fuel they carry through the cold plate of the electronics enclosure as it flows towards the combustion chamber.

15-31C Dynamic temperature is the rise in the temperature of a fluid as a result of the ramming effect or the stagnation process. This is due to the conversion of kinetic energy to internal energy which is significant at high velocities. It is determined from $T_{dynamic} = V^2/(2c_p)$ where V is the velocity and c_p is the specific heat of the fluid. It is significant at velocities above 100 m/s.

15-32C The electronic equipment in ships and submarines are usually housed in rugged cabinets to protect them from vibrations and shock during stormy weather. Because of easy access to water, water cooled heat exchangers are commonly used to cool sea-born electronics. Often air in a closed or open loop is cooled in an air-to-water heat exchanger, and is forced to the electronic cabinet by a fan.

15-33C The electronics of communication systems operate for long periods of time under adverse conditions such as rain, snow, high winds, solar radiation, high altitude, high humidity, and too high or too low temperatures. Large communication systems are housed in specially built shelters. Sometimes it is necessary to air-condition these shelters to safely dissipate the large quantities of heat generated by the electronics of communication systems.

15-34C The electronic components used in the high power microwave equipment such as radars generate enormous amounts of heat because of the low conversion efficiency of electrical energy to microwave energy. The klystron tubes of high power radar systems where radio frequency (RF) energy is generated can yield local heat fluxes as high as $2000\ \mathrm{W/cm^2}$. The safe and reliable dissipation of such high heat fluxes usually require the immersion of such equipment into a suitable dielectric fluid which can remove large quantities of heat by boiling.

15-35C The electronic equipment in space vehicles are usually cooled by a liquid circulated through the components where heat is picked up, and then through a space radiator where the waste heat is radiated into deep space at 0 K. In such systems it may be necessary to run a fan in the box to circulate the air since there is no natural convection currents in space because of the absence of a gravity field.

15-36 An airplane cruising in the air at a temperature of -25°C at a velocity of 850 km/h is considered. The temperature rise of air is to be determined.

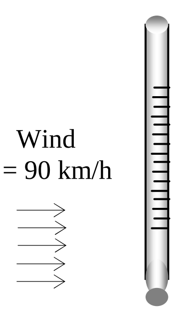
V = 850 km/h

Assumptions Steady operating conditions exist.

Analysis The temperature rise of air (dynamic temperature) at this speed is

$$T_{dynamic} = \frac{V^2}{2c_p} = \frac{(850\times 1000/3600\ \mathrm{m/s})^2}{(2)(1003\ \mathrm{J/kg.°C})}\left(\frac{1\ \mathrm{J/kg}}{1\ \mathrm{m^2/s^2}}\right) = \mathbf{27.8°C}$$

15-37 The temperature of air in the wind at a wind velocity of 90 km/h is measured to be 12°C. The true temperature of air is to be determined.



Assumptions Steady operating conditions exist.

Analysis The temperature rise of air (dynamic temperature) at this speed is

$$T_{dynamic} = \frac{V^2}{2c_p} = \frac{(90\times 1000/3600\ \mathrm{m/s})^2}{(2)(1005\ \mathrm{J/kg.°C})}\left(\frac{1\ \mathrm{J/kg}}{1\ \mathrm{m^2/s^2}}\right) = \mathbf{0.3°C}$$

Therefore, the true temperature of air is

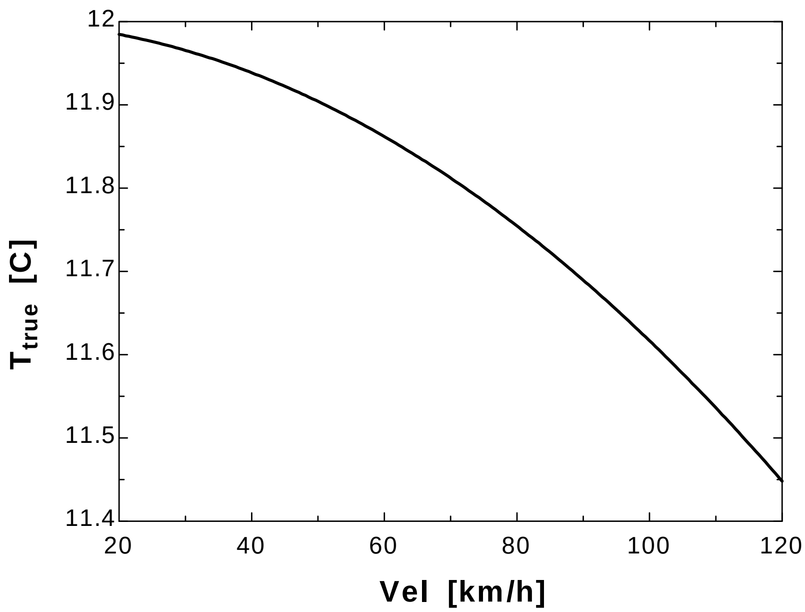
$$T_{true} = T_{measured} - T_{dynamic} = (12-0.3)°C = \mathbf{11.7°C}$$

15-38 EES Prob. 15-37 is reconsidered. The true temperature of air as a function of the wind velocity is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_measured=12 [C]
Vel=90 [km/h]
"PROPERTIES"
c_p=CP(air, T=T_measured)*Convert(kJ/kg-C, J/kg-C)
"ANALYSIS"
T_dynamic=(Vel*Convert(km/h, m/s))^2/(2*c_p)*Convert(m^2/s^2, J/kg)
T_true=T_measured-T_dynamic
```

Vel [km/h]	T_{true} [C]
20	11.98
25	11.98
30	11.97
35	11.95
40	11.94
45	11.92
50	11.9
55	11.88
60	11.86
65	11.84
70	11.81
75	11.78
80	11.75
85	11.72
90	11.69
95	11.65
100	11.62
105	11.58
110	11.54
115	11.49
120	11.45

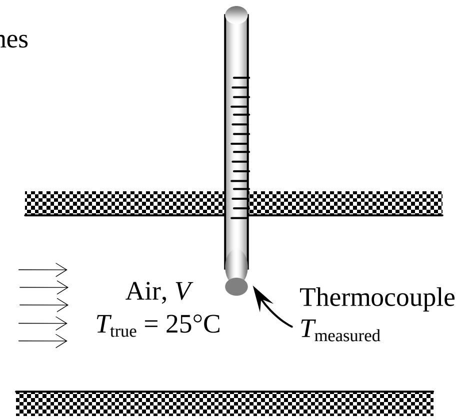


15-39 Air at 25°C is flowing in a channel. The temperature a stationary probe inserted into the channel will read is to be determined for different air velocities.

Assumptions Steady operating conditions exist.

Analysis (*a*) The temperature rise of air (dynamic temperature) for an air velocity of 1 m/s is

$$T_{dynamic} = \frac{V^2}{2c_p} = \frac{(1\,\text{m/s})^2}{(2)(1005\,\text{J/kg.°C})}\left(\frac{1\,\text{J/kg}}{1\,\text{m}^2/\text{s}^2}\right) = 0.0005°\text{C}$$



Then the temperature which a stationary probe will read becomes

$$T_{measured} = T_{true} + T_{dynamic} = 25 + 0.0005 = \mathbf{25.0005°C}$$

(*b*) For an air velocity of 10 m/s the temperature rise is

$$T_{dynamic} = \frac{V^2}{2c_p} = \frac{(10\,\text{m/s})^2}{(2)(1005\,\text{J/kg.°C})}\left(\frac{1\,\text{J/kg}}{1\,\text{m}^2/\text{s}^2}\right) = 0.05°\text{C}$$

Then, $T_{measured} = T_{true} + T_{dynamic} = 25 + 0.05 = \mathbf{25.05°C}$

(*c*) For an air velocity of 100 m/s the temperature rise is

$$T_{dynamic} = \frac{V^2}{2c_p} = \frac{(100\,\text{m/s})^2}{(2)(1005\,\text{J/kg.°C})}\left(\frac{1\,\text{J/kg}}{1\,\text{m}^2/\text{s}^2}\right) = 4.98°\text{C}$$

Then, $T_{measured} = T_{true} + T_{dynamic} = 25 + 4.98 = \mathbf{29.98°C}$

(*d*) For an air velocity of 1000 m/s the temperature rise is

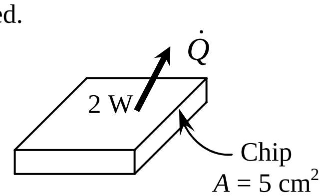
$$T_{dynamic} = \frac{V^2}{2c_p} = \frac{(1000\,\text{m/s})^2}{(2)(1005\,\text{J/kg.°C})}\left(\frac{1\,\text{J/kg}}{1\,\text{m}^2/\text{s}^2}\right) = 497.5°\text{C}$$

Then, $T_{measured} = T_{true} + T_{dynamic} = 25 + 497.5 = \mathbf{522.5°C}$

15-40 Power dissipated by an electronic device as well as its surface area and surface temperature are given. A suitable cooling technique for this device is to be determined.

Assumptions Steady operating conditions exist.

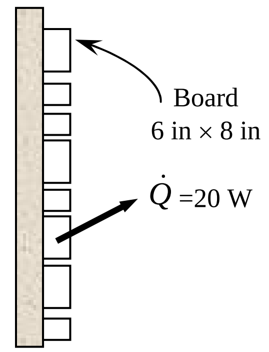
Analysis The heat flux on the surface of this electronic device is



$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{2\,\text{W}}{5\,\text{cm}^2} = \mathbf{0.4\ W/cm^2}$$

For an allowable temperature rise of 50°C, the suitable cooling technique for this device is determined from Fig. 15-17 to be **forced convection** with direct air.

15-41E Power dissipated by a circuit board as well as its surface area and surface temperature are given. A suitable cooling mechanism is to be selected.



Assumptions Steady operating conditions exist.

Analysis The heat flux on the surface of this electronic device is

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{20\text{ W}}{(6\text{ in} \times 2.54\text{ cm/in})(8\text{ in} \times 2.54\text{ cm/in})} = \mathbf{0.065\ W/cm^2}$$

For an allowable temperature rise of 80°F, the suitable cooling technique for this device is determined from Fig. 15-17 to be **natural convection** with direct air.

Conduction Cooling

15-42C The major considerations in the selection of a cooling technique are the magnitude of the heat generated, the reliability requirements, the environmental conditions, and the cost.

15-43C Thermal resistance is the resistance of a materiel or device against heat flow through it. It is analogous to electrical resistance in electrical circuits, and the thermal resistance networks can be analyzed like electrical circuits.

15-44C If the rate of heat conduction through a medium $\dot{Q}$, and the thermal resistance R of the medium are known, then the temperature difference across the medium can be determined from $\Delta T = \dot{Q}R$.

15-45C The voltage drop across the wire is determined from $\Delta \mathbf{V} = IR$. The length of the wire is proportional to the electrical resistance [$R = L/(\rho A)$], which is proportional to the voltage drop. Therefore, doubling the wire length while the current I is held constant will double the voltage drop.

The temperature drop across the wire is determined from $\Delta T = \dot{Q}R$. The length of the wire is proportional to the thermal resistance [$R = L/(kA)$], which is proportional to the temperature drop. Therefore, doubling the wire length while the heat flow $\dot{Q}$ is held constant will double the temperature drop.

15-46C A heat frame is a thick metal plate attached to a circuit board. It enhances heat transfer by providing a low resistance path for the heat flow from the circuit board to the heat sink. The thicker the heat frame, the lower the thermal resistance and thus the smaller the temperature difference between the center and the ends of the heat frame. The electronic components at the middle of a PCB operate at the highest temperature since they are furthest away from the heat sink.

15-47C Heat flow from the junction to the body of a chip is three-dimensional, but can be approximated as being one-dimensional by adding a constriction thermal resistance to the thermal resistance network. For a small heat generation area of diameter *a* on a considerably larger body, the constriction resistance is given by $R_{constriction} = 1/(2\sqrt{\pi}ak)$ where k is the thermal conductivity of the larger body. The constriction resistance is analogous to a partially closed valve in fluid flow, and a sudden drop in the cross-sectional area of an wire in electric flow.

15-48C The junction-to-case thermal resistance of an electronic component is the overall thermal resistance of all parts of the electronic component between the junction and case. In practice, this value is determined experimentally. When the junction-to-case resistance, the power dissipation, and the case temperature are known, the junction temperature of a component is determined from

$$T_{junctiion} = T_{case} + \dot{Q}R_{junction-case}$$

15-49C The case-to-ambient thermal resistance of an electronic device is the total thermal resistance of all parts of the electronic device between its outer surface and the ambient. In practice, this value is determined experimentally. Usually, manufacturers list the total resistance between the junction and the ambient for devices they manufacture for various configurations and ambient conditions likely to be encountered. When the case-to-ambient resistance, the power dissipation, and the ambient temperature are known, the junction temperature of the device is determined from $T_{junctiion} = T_{ambient} + \dot{Q}R_{junction-ambient}$

15-50C The junction temperature in this case is determined from

$$T_{junctiion} = T_{ambient} + \dot{Q}\left(R_{junction-case} + R_{case-ambient}\right).$$

When $R_{junction-case} > R_{case-ambient}$, the case temperature will be closer to the ambient temperature.

15-51C The PCBs are made of electrically insulating materials such as glass-epoxy laminates which are poor conductors of heat. Therefore, the rate of heat conduction along a PCB is very low. Heat conduction from the mid parts of a PCB to its outer edges can be improved by attaching heat frames or clamping cold plates to it. Heat conduction across the thickness of the PCB can be improved by planting copper or aluminum pins across the thickness of the PCB to serve as thermal bridges.

15-52C The thermal expansion coefficients of aluminum and copper are about twice as large as that of the epoxy-glass. This large difference in the thermal expansion coefficients can cause warping on the PCBs if the epoxy and the metal are not bonded properly. Warping is a major concern because it decreases reliability. One way of avoiding warping is to use PCBs with components on both sides.

15-53C The thermal conduction module received a lot of attention from thermal designers because the thermal design was incorporated at the initial stages of electrical design. The TCM was different from previous chip designs in that it incorporated both electrical and thermal considerations in early stages of design. The cavity in the TCM is filled with helium (instead of air) because of its very high thermal conductivity (about six times that of air).

15-54 The dimensions and power dissipation of a chip are given. The junction temperature of the chip is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through various components is one-dimensional. **3** Heat transfer through the air gap and the lid on top of the chip is negligible because of the very large thermal resistance involved along this path.

Analysis The various thermal resistances on the path of primary heat flow are

$$R_{constriction} = \frac{1}{2\sqrt{\pi}ak} = \frac{1}{2\sqrt{\pi}(0.5\times10^{-3}\text{ m})(120\text{ W/m.°C})} = 4.7\text{°C/W}$$

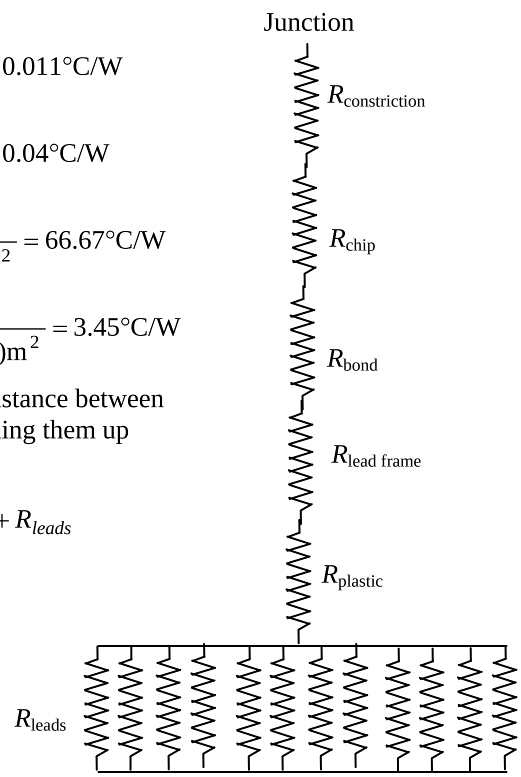
$$R_{chip} = \frac{L}{kA} = \frac{0.5\times10^{-3}\text{ m}}{(120\text{ W/m.°C})(0.004\times0.004)\text{m}^2} = 0.26\text{°C/W}$$

$$R_{bond} = \frac{L}{kA} = \frac{0.05\times10^{-3}\text{ m}}{(296\text{ W/m.°C})(0.004\times0.004)\text{m}^2} = 0.011\text{°C/W}$$

$$R_{\substack{lead\\frame}} = \frac{L}{kA} = \frac{0.25\times10^{-3}\text{ m}}{(386\text{ W/m.°C})(0.004\times0.004)\text{m}^2} = 0.04\text{°C/W}$$

$$R_{plastic} = \frac{L}{kA} = \frac{0.3\times10^{-3}\text{ m}}{(1\text{ W/m.°C})(18\times0.001\times0.00025)\text{m}^2} = 66.67\text{°C/W}$$

$$R_{leads} = \frac{L}{kA} = \frac{6\times10^{-3}\text{ m}}{(386\text{ W/m.°C})(18\times0.001\times0.00025)\text{m}^2} = 3.45\text{°C/W}$$



Since all resistances are in series, the total thermal resistance between the junction and the leads is determined by simply adding them up

$$\begin{aligned} R_{total} &= R_{junction-lead} \\ &= R_{constriction} + R_{chip} + R_{bond} + R_{\substack{lead\\frame}} + R_{plastic} + R_{leads} \\ &= 4.7 + 0.26 + 0.011 + 0.04 + 66.67 + 3.45 \\ &= 75.13\text{°C/W} \end{aligned}$$

Knowing the junction-to-leads thermal resistance, the junction temperature is determined from

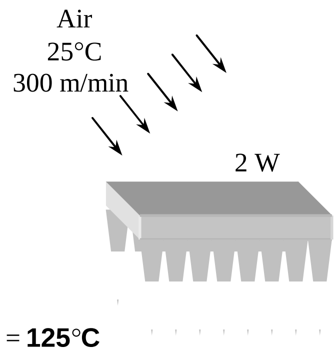
$$\dot{Q} = \frac{T_{junction} - T_{leads}}{R_{junction-case}}$$

$$T_{junction} = T_{leads} + \dot{Q}R_{junction-case} = 50\text{°C} + (0.8\text{ W})(75.13\text{°C/W}) = \mathbf{110.1°C}$$

15-55 A plastic DIP with 16 leads is cooled by forced air. Using data supplied by the manufacturer, the junction temperature is to be determined.

Assumptions Steady operating conditions exist.

Analysis The junction-to-ambient thermal resistance of the device with 16 leads corresponding to an air velocity of 300 m/min is determined from Fig.15-23 to be



$$R_{junction-ambient} = 50°\text{C/W}$$

Then the junction temperature becomes

$$\dot{Q} = \frac{T_{junction} - T_{ambient}}{R_{junction-ambient}}$$

$$T_{junction} = T_{ambient} + \dot{Q}R_{junction-ambient} = 25°\text{C} + (2\text{ W})(50°\text{C/W}) = \mathbf{125°C}$$

When the fan fails the total thermal resistance is determined from Fig.15-23 by reading the value for zero air velocity (the intersection point of the curve with the vertical axis) to be

$$R_{junction-ambient} = 70°\text{C/W}$$

which yields

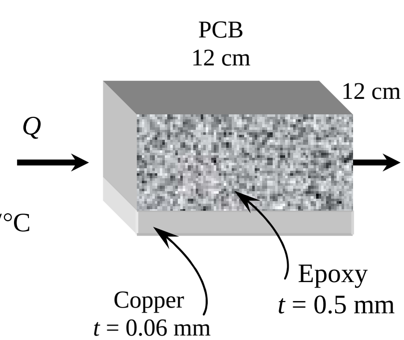
$$\dot{Q} = \frac{T_{junction} - T_{ambient}}{R_{junction-ambient}}$$

$$T_{junction} = T_{ambient} + \dot{Q}R_{junction-ambient} = 25°\text{C} + (2\text{ W})(70°\text{C/W}) = \mathbf{165°C}$$

15-56 A PCB with copper cladding is given. The percentages of heat conduction along the copper and epoxy layers as well as the effective thermal conductivity of the PCB are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat conduction along the PCB is one-dimensional since heat transfer from side surfaces is negligible. **3** The thermal properties of epoxy and copper layers are constant.

Analysis Heat conduction along a layer is proportional to the thermal conductivity-thickness product (kt) which is determined for each layer and the entire PCB to be



$$(kt)_{copper} = (386\text{ W/m.°C})(0.06\times10^{-3}\text{ m}) = 0.02316\text{ W/°C}$$

$$(kt)_{epoxy} = (0.26\text{ W/m.°C})(0.5\times10^{-3}\text{ m}) = 0.00013\text{ W/°C}$$

$$(kt)_{PCB} = (kt)_{copper} + (kt)_{epoxy} = 0.02316 + 0.00013 = 0.02329\text{ W/°C}$$

Therefore the percentages of heat conduction along the epoxy board are

$$f_{epoxy} = \frac{(kt)_{epoxy}}{(kt)_{PCB}} = \frac{0.00013\text{ W/°C}}{0.02316\text{ W/°C}} = 0.0056 \cong \mathbf{0.6\%}$$

and $\quad f_{copper} = (100 - 0.6)\% = \mathbf{99.4\%}$

Then the effective thermal conductivity becomes

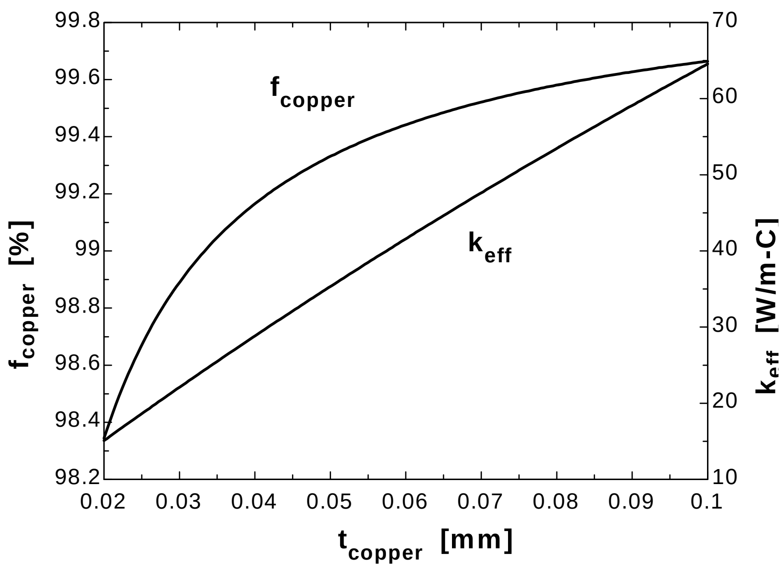
$$k_{eff} = \frac{(kt)_{epoxy} + (kt)_{copper}}{t_{epoxy} + t_{copper}} = \frac{(0.02316 + 0.00013)\text{ W/°C}}{(0.06 + 0.5)\times10^{-3}\text{ m}} = \mathbf{41.6\ W/m.°C}$$

15-57 EES Prob. 15-56 is reconsidered. The effect of the thickness of the copper layer on the percentage of heat conducted along the copper layer and the effective thermal conductivity of the PCB is to be investigated.

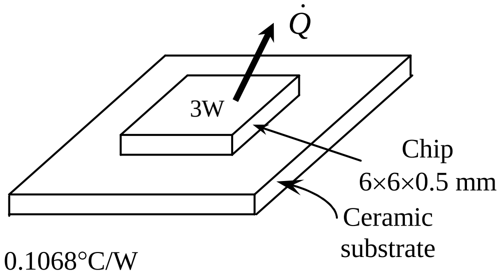
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
length=0.12 [m]
width=0.12 [m]
t_copper=0.06 [mm]
t_epoxy=0.5 [mm]
k_copper=386 [W/m-C]
k_epoxy=0.26 [W/m-C]

"ANALYSIS"
kt_copper=k_copper*t_copper*Convert(mm, m)
kt_epoxy=k_epoxy*t_epoxy*Convert(mm, m)
kt_PCB=kt_copper+kt_epoxy
f_copper=kt_copper/kt_PCB*Convert(, %)
f_epoxy=100-f_copper
k_eff=(kt_epoxy+kt_copper)/((t_epoxy+t_copper)*Convert(mm, m))
```

T_{copper} [mm]	f_{copper} [%]	k_{eff} [W/mC]
0.02	98.34	15.1
0.025	98.67	18.63
0.03	98.89	22.09
0.035	99.05	25.5
0.04	99.17	28.83
0.045	99.26	32.11
0.05	99.33	35.33
0.055	99.39	38.49
0.06	99.44	41.59
0.065	99.48	44.64
0.07	99.52	47.63
0.075	99.55	50.57
0.08	99.58	53.47
0.085	99.61	56.31
0.09	99.63	59.1
0.095	99.65	61.85
0.1	99.66	64.55



15-58 The heat generated in a silicon chip is conducted to a ceramic substrate to which it is attached. The temperature difference between the front and back surfaces of the chip is to be determined.



Assumptions **1** Steady operating conditions exist. **2** Heat conduction along the chip is one-dimensional.

Analysis The thermal resistance of silicon chip is

$$R_{chip} = \frac{L}{kA} = \frac{0.5\times 10^{-3}\text{ m}}{(130\text{ W/m.°C})(0.006\times 0.006)\text{m}^2} = 0.1068\text{°C/W}$$

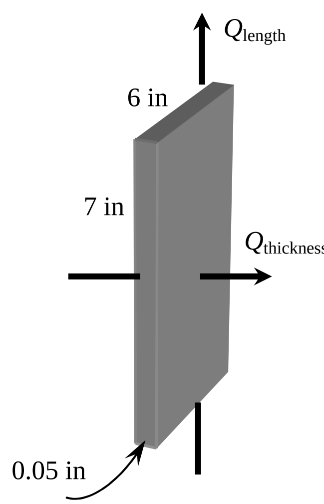
Then the temperature difference across the chip becomes

$$\Delta T = \dot{Q}R_{chip} = (3\text{ W})(0.1068\text{°C/W}) = \mathbf{0.32°C}$$

15-59E The dimensions of an epoxy glass laminate are given. The thermal resistances for heat flow along the layers and across the thickness are to be determined.

Assumptions **1** Heat conduction in the laminate is one-dimensional in either case. **2** Thermal properties of the laminate are constant.

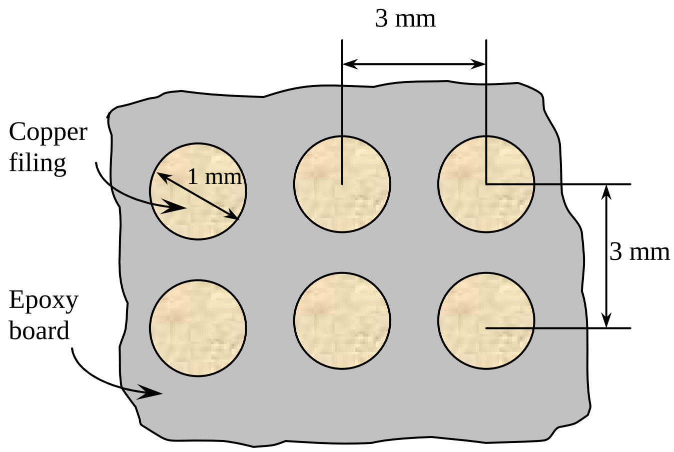
Analysis The thermal resistances of the PCB along the 7 in long side and across its thickness are



(*a*)
$$R_{\substack{along\\length}} = \frac{L}{kA} = \frac{(7/12)\text{ ft}}{(0.15\text{ Btu/h.ft.°F})(6/12\text{ ft})(0.05/12\text{ ft})} = \mathbf{1867\ h.°F/Btu}$$

(*b*)
$$R_{\substack{across\\thickness}} = \frac{L}{kA} = \frac{(0.05/12)\text{ ft}}{(0.15\text{ Btu/h.ft.°F})(7/12\text{ ft})(6/12\text{ ft})} = \mathbf{0.095\ h.°F/Btu}$$

15-60 Cylindrical copper fillings are planted throughout an epoxy glass board. The thermal resistance of the board across its thickness is to be determined.



Assumptions **1** Heat conduction along the board is one-dimensional. **2** Thermal properties of the board are constant.

Analysis The number of copper fillings on the board is

$$n = \frac{\text{Area of board}}{\text{Area of one square}} = \frac{(150\text{ mm})(180\text{ mm})}{(3\text{ mm})(3\text{ mm})} = 3000$$

The surface areas of the copper fillings and the remaining part of the epoxy layer are

$$A_{copper} = n\frac{\pi D^2}{4} = (3000)\frac{\pi(0.001\text{ m})^2}{4} = 0.002356\text{ m}^2$$

$$A_{total} = (length)(width) = (0.15\text{ m})(0.18\text{ m}) = 0.027\text{ m}^2$$

$$A_{epoxy} = A_{total} - A_{copper} = 0.027 - 0.002356 = 0.024644\text{ m}^2$$

The thermal resistance of each material is

$$R_{copper} = \frac{L}{kA} = \frac{0.0014\text{ m}}{(386\text{ W/m.°C})(0.002356\text{ m}^2)} = 0.00154\text{°C/W}$$

$$R_{epoxy} = \frac{L}{kA} = \frac{0.0014\text{ m}}{(0.26\text{ W/m.°C})(0.024644\text{ m}^2)} = 0.2185\text{°C/W}$$

Since these two resistances are in parallel, the equivalent thermal resistance of the entire board is

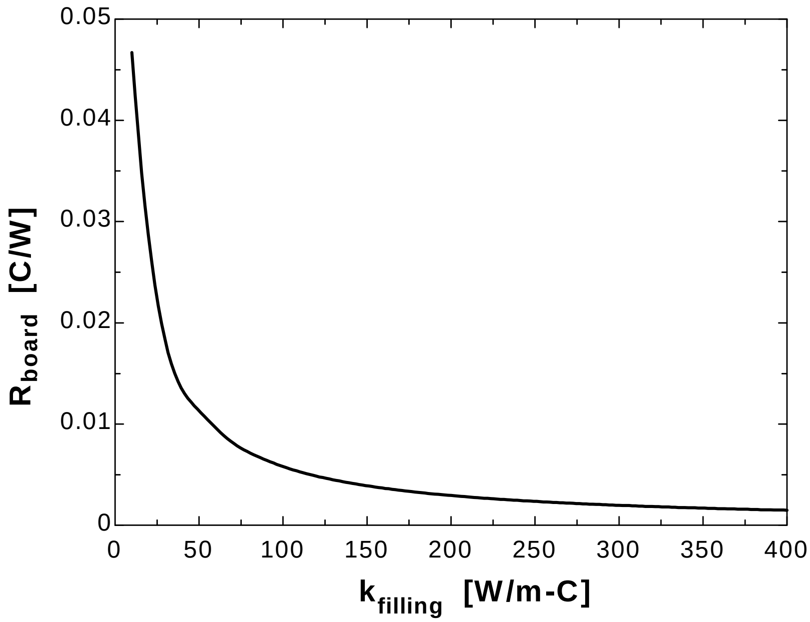
$$\frac{1}{R_{board}} = \frac{1}{R_{epoxy}} + \frac{1}{R_{copper}} = \frac{1}{0.2185\text{°C/W}} + \frac{1}{0.00154\text{°C/W}} \longrightarrow R_{board} = \mathbf{0.00153\text{°C/W}}$$

15-61 EES Prob. 15-60 is reconsidered. The effects of the thermal conductivity and the diameter of the filling material on the thermal resistance of the epoxy board are to be investigated.

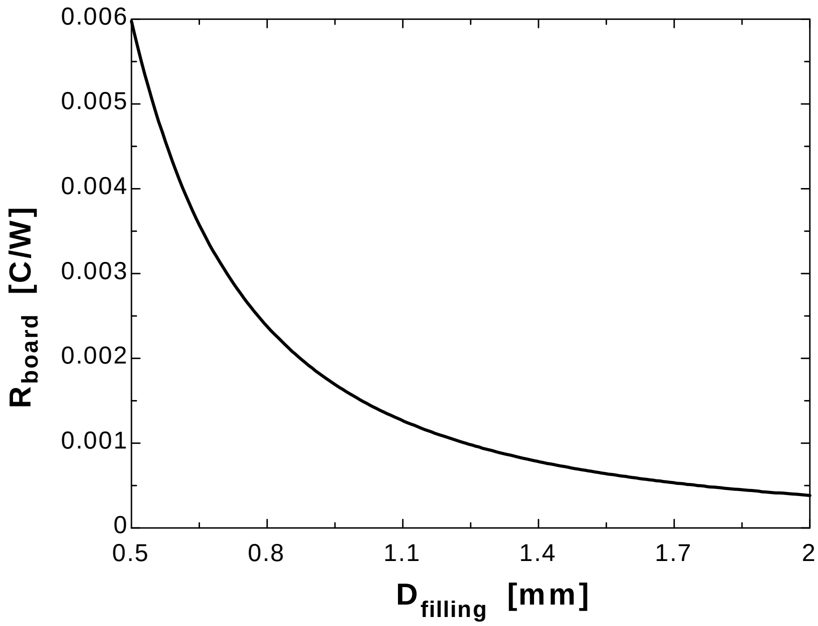
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
length=0.18 [m]
width=0.15 [m]
k_epoxy=0.26 [W/m-C]
t_board=(1.4/1000) [m]
k_filling=386 [W/m-C]
D_filling=1 [mm]
s=(3/1000) [m]

"ANALYSIS"
A_board=length*width
n_filling=A_board/s^2
A_filling=n_filling*pi*(D_filling*Convert(mm, m))^2/4
A_epoxy=A_board-A_filling
R_filling=t_board/(k_filling*A_filling)
R_epoxy=t_board/(k_epoxy*A_epoxy)
1/R_board=1/R_epoxy+1/R_filling
```

$k_{filling}$ [W/m-C]	R_{board} [C/W]
10	0.04671
29.5	0.01844
49	0.01149
68.5	0.008343
88	0.00655
107.5	0.005391
127	0.00458
146.5	0.003982
166	0.003522
185.5	0.003157
205	0.00286
224.5	0.002615
244	0.002408
263.5	0.002232
283	0.00208
302.5	0.001947
322	0.00183
341.5	0.001726
361	0.001634
380.5	0.00155
400	0.001475

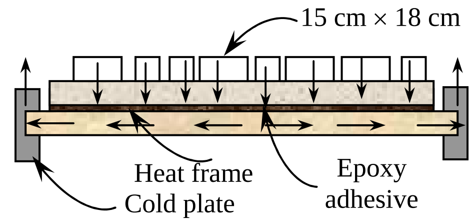


$D_{filling}$ [mm]	R_{board} [C/W]
0.5	0.005977
0.6	0.004189
0.7	0.003095
0.8	0.002378
0.9	0.001884
1	0.001529
1.1	0.001265
1.2	0.001064
1.3	0.0009073
1.4	0.0007828
1.5	0.0006823
1.6	0.0005999
1.7	0.0005316
1.8	0.0004743
1.9	0.0004258
2	0.0003843



15-62 A circuit board with uniform heat generation is to be conduction cooled by a copper heat frame. Temperature distribution along the heat frame and the maximum temperature in the PCB are to be determined.

Assumptions **1** Steady operating conditions exist **2** Thermal properties are constant. **3** There is no direct heat dissipation from the surface of the PCB, and thus all the heat generated is conducted by the heat frame to the heat sink.



Analysis The properties and dimensions of various section of the PCB are summarized below as

Section and material	Thermal conductivity	Thickness	Heat transfer surface area
Epoxy board	0.26 W/m.°C	2 mm	10 mm × 120 mm
Epoxy adhesive	1.8 W/m.°C	0.12 mm	10 mm × 120 mm
Copper heat frame (normal to frame)	386 W/m.°C	1.5 mm	10 mm × 120 mm
Copper heat frame (along the frame)	386 W/m.°C	10 mm	15 mm × 120 mm

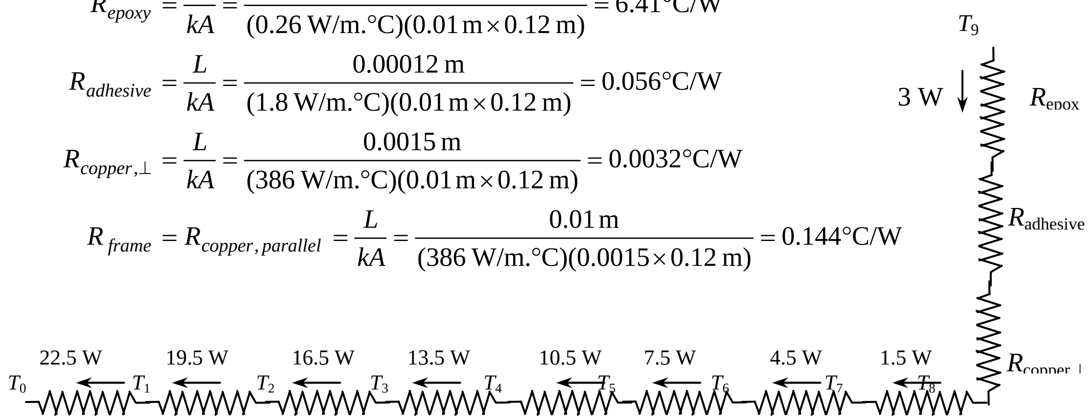
Using the values in the table, the various thermal resistances are determined to be

$$R_{epoxy} = \frac{L}{kA} = \frac{0.002\text{ m}}{(0.26\text{ W/m.°C})(0.01\text{ m}\times 0.12\text{ m})} = 6.41\text{°C/W}$$

$$R_{adhesive} = \frac{L}{kA} = \frac{0.00012\text{ m}}{(1.8\text{ W/m.°C})(0.01\text{ m}\times 0.12\text{ m})} = 0.056\text{°C/W}$$

$$R_{copper,\perp} = \frac{L}{kA} = \frac{0.0015\text{ m}}{(386\text{ W/m.°C})(0.01\text{ m}\times 0.12\text{ m})} = 0.0032\text{°C/W}$$

$$R_{frame} = R_{copper,parallel} = \frac{L}{kA} = \frac{0.01\text{ m}}{(386\text{ W/m.°C})(0.0015\times 0.12\text{ m})} = 0.144\text{°C/W}$$



The combined resistance between the electronic components on each strip and the heat frame can be determined by adding the three thermal resistances in series to be

$$R_{vertical} = R_{epoxy} + R_{adhesive} + R_{copper,\perp} = 6.41 + 0.056 + 0.0032 = 6.469\text{°C/W}$$

The temperatures along the heat frame can be determined from the relation $\Delta T = T_{high} - T_{low} = \dot{Q}R$. Then,

$$T_1 = T_0 + \dot{Q}_{1-0}R_{1-0} = 30\text{°C} + (22.5\text{ W})(0.144\text{°C/W}) = \mathbf{33.24°C}$$

$$T_2 = T_1 + \dot{Q}_{2-1}R_{2-1} = 33.24\text{°C} + (19.5\text{ W})(0.144\text{°C/W}) = \mathbf{36.05°C}$$

$$T_3 = T_2 + \dot{Q}_{3-2}R_{3-2} = 36.05\text{°C} + (16.5\text{ W})(0.144\text{°C/W}) = \mathbf{38.42°C}$$

$$T_4 = T_3 + \dot{Q}_{4-3}R_{4-3} = 38.42\text{°C} + (13.5\text{ W})(0.144\text{°C/W}) = \mathbf{40.36°C}$$

$$T_5 = T_4 + \dot{Q}_{5-4}R_{5-4} = 40.36\text{°C} + (10.5\text{ W})(0.144\text{°C/W}) = \mathbf{41.87°C}$$

$$T_6 = T_5 + \dot{Q}_{6-5}R_{6-5} = 41.87\text{°C} + (7.5\text{ W})(0.144\text{°C/W}) = \mathbf{42.95°C}$$

$$T_7 = T_6 + \dot{Q}_{7-6}R_{7-6} = 42.95\text{°C} + (4.5\text{ W})(0.144\text{°C/W}) = \mathbf{43.60°C}$$

$$T_8 = T_7 + \dot{Q}_{8-7}R_{8-7} = 43.88\text{°C} + (1.5\text{ W})(0.144\text{°C/W}) = \mathbf{43.81°C}$$

The maximum surface temperature on the PCB is

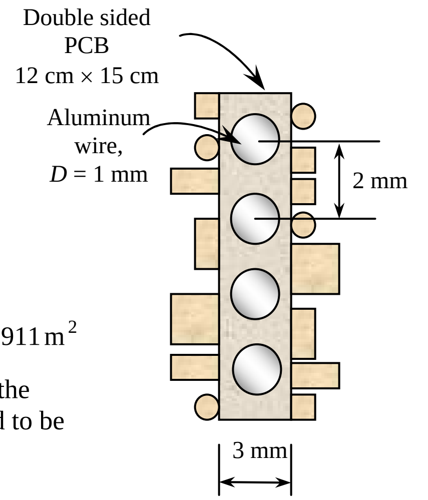
$$T_{\max} = T_9 = T_8 + \dot{Q}_{vertical}R_{vertical} = 43.81\text{°C} + (3\text{ W})(6.469\text{°C/W}) = \mathbf{63.2°C}$$

15-63 A circuit board with uniform heat generation is to be conduction cooled by aluminum wires inserted into it. The magnitude and location of the maximum temperature in the PCB is to be determined.

Assumptions **1** Steady operating conditions exist **2** Thermal properties are constant. **3** There is no direct heat dissipation from the surface of the PCB.

Analysis The number of wires in the board is

$$n = \frac{150\ \text{mm}}{2\ \text{mm}} = 75$$



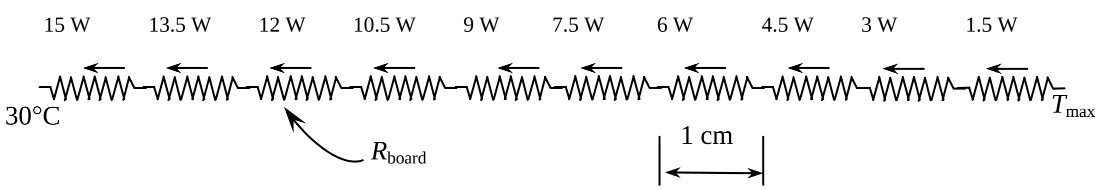
The surface areas of the aluminum wires and the remaining part of the epoxy layer are

$$A_{aluminum} = n\frac{\pi D^2}{4} = (75)\frac{\pi(0.001\ \text{m})^2}{4} = 0.0000589\ \text{m}^2$$

$$A_{total} = (length)(width) = (0.003\ \text{m})(0.15\ \text{m}) = 0.00045\ \text{m}^2$$

$$A_{epoxy} = A_{total} - A_{aluminum} = 0.00045 - 0.0000589 = 0.0003911\ \text{m}^2$$

Considering only half of the circuit board because of symmetry, the thermal resistance of each material per 1-cm length is determined to be



$$R_{aluminum} = \frac{L}{kA} = \frac{0.01\ \text{m}}{(237\ \text{W/m.°C})(0.0000589\ \text{m}^2)} = 0.716\text{°C/W}$$

$$R_{epoxy} = \frac{L}{kA} = \frac{0.01\ \text{m}}{(0.26\ \text{W/m.°C})(0.0003911\ \text{m}^2)} = 98.34\text{°C/W}$$

Since these two resistances are in parallel, the equivalent thermal resistance per cm is determined from

$$\frac{1}{R_{board}} = \frac{1}{R_{epoxy}} + \frac{1}{R_{aluminum}} = \frac{1}{0.716\text{°C/W}} + \frac{1}{98.34\text{°C/W}} \longrightarrow R_{board} = 0.711\text{°C/W}$$

Maximum temperature occurs in the middle of the plate along the 20 cm length, which is determined to be

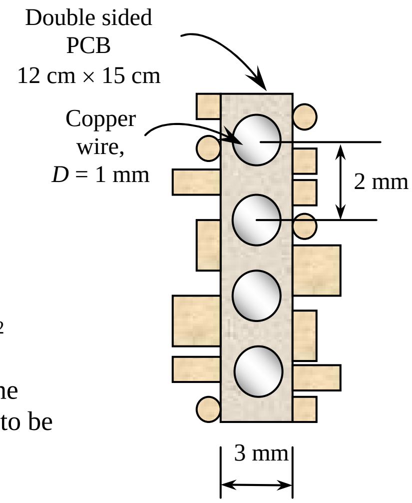
$$T_{\max} = T_{end} + \Delta T_{board,total} = T_{end} + \sum \dot{Q}_i R_{board,1-cm} = T_{end} + R_{board,1-cm}\sum \dot{Q}_i$$
$$= 30\text{°C} + (0.711\text{°C/W})(15+13.5+12+10.5+9+7.5+6+4.5+3+1.5)\text{W} = \mathbf{88.7°C}$$

15-64 A circuit board with uniform heat generation is to be conduction cooled by copper wires inserted in it. The magnitude and location of the maximum temperature in the PCB is to be determined.

Assumptions **1** Steady operating conditions exist **2** Thermal properties are constant. **3** There is no direct heat dissipation from the surface of the PCB.

Analysis The number of wires in the circuit board is

$$n = \frac{150\text{ mm}}{2\text{ mm}} = 75$$



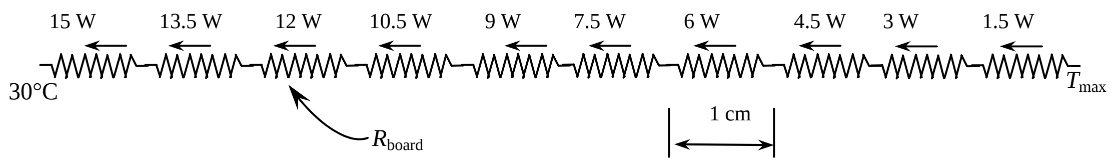
The surface areas of the copper wires and the remaining part of the epoxy layer are

$$A_{copper} = n\frac{\pi D^2}{4} = (75)\frac{\pi(0.001\text{ m})^2}{4} = 0.0000589\text{ m}^2$$

$$A_{total} = (length)(width) = (0.003\text{ m})(0.15\text{ m}) = 0.00045\text{ m}^2$$

$$A_{epoxy} = A_{total} - A_{copper} = 0.00045 - 0.0000589 = 0.0003911\text{ m}^2$$

Considering only half of the circuit board because of symmetry, the thermal resistance of each material per 1-cm length is determined to be



$$R_{copper} = \frac{L}{kA} = \frac{0.01\text{ m}}{(386\text{ W/m.°C})(0.0000589\text{ m}^2)} = 0.440\text{°C/W}$$

$$R_{epoxy} = \frac{L}{kA} = \frac{0.01\text{ m}}{(0.26\text{ W/m.°C})(0.0003911\text{ m}^2)} = 98.34\text{°C/W}$$

Since these two resistances are in parallel, the equivalent thermal resistance is determined from

$$\frac{1}{R_{board}} = \frac{1}{R_{epoxy}} + \frac{1}{R_{copper}} = \frac{1}{0.440\text{°C/W}} + \frac{1}{98.34\text{°C/W}} \longrightarrow R_{board} = 0.438\text{°C/W}$$

Maximum temperature occurs in the middle of the plate along the 20 cm length which is determined to be

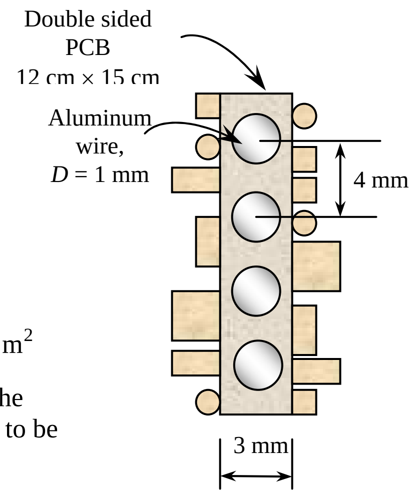
$$T_{\max} = T_{end} + \Delta T_{board,total} = T_{end} + \sum \dot{Q}_i R_{board,1-cm} = T_{end} + R_{board,1-cm}\sum \dot{Q}_i$$
$$= 30\text{°C} + (0.438\text{°C/W})(15+13.5+12+10.5+9+7.5+6+4.5+3+1.5)\text{W} = \mathbf{66.1°C}$$

15-65 A circuit board with uniform heat generation is to be conduction cooled by aluminum wires inserted into it. The magnitude and location of the maximum temperature in the PCB is to be determined.

Assumptions **1** Steady operating conditions exist **2** Thermal properties are constant. **3** There is no direct heat dissipation from the surface of the PCB.

Analysis The number of wires in the board is

$$n = \frac{150\text{ mm}}{4\text{ mm}} = 37$$



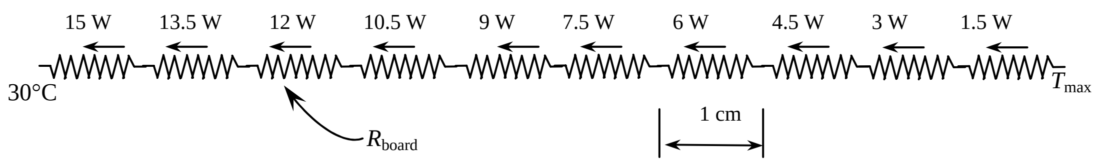
The surface areas of the aluminum wires and the remaining part of the epoxy layer are

$$A_{alu\min um} = n\frac{\pi D^2}{4} = (37)\frac{\pi(0.001\text{ m})^2}{4} = 0.000029\text{ m}^2$$

$$A_{total} = (length)(width) = (0.003\text{ m})(0.15\text{ m}) = 0.00045\text{ m}^2$$

$$A_{epoxy} = A_{total} - A_{alu\min um} = 0.00045 - 0.000029 = 0.000421\text{ m}^2$$

Considering only half of the circuit board because of symmetry, the thermal resistance of each material per 1-cm length is determined to be



$$R_{alu\min um} = \frac{L}{kA} = \frac{0.01\text{ m}}{(237\text{ W/m.°C})(0.000029\text{ m}^2)} = 1.455\text{°C/W}$$

$$R_{epoxy} = \frac{L}{kA} = \frac{0.01\text{ m}}{(0.26\text{ W/m.°C})(0.000421\text{ m}^2)} = 91.36\text{°C/W}$$

Since these two resistances are in parallel, the equivalent thermal resistance is determined from

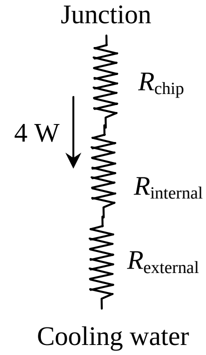
$$\frac{1}{R_{board}} = \frac{1}{R_{epoxy}} + \frac{1}{R_{alu\min um}} = \frac{1}{1.455\text{°C/W}} + \frac{1}{91.36\text{°C/W}} \longrightarrow R_{board} = 1.432\text{°C/W}$$

Maximum temperature occurs in the middle of the plate along the 20 cm length which is determined to be

$$T_{\max} = T_{end} + \Delta T_{board,total} = T_{end} + \sum \dot{Q}_i R_{board,1-cm} = T_{end} + R_{board,1-cm}\sum \dot{Q}_i$$

$$= 30\text{°C} + (1.432\text{°C/W})(15 + 13.5 + 12 + 10.5 + 9 + 7.5 + 6 + 4.5 + 3 + 1.5)\text{W} = \mathbf{148.1°C}$$

15-66 A thermal conduction module with 80 chips is cooled by water. The junction temperature of the chip is to be determined.



Assumptions **1** Steady operating conditions exist **2** Heat transfer through various components is one-dimensional.

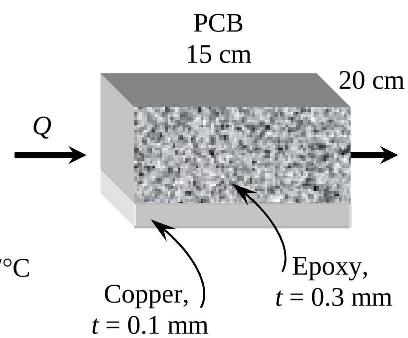
Analysis The total thermal resistance between the junction and cooling water is

$$R_{total} = R_{junction-water} = R_{chip} + R_{internal} + R_{external} = 1.2 + 9 + 7 = 17.2°\text{C}$$

Then the junction temperature becomes

$$T_{junction} = T_{water} + \dot{Q}R_{junction-water} = 18°\text{C} + (4\text{ W})(17.2°\text{C/W}) = \mathbf{86.8°C}$$

15-67 A layer of copper is attached to the back surface of an epoxy board. The effective thermal conductivity of the board and the fraction of heat conducted through copper are to be determined.



Assumptions **1** Steady operating conditions exist **2** Heat transfer is one-dimensional.

Analysis Heat conduction along a layer is proportional to the thermal conductivity-thickness product (kt) which is determined for each layer and the entire PCB to be

$$(kt)_{copper} = (386\text{ W/m.°C})(0.0001\text{ m}) = 0.0386\text{ W/°C}$$

$$(kt)_{epoxy} = (0.26\text{ W/m.°C})(0.0003\text{ m}) = 0.000078\text{ W/°C}$$

$$(kt)_{PCB} = (kt)_{copper} + (kt)_{epoxy} = 0.0386 + 0.000078 = 0.038678\text{ W/°C}$$

The effective thermal conductivity can be determined from

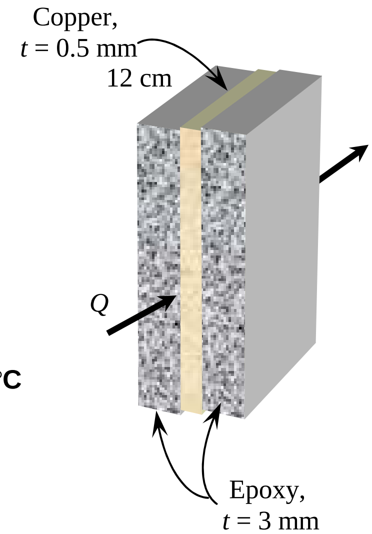
$$k_{eff} = \frac{(kt)_{epoxy} + (kt)_{copper}}{t_{epoxy} + t_{copper}} = \frac{(0.0386 + 0.000078)\text{ W/°C}}{(0.0003\text{ m} + 0.0001\text{ m})} = \mathbf{96.7\ W/m.°C}$$

Then the fraction of the heat conducted along the copper becomes

$$f = \frac{(kt)_{copper}}{(kt)_{PCB}} = \frac{0.0386\text{ W/°C}}{0.038678\text{ W/°C}} = 0.998 = \mathbf{99.8\%}$$

Discussion Note that heat is transferred almost entirely through the copper layer.

15-68 A copper plate is sandwiched between two epoxy boards. The effective thermal conductivity of the board and the fraction of heat conducted through copper are to be determined.



Assumptions **1** Steady operating conditions exist **2** Heat transfer is one-dimensional.

Analysis Heat conduction along a layer is proportional to the thermal conductivity-thickness product (kt) which is determined for each layer and the entire PCB to be

$$(kt)_{copper} = (386\text{ W/m.°C})(0.0005\text{ m}) = 0.193\text{ W/°C}$$

$$(kt)_{epoxy} = (2)(0.26\text{ W/m.°C})(0.003\text{ m}) = 0.00156\text{ W/°C}$$

$$(kt)_{PCB} = (kt)_{copper} + (kt)_{epoxy} = 0.193 + 0.00156 = 0.19456\text{ W/°C}$$

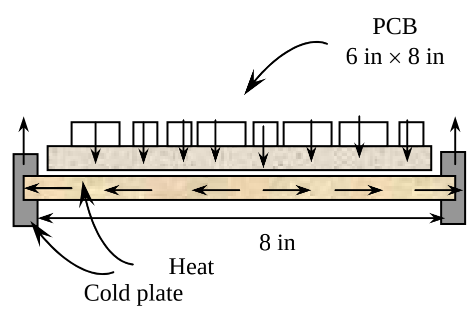
The effective thermal conductivity can be determined from

$$k_{eff} = \frac{(kt)_{epoxy} + (kt)_{copper}}{t_{epoxy} + t_{copper}} = \frac{(0.00156 + 0.193)\text{ W/°C}}{[(2\times 0.003\text{ m}) + 0.0005\text{ m}]} = \mathbf{29.9\ W/m.°C}$$

Then the fraction of the heat conducted along the copper becomes

$$f = \frac{(kt)_{copper}}{(kt)_{PCB}} = \frac{0.193\text{ W/°C}}{0.19456\text{ W/°C}} = 0.992 = \mathbf{99.2\%}$$

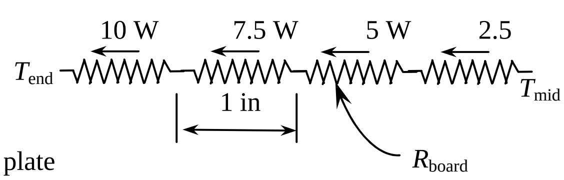
15-69E A copper heat frame is used to conduct heat generated in a PCB. The temperature difference between the mid section and either end of the heat frame is to be determined.



Assumptions **1** Steady operating conditions exist **2** Heat transfer is one-dimensional.

Analysis We assume heat is generated uniformly on the 6 in × 8 in board, and all the heat generated is conducted by the heat frame along the 8-in side. Noting that the rate of heat transfer along the heat frame is variable, we consider 1 in × 8 in strips of the board. The rate of heat generation in each strip is (20 W)/8 = 2.5 W, and the thermal resistance along each strip of the heat frame is

$$R_{frame} = \frac{L}{kA} = \frac{(1/12)\text{ ft}}{(223\text{ Btu/h.ft.°F})(6/12\text{ ft})(0.06/12\text{ ft})} = 0.149\text{ h.°F/Btu}$$

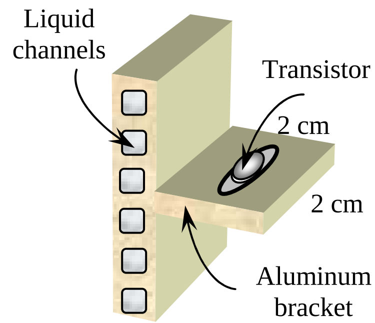


Maximum temperature occurs in the middle of the plate along the 20 cm length. Then the temperature difference between the mid section and either end of the heat frame becomes

$$\Delta T_{max} = \Delta T_{\text{mid section - edge of frame}} = \sum \dot{Q}_i R_{frame,1-in} = R_{frame,1-in} \sum \dot{Q}_i$$
$$= (0.149\text{°F.h/Btu})(10 + 7.5 + 5 + 2.5\ \text{ W})(3.4121\text{ Btu/h.W}) = \mathbf{12.8°F}$$

15-70 A power transistor is cooled by mounting it on an aluminum bracket that is attached to a liquid-cooled plate. The temperature of the transistor case is to be determined.

Assumptions **1** Steady operating conditions exist
2 Conduction heat transfer is one-dimensional.



Analysis The rate of heat transfer by conduction is

$$\dot{Q}_{conduction} = (0.80)(12\text{ W}) = 9.6\text{ W}$$

The thermal resistance of aluminum bracket and epoxy adhesive are

$$R_{aluminum} = \frac{L}{kA} = \frac{0.01\text{ m}}{(237\text{ W/m.°C})(0.003\text{ m})(0.02\text{ m})} = 0.703\text{°C/W}$$

$$R_{epoxy} = \frac{L}{kA} = \frac{0.0002\text{ m}}{(1.8\text{ W/m.°C})(0.003\text{ m})(0.02\text{ m})} = 1.852\text{°C/W}$$

The total thermal resistance between the transistor and the cold plate is

$$R_{total} = R_{case-cold\ \ plate} = R_{plastic} + R_{epoxy} + R_{aluminum} = 2.5 + 1.852 + 0.703 = 5.055\text{°C/W}$$

Then the temperature of the transistor case is determined from

$$T_{case} = T_{\substack{cold \\ plate}} + \dot{Q}R_{case-cold\ plate} = 50\text{°C} + (9.6\text{ W})(5.055\text{°C/W}) = \mathbf{98.5°C}$$

Air Cooling: Natural Convection and Radiation

15-71C As the student watches the movie, the temperature of the electronic components in the DVD player will keep increasing because of the blocked air passages. The DVD player eventually may overheat and fail.

15-72C There is no natural convection in space because of the absence of gravity (and because of the absence of a medium outside). However, it can be cooled by radiation since radiation does not need a medium.

15-73C The openings on the side surfaces of a TV, DVD player or other electronic enclosures provide passage ways for the cold air to enter and warm air to leave. If a TV or DVD player is enclosed in a cabinet with no free space around, and if there is no other cooling process involved, the temperature of device will keep rising due to the heat generation in device, which may cause the device to fail eventually.

15-74C The magnitude of radiation, in general, is comparable to the magnitude of natural convection. Therefore, radiation heat transfer should be always considered in the analysis of natural convection cooled electronic equipment.

15-75C The effect of atmospheric pressure to heat transfer coefficient can be written as $h_{conv,P\,atm} = h_{conv,1\,atm}\sqrt{P}$ $\text{(W/m}^2\text{.°C)}$ where P is the air pressure in atmosphere. Therefore, the greater the air pressure, the greater the heat transfer coefficient. The best and the worst orientation for heat transfer from a square surface are vertical and horizontal, respectively, since the former maximizes and the latter minimizes natural convection.

15-76C The view factor from surface 1 to surface 2 is the fraction of radiation which leaves surface 1 and strikes surface 2 directly. The magnitude of radiation heat transfer between two surfaces is proportional to the view factor. The larger the view factor, the larger the radiation exchange between the two surfaces.

15-77C Emissivity of a surface is the ratio of the radiation emitted by a surface at a specified temperature to the radiation emitted by a blackbody (which is the maximum amount) at the same temperature. The magnitude of radiation heat transfer between a surfaces and it surrounding surfaces is proportional to the emissivity. The larger the emissivity, the larger the radiation heat exchange between the two surfaces.

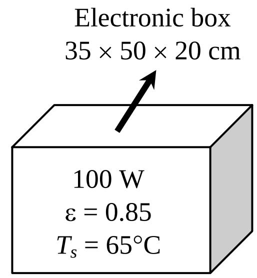
15-78C For most effective natural convection cooling of a PCB array, the PCB should be placed vertically to take advantage of natural convection currents which tend to rise naturally, and to minimize trapped air pockets. Placing the PCBs too close to each other tends to choke the flow because of the increased resistance. Therefore, the PCBs should be placed far from each other for effective heat transfer (A distance of about 2 cm between the PCBs turns out to be adequate for effective natural convection cooling.)

15-79C Radiation heat transfer from the components on the PCBs in an enclosure is negligible since the view of the components is largely blocked by other heat generating components at about the same temperature, and hot components face other hot surfaces instead of cooler surfaces.

15-80 The surface temperature of a sealed electronic box placed on top of a stand is not to exceed 65°C. It is to be determined if this box can be cooled by natural convection and radiation alone.

Assumptions **1** Steady operating conditions exist. **2** The local atmospheric pressure is 1 atm.

Analysis Using Table 15-1, the heat transfer coefficient and the natural convection heat transfer from side surfaces are determined to be



$$L = 0.2\text{ m}$$

$$A_{side} = (2)(0.5\text{ m} + 0.35\text{ m})(0.2\text{ m}) = 0.34\text{ m}^2$$

$$h_{conv,side} = 1.42\left(\frac{\Delta T}{L}\right)^{0.25} = 1.42\left(\frac{65-30}{0.2}\right)^{0.25} = 5.16\text{ W/m.°C}$$

$$\dot{Q}_{conv,side} = h_{conv,side} A_{side}(T_s - T_{fluid})$$
$$= (5.16\text{ W/m.°C})(0.34\text{ m}^2)(65-30)\text{°C} = 61.5\text{ W}$$

The heat transfer from the horizontal top surface by natural convection is

$$L = \frac{4A_{top}}{p} = \frac{4(0.5\text{ m})(0.35\text{ m})}{(2)(0.5\text{ m} + 0.35\text{ m})} = 0.41\text{ m}$$

$$A_{top} = (0.5\text{ m})(0.35\text{ m}) = 0.175\text{ m}^2$$

$$h_{conv,top} = 1.32\left(\frac{\Delta T}{L}\right)^{0.25} = 1.32\left(\frac{65-30}{0.41}\right)^{0.25} = 4.01\text{ W/m.°C}$$

$$\dot{Q}_{conv,top} = h_{conv,top} A_{top}(T_s - T_{fluid}) = (4.01\text{ W/m.°C})(0.175\text{ m}^2)(65-30)\text{°C} = 24.6\text{ W}$$

The rate of heat transfer from the box by radiation is determined from

$$\dot{Q}_{rad} = \varepsilon A_s \sigma (T_s^4 - T_{surr}^4)$$
$$= (0.85)(0.34\text{ m}^2 + 0.175\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2.\text{K}^4)[(65+273\text{ K})^4 - (30+273\text{ K})^4] = 114.7\text{ W}$$

Then the total rate of heat transfer from the box becomes

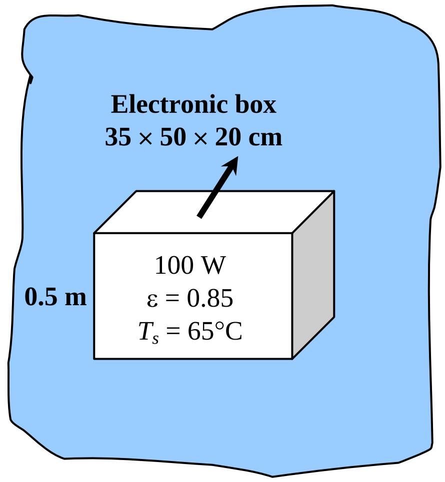
$$\dot{Q}_{total} = \dot{Q}_{conv,side} + \dot{Q}_{conv,top} + \dot{Q}_{rad} = 61.5 + 24.6 + 114.7 = \mathbf{200.8\text{ W}}$$

which is greater than 100 W. Therefore, this box can be cooled by combined natural convection and radiation.

15-81 The surface temperature of a sealed electronic box placed on top of a stand is not to exceed 65°C. It is to be determined if this box can be cooled by natural convection and radiation alone.

Assumptions **1** Steady operating conditions exist. **2** The local atmospheric pressure is 1 atm.

Analysis In given orientation, two side surfaces and the top surface will be vertical and other two side surfaces will be horizontal. Using Table 15-1, the heat transfer coefficient and the natural convection heat transfer from the vertical surfaces are determined to be



$$L = 0.5\text{ m}$$

$$A_{vertical} = (2\times 0.2\times 0.5 + 0.5\times 0.35) = 0.375\text{ m}^2$$

$$h_{\substack{conv\\vertical}} = 1.42\left(\frac{\Delta T}{L}\right)^{0.25} = 1.42\left(\frac{65-30}{0.5}\right)^{0.25} = 4.107\text{ W/m.°C}$$

$$\dot{Q}_{\substack{conv\\vertical}} = h_{\substack{conv\\vertical}} A_{vertical}(T_s - T_{fluid})$$

$$= (4.107\text{ W/m.°C})(0.375\text{ m}^2)(65-30)\text{°C} = 53.9\text{ W}$$

The heat transfer from the horizontal top surface by natural convection is

$$A_{top} = (0.2\text{ m})(0.35\text{ m}) = 0.07\text{ m}^2$$

$$L = \frac{4A_{top}}{p} = \frac{(4)(0.07\text{ m}^2)}{(4)(0.2\text{ m} + 0.35\text{ m})} = 0.1273\text{ m}$$

$$h_{\substack{conv\\top}} = 1.32\left(\frac{\Delta T}{L}\right)^{0.25} = 1.32\left(\frac{65-30}{0.1273}\right)^{0.25} = 5.4\text{ W/m.°C}$$

$$\dot{Q}_{\substack{conv\\top}} = h_{\substack{conv\\top}} A_{top}(T_s - T_{fluid}) = (5.4\text{ W/m.°C})(0.07\text{ m}^2)(65-30)\text{°C} = 13.2\text{ W}$$

The heat transfer from the horizontal top surface by natural convection is

$$h_{\substack{conv\\bottom}} = 0.59\left(\frac{\Delta T}{L}\right)^{0.25} = 0.59\left(\frac{65-30}{0.1273}\right)^{0.25} = 2.4\text{ W/m.°C}$$

$$\dot{Q}_{\substack{conv\\bottom}} = h_{\substack{conv\\bottom}} A_{bottom}(T_s - T_{fluid}) = (2.4\text{ W/m.°C})(0.07\text{ m}^2)(65-30)\text{°C} = 5.9\text{ W}$$

The rate of heat transfer from the box by radiation is determined from

$$\dot{Q}_{rad} = \varepsilon A_s \sigma (T_s{}^4 - T_{surr}{}^4)$$

$$= (0.85)(0.34\text{ m}^2 + 0.175\text{ m}^2)(5.67\times 10^{-8}\text{ W/m}^2.\text{K}^4)[(65+273\text{ K})^4 - (30+273\text{ K})^4] = 114.7\text{ W}$$

Then the total rate of heat transfer from the box becomes

$$\dot{Q}_{total} = \dot{Q}_{\substack{conv\\vertical}} + \dot{Q}_{\substack{conv\\top}} + \dot{Q}_{\substack{conv\\bottom}} + \dot{Q}_{rad} = 53.9 + 13.2 + 5.9 + 114.7 = \mathbf{187.7\text{ W}}$$

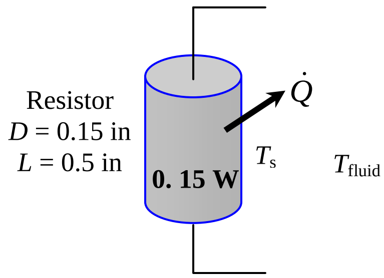
which is greater than 100 W. Therefore, this box can be cooled by combined natural convection and radiation.

15-82E A small cylindrical resistor mounted on a PCB is being cooled by natural convection and radiation. The surface temperature of the resistor is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The local atmospheric pressure is 1 atm. **3** Radiation is negligible in this case since the resistor is surrounded by surfaces which are at about the same temperature, and the radiation heat transfer between two surfaces at the same temperature is zero. This leaves natural convection as the only mechanism of heat transfer from the resistor.

Analysis For components on a circuit board, the heat transfer coefficient relation from Table 15-1 is

$$h_{conv} = 0.50\left(\frac{T_s - T_{fluid}}{D}\right)^{0.25} \qquad (L = D)$$



Substituting it into the heat transfer relation to get

$$\begin{aligned}\dot{Q}_{conv} &= h_{conv} A_s (T_s - T_{fluid}) \\ &= 0.50\left(\frac{T_s - T_{fluid}}{D}\right)^{0.25} A_s (T_s - T_{fluid}) \\ &= 0.50 A_s \frac{(T_s - T_{fluid})^{1.25}}{D^{0.25}}\end{aligned}$$

Calculating surface area and substituting it into above equation for the surface temperature yields

$$A_s = 2\left(\frac{\pi D^2}{4}\right) + \pi DL = 2\left[\frac{\pi (0.15/12 \text{ ft})^2}{4}\right] + \pi (0.15/12 \text{ ft})(0.5/12 \text{ ft}) = 0.00188 \text{ ft}^2$$

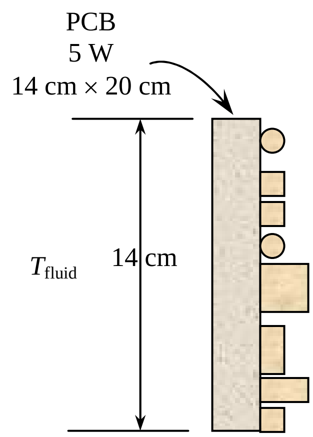
$$(0.15 \text{ W} \times 3.41214 \text{ Btu/h.W}) = (0.50)(0.00188 \text{ ft}^2)\frac{(T_s - 130)^{1.25}}{(0.15/12 \text{ ft})^{0.25}} \longrightarrow T_s = \mathbf{194°F}$$

15-83 The surface temperature of a PCB is not to exceed 90°C. The maximum environment temperatures for safe operation at sea level and at 3,000 m altitude are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation heat transfer is negligible since the PCB is surrounded by other PCBs at about the same temperature. **3** Heat transfer from the back surface of the PCB will be very small and thus negligible.

Analysis Using the simplified relation for a vertical orientation from Table 15-1, the natural convection heat transfer coefficient is determined to be

$$h_{conv} = 1.42\left(\frac{T_s - T_{fluid}}{L}\right)^{0.25}$$



Substituting it into the heat transfer relation to get

$$\begin{aligned}\dot{Q}_{conv} &= h_{conv} A_s (T_s - T_{fluid}) \\ &= 1.42\left(\frac{T_s - T_{fluid}}{L}\right)^{0.25} A_s (T_s - T_{fluid}) \\ &= 1.42 A_s \frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}\end{aligned}$$

Calculating surface area and characteristic length and substituting them into above equation for the surface temperature yields

$$L = 0.14\ \text{m}$$

$$A_s = (0.14\ \text{m})(0.2\ \text{m}) = 0.028\ \text{m}^2$$

$$5\ \text{W} = (1.42)(0.028\ \text{m}^2)\frac{(90 - T_{fluid})^{1.25}}{(0.14\ \text{m})^{0.25}} \longrightarrow T_{fluid} = \mathbf{57.7°C}$$

At an altitude of 3000 m, the atmospheric pressure is 70.12 kPa which is equivalent to

$$P = (70.12\ \text{kPa})\frac{1\ \text{atm}}{101.325\ \text{kPa}} = 0.692\ \text{atm}$$

Modifying the heat transfer relation for this pressure (by multiplying by the square root of it) yields

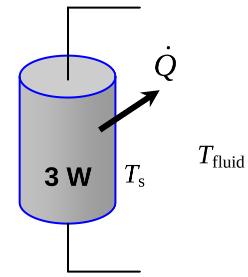
$$5\ \text{W} = (1.42)(0.028\ \text{m}^2)\frac{(90 - T_{fluid})^{1.25}}{(0.14\ \text{m})^{0.25}}\sqrt{0.692} \longrightarrow T_{fluid} = \mathbf{52.6°C}$$

15-84 A cylindrical electronic component is mounted on a board with its axis in the vertical direction. The average surface temperature of the component is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The local atmospheric pressure is 1 atm.

Analysis The natural convection heat transfer coefficient for vertical orientation using Table 15-1 can be determined from

$$h_{conv} = 1.42\left(\frac{T_s - T_{fluid}}{L}\right)^{0.25}$$



Substituting it into the heat transfer relation gives

$$\dot{Q}_{conv} = h_{conv} A_s (T_s - T_{fluid})$$
$$= 1.42\left(\frac{T_s - T_{fluid}}{L}\right)^{0.25} A_s (T_s - T_{fluid})$$
$$= 1.42 A_s \frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

The rate of heat transfer from the cylinder by radiation is

$$\dot{Q}_{rad} = \varepsilon A_s \sigma (T_s{}^4 - T_{surr}{}^4)$$

Then the total rate of heat transfer can be written as

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 1.42 A_s \frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}} + \varepsilon A_s \sigma (T_s{}^4 - T_{surr}{}^4)$$

We will calculate total surface area of the cylindrical component including top and bottom surfaces, and assume the natural heat transfer coefficient to be the same throughout all surfaces of the component

$$A_s = 2\left(\frac{\pi D^2}{4}\right) + \pi DL = 2\left[\frac{\pi (0.02\text{ m})^2}{4}\right] + \pi (0.02\text{ m})(0.04\text{ m}) = 0.00314\text{ m}^2$$

Substituting

$$3\text{ W} = (1.42)(0.00314\text{ m}^2)\frac{[T_s - (30+273\text{ K})]^{1.25}}{(0.04\text{ m})^{0.25}}$$
$$+ (0.8)(0.00314\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{K}^4)[T_s{}^4 - (20+273\text{ K})^4]$$

Solving for the surface temperature gives

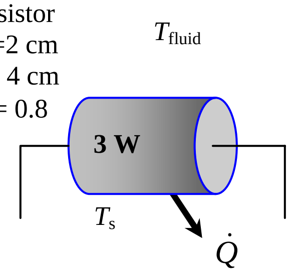
$$T_s = 363\text{ K} = \mathbf{90°C}$$

15-85 A cylindrical electronic component is mounted on a board with its axis in horizontal direction. The average surface temperature of the component is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The local atmospheric pressure is 1 atm.

Analysis Since atmospheric pressure is not given, we assume it to be 1 atm. The natural convection heat transfer coefficient for horizontal orientation using Table 15-1 can be determined from

$$h_{conv} = 1.32\left(\frac{T_s - T_{fluid}}{D}\right)^{0.25}$$



Substituting it into the heat transfer relation to get

$$\dot{Q}_{conv} = h_{conv} A_s (T_s - T_{fluid})$$

$$= 1.32\left(\frac{T_s - T_{fluid}}{D}\right)^{0.25} A_s (T_s - T_{fluid})$$

$$= 1.32 A_s \frac{(T_s - T_{fluid})^{1.25}}{D^{0.25}}$$

The rate of heat transfer from the cylinder by radiation is

$$\dot{Q}_{rad} = \varepsilon A_s \sigma (T_s{}^4 - T_{surr}{}^4)$$

Then the total heat transfer can be written as

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 1.32 A_s \frac{(T_s - T_{fluid})^{1.25}}{D^{0.25}} + \varepsilon A \sigma (T_s{}^4 - T_{surr}{}^4)$$

We will calculate total surface area of the cylindrical component including top and bottom surfaces, and assume the natural heat transfer coefficient to be the same throughout all surfaces of the component

$$A_s = 2\left(\frac{\pi D^2}{4}\right) + \pi D L = 2\left[\frac{\pi (0.02\text{ m})^2}{4}\right] + \pi (0.02\text{ m})(0.04\text{ m}) = 0.00314\text{ m}^2$$

Substituting,

$$3\text{ W} = (1.32)(0.00314\text{ m}^2)\frac{[T_s - (30+273)\text{ K}]^{1.25}}{(0.02\text{ m})^{0.25}}$$
$$+ (0.8)(0.00314\text{ m}^2)(5.67\times 10^{-8}\text{ W/m}^2\text{K}^4)[T_s{}^4 - (20+273\text{ K})^4]$$

Solving for the surface temperature gives

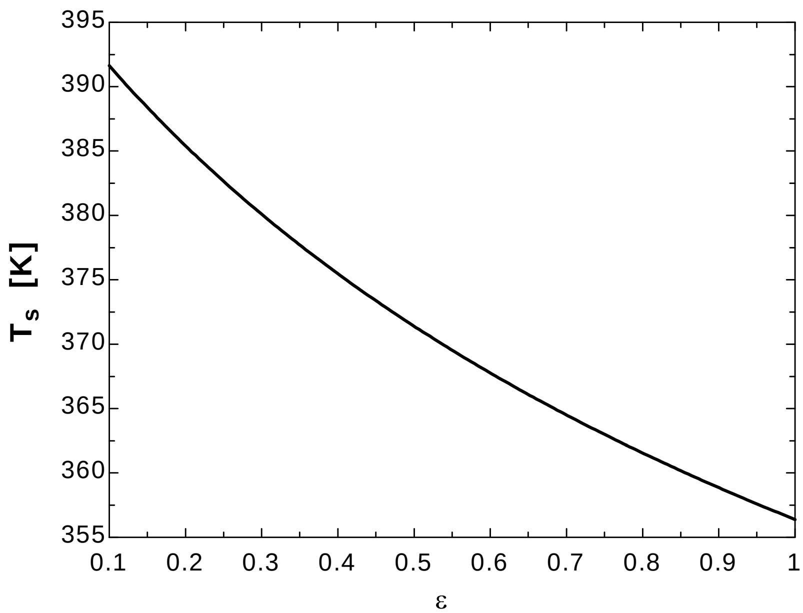
$$T_s = 361\text{ K} = \mathbf{88^\circ C}$$

15-86 EES Prob. 15-84 is reconsidered. The effects of surface emissivity and ambient temperature on the average surface temperature of the component are to be investigated.

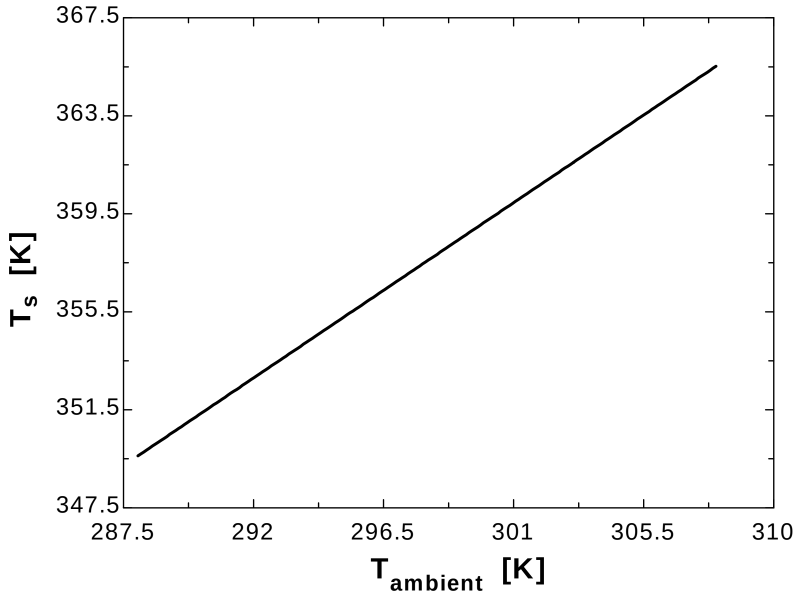
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
D=0.02 [m]
L=0.04 [m]
Q_dot=3 [W]
epsilon=0.8
T_ambient=30+273 "[K]"
T_surr=T_ambient-10

"ANALYSIS"
Q_dot=Q_dot_conv+Q_dot_rad
Q_dot_conv=h*A*(T_s-T_ambient)
h=1.42*((T_s-T_ambient)/L)^0.25
A=2*(pi*D^2)/4+pi*D*L
Q_dot_rad=epsilon*A*sigma*(T_s^4-T_surr^4)
sigma=5.67E-8 [W/m^2-K^4]
```

ε	T_s [K]
0.1	391.6
0.15	388.4
0.2	385.4
0.25	382.6
0.3	380.1
0.35	377.7
0.4	375.5
0.45	373.4
0.5	371.4
0.55	369.5
0.6	367.8
0.65	366.1
0.7	364.5
0.75	363
0.8	361.5
0.85	360.2
0.9	358.9
0.95	357.6
1	356.4



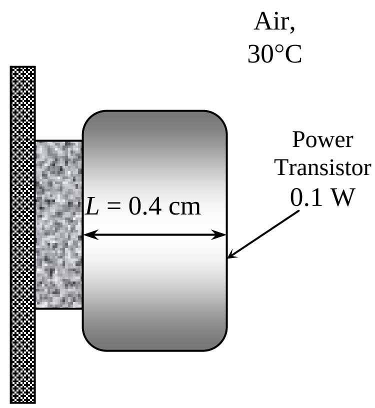
$T_{ambient}$ [K]	T_s [K]
288	349.6
289	350.4
290	351.2
291	352
292	352.8
293	353.6
294	354.4
295	355.2
296	356
297	356.8
298	357.6
299	358.4
300	359.2
301	360
302	360.7
303	361.5
304	362.3
305	363.1
306	363.9
307	364.7
308	365.5



15-87 A power transistor dissipating 0.1 W of power is considered. The heat flux on the surface of the transistor and the surface temperature of the transistor are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat is transferred uniformly from all surfaces of the transistor.

Analysis (*a*) The heat flux on the surface of the transistor is



$$A_s = 2\left(\frac{\pi D^2}{4}\right) + \pi DL$$

$$= 2\left[\frac{\pi(0.4\,\text{cm})^2}{4}\right] + \pi(0.4\,\text{cm})(0.4\,\text{cm}) = 0.754\,\text{cm}^2$$

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{0.1\,\text{W}}{0.754\,\text{cm}^2} = 0.1326\,\text{W/cm}^2$$

(*b*) The surface temperature of the transistor is determined from Newton's law of cooling to be

$$\dot{q} = h_{combined}(T_s - T_{fluid})$$

$$T_s = T_{fluid} + \frac{\dot{q}}{h_{combined}} = 30°\text{C} + \frac{1326\,\text{W/m}^2}{18\,\text{W/m}^2.°\text{C}} = \mathbf{103.7°C}$$

15-88 The components of an electronic equipment located in a horizontal duct with rectangular cross-section are cooled by forced air. The heat transfer from the outer surfaces of the duct by natural convection and the average temperature of the duct are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation heat transfer from the outer surfaces is negligible.

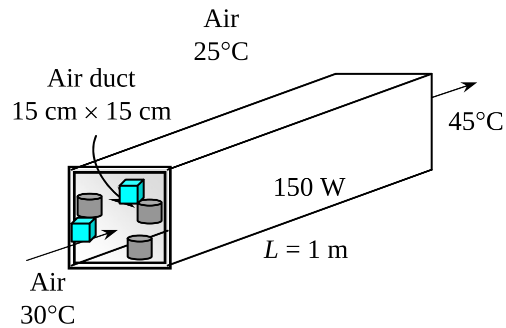
Analysis (*a*) Using air properties at 300 K and 1 atm, the mass flow rate of air and the heat transfer rate by forced convection are determined to be

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1.177\ \text{kg/m}^3)(0.4 / 60\ \text{m}^3/\text{s}) = 0.00785\ \text{kg/s}$$

$$\dot{Q}_{\substack{forced\\convection}} = \dot{m}c_p\Delta T = (0.00785\ \text{kg/s})(1005\ \text{J/kg.°C})(45-30)\text{°C} = 118.3\ \text{W}$$

Noting that radiation heat transfer is negligible, the rest of the 150 W heat generated must be dissipated by natural convection,

$$\dot{Q}_{\substack{natural\\convection}} = \dot{Q}_{total} - \dot{Q}_{\substack{forced\\convection}} = 150 - 118.3 = \mathbf{31.7\ W}$$



(*b*) The natural convection heat transfer from the vertical side surfaces of the duct is

$$A_{side} = 2\times(0.15\ \text{m})(1\ \text{m}) = 0.3\ \text{m}^2$$

$$h_{conv,side} = 1.42\left(\frac{\Delta T}{L}\right)^{0.25}$$

$$\dot{Q}_{conv,side} = h_{conv,side}A_{side}(T_s - T_{fluid}) = 1.42\left(\frac{(T_s - T_{fluid})}{L}\right)^{0.25} A_{side}(T_s - T_{fluid}) = 1.42A_{side}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

Natural convection from the top and bottom surfaces of the duct is

$$L = \frac{4A_{top}}{p} = \frac{(4)(0.15\ \text{m})(1\ \text{m})}{(2)(0.15\ \text{m} + 1\ \text{m})} = 0.26\ \text{m}, \qquad A_{top} = (0.15\ \text{m})(1\ \text{m}) = 0.15\ \text{m}^2$$

$$h_{conv,top} = 1.32\left(\frac{\Delta T}{L}\right)^{0.25}$$

$$\dot{Q}_{conv,top} = h_{conv,top}A_{top}(T_s - T_{fluid})$$

$$= 1.32\left(\frac{(T_s - T_{fluid})}{L}\right)^{0.25} A_{top}(T_s - T_{fluid}) = 1.32A_{top}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

$$h_{conv,bottom} = 0.59\left(\frac{\Delta T}{L}\right)^{0.25}$$

$$\dot{Q}_{conv,bottom} = h_{conv,bot}A_{top}(T_s - T_{fluid})$$

$$= 0.59\left(\frac{(T_s - T_{fluid})}{L}\right)^{0.25} A_{bot}(T_s - T_{fluid}) = 0.59A_{bot}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

Then the total heat transfer by natural convection becomes

$$\dot{Q}_{total,conv} = \dot{Q}_{conv,side} + \dot{Q}_{conv,top} + \dot{Q}_{conv,bottom}$$

$$\dot{Q}_{total,conv} = 1.42A_{side}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}} + 1.32A_{top}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}} + 0.59A_{bottom}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

Substituting all known quantities with proper units gives the average temperature of the duct to be

$$31.7 = (1.42)(0.3)\frac{(T_s - 25)^{1.25}}{0.15^{0.25}} + (1.32)(0.15)\frac{(T_s - 25)^{1.25}}{0.26^{0.25}} + (0.59)(0.15)\frac{(T_s - 25)^{1.25}}{0.26^{0.25}}$$

$$31.7 = (1.086)(T_s - 25)^{1.25} \longrightarrow T_s = \mathbf{40°C}$$

15-89 The components of an electronic equipment located in a circular horizontal duct are cooled by forced air. The heat transfer from the outer surfaces of the duct by natural convection and the average temperature of the duct are to be determined.

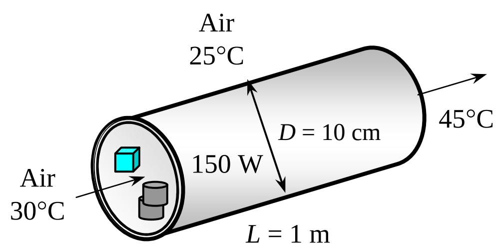
Assumptions **1** Steady operating conditions exist. **2** Radiation heat transfer from the outer surfaces is negligible.

Analysis (*a*) Using air properties at 300 K and 1 atm, the mass flow rate of air and the heat transfer rate by forced convection are determined to be

$$\dot{m} = \rho\dot{V} = (1.177\ \text{kg/m}^3)(0.4/60\ \text{m}^3/\text{s}) = 0.00785\ \text{kg/s}$$

$$\dot{Q}_{\substack{forced\\convection}} = \dot{m}c_p\Delta T = (0.00785\ \text{kg/s})(1005\ \text{J/kg.°C})(45-30)\text{°C} = 118.3\ \text{W}$$

Noting that radiation heat transfer is negligible, the rest of the 150 W heat generated must be dissipated by natural convection,



$$\dot{Q}_{\substack{natural\\convection}} = \dot{Q}_{total} - \dot{Q}_{\substack{forced\\convection}} = 150 - 118.3 = \mathbf{31.7\ W}$$

(*b*) The natural convection heat transfer from the circular duct is

$$L = D = 0.1\ \text{m}$$

$$A_s = 2\left(\frac{\pi D^2}{4}\right) + \pi DL = 2\left[\frac{\pi(0.1\ \text{m})^2}{4}\right] + \pi(0.1\ \text{m})(1\ \text{m}) = 0.33\ \text{m}^2$$

$$h_{conv} = 1.32\left(\frac{\Delta T}{D}\right)^{0.25}$$

$$\dot{Q}_{conv} = h_{conv}A(T_s - T_{fluid}) = 1.32\left(\frac{(T_s - T_{fluid})}{D}\right)^{0.25} A_s(T_s - T_{fluid})$$

$$= 1.32A_s\frac{(T_s - T_{fluid})^{1.25}}{D^{0.25}}$$

Substituting all known quantities with proper units gives the average temperature of the duct to be

$$31.7\ \text{W} = (1.32)(0.33\ \text{m}^2)\frac{(T_s - 25)^{1.25}}{(0.1\ \text{m})^{0.25}} \longrightarrow T_s = \mathbf{44°C}$$

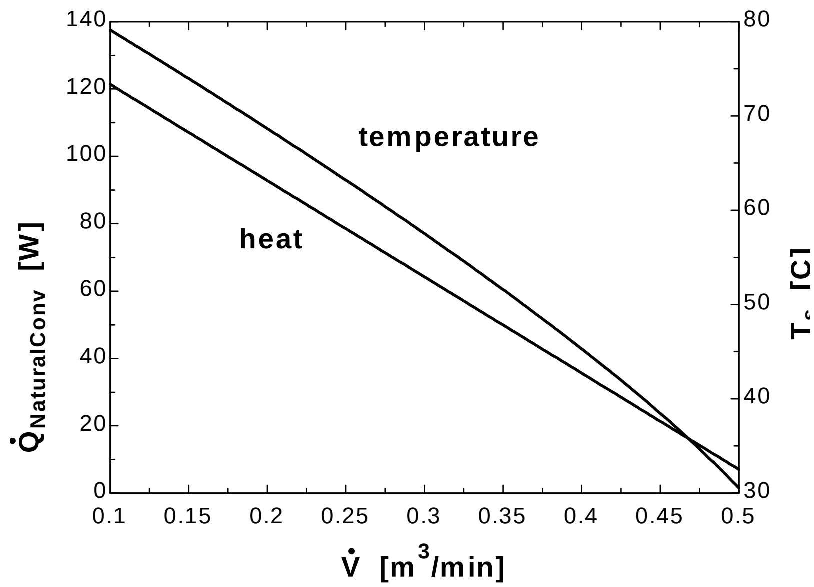
15-90 EES Prob. 15-88 is reconsidered. The effects of the volume flow rate of air and the side-length of the duct on heat transfer by natural convection and the average temperature of the duct are to be investigated.

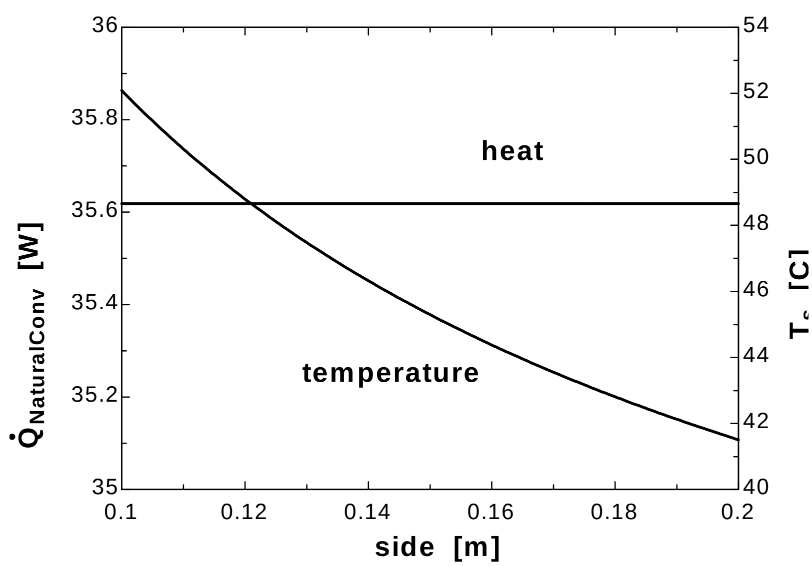
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
Q_dot_total=150 [W]
L=1 [m]
side=0.15 [m]
T_in=30 [C]
T_out=45 [C]
V_dot=0.4 [m^3/min]
T_ambient=25 [C]

"PROPERTIES"
rho=Density(air, T=T_ave, P=101.3)
c_p=CP(air, T=T_ave)*Convert(kJ/kg-C, J/kg-C)
T_ave=1/2*(T_in+T_out)

"ANALYSIS"
"(a)"
m_dot=rho*V_dot*Convert(m^3/min, m^3/s)
Q_dot_ForcedConv=m_dot*c_p*(T_out-T_in)
Q_dot_NaturalConv=Q_dot_total-Q_dot_ForcedConv
"(b)"
A_side=2*side*L
h_conv_side=1.42*((T_s-T_ambient)/L)^0.25
Q_dot_conv_side=h_conv_side*A_side*(T_s-T_ambient)
L_top=(4*A_top)/p_top
A_top=side*L
p_top=2*(side+L)
h_conv_top=1.32*((T_s-T_ambient)/L_top)^0.25
Q_dot_conv_top=h_conv_top*A_top*(T_s-T_ambient)
h_conv_bottom=0.59*((T_s-T_ambient)/L_top)^0.25
Q_dot_conv_bottom=h_conv_bottom*A_top*(T_s-T_ambient)
Q_dot_NaturalConv=Q_dot_conv_side+Q_dot_conv_top+Q_dot_conv_bottom
```

V [m^3/min]	$Q_{NaturalConv}$ [W]	T_s [C]
0.1	121.4	79.13
0.15	107.1	73.97
0.2	92.81	68.66
0.25	78.51	63.19
0.3	64.21	57.52
0.35	49.92	51.58
0.4	35.62	45.29
0.45	21.32	38.46
0.5	7.023	30.54

side [m]	$Q_{NaturalConv}$ [W]	T_s [C]
0.1	35.62	52.08
0.11	35.62	50.31
0.12	35.62	48.8
0.13	35.62	47.48
0.14	35.62	46.32
0.15	35.62	45.29
0.16	35.62	44.38
0.17	35.62	43.55
0.18	35.62	42.81
0.19	35.62	42.13
0.2	35.62	41.5





15-91 The components of an electronic equipment located in a horizontal duct with rectangular cross-section are cooled by forced air. The heat transfer from the outer surfaces of the duct by natural convection and the average temperature of the duct are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation heat transfer from the outer surfaces is negligible.

Analysis In this case the entire 150 W must be dissipated by natural convection from the outer surface of the duct. Natural convection from the vertical side surfaces of the duct can be expressed as

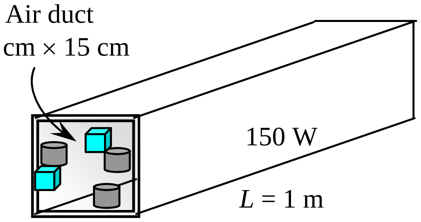
$$L = 0.15\ \text{m}$$

$$A_{side} = 2\times(0.15\,\text{m})(1\,\text{m}) = 0.3\,\text{m}^2$$

$$h_{conv,side} = 1.42\left(\frac{\Delta T}{L}\right)^{0.25}$$

$$\dot{Q}_{conv,side} = h_{conv,side}A_{side}(T_s - T_{fluid}) = 1.42\left(\frac{(T_s - T_{fluid})}{L}\right)^{0.25} A_{side}(T_s - T_{fluid})$$

$$= 1.42A_{side}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$



Natural convection from the top surface of the duct is

$$L = \frac{4A_{top}}{p} = \frac{(4)(0.15\,\text{m})(1\,\text{m})}{(2)(0.15\,\text{m} + 1\,\text{m})} = 0.26\,\text{m}$$

$$A_{top} = (0.15\,\text{m})(1\,\text{m}) = 0.15\,\text{m}^2$$

$$h_{conv,top} = 1.32\left(\frac{\Delta T}{L}\right)^{0.25}$$

$$\dot{Q}_{conv,top} = h_{conv,top}A_{top}(T_s - T_{fluid}) = 1.32\left(\frac{(T_s - T_{fluid})}{L}\right)^{0.25} A_{top}(T_s - T_{fluid})$$

$$= 1.32A_{top}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

Natural convection from the bottom surface of the duct is

$$h_{conv,bottom} = 0.59\left(\frac{\Delta T}{L}\right)^{0.25}$$

$$\dot{Q}_{conv,bottom} = h_{conv,bottom}A_{top}(T_s - T_{fluid}) = 0.59\left(\frac{(T_s - T_{fluid})}{L}\right)^{0.25} A_{bottom}(T_s - T_{fluid})$$

$$= 0.59A_{bottom}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

Then the total heat transfer by natural convection becomes

$$\dot{Q}_{total,conv} = \dot{Q}_{conv,side} + \dot{Q}_{conv,top} + \dot{Q}_{conv,bottom}$$

$$\dot{Q}_{total,conv} = 1.42A_{side}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}} + 1.32A_{top}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}} + 0.59A_{bottom}\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

Substituting all known quantities with proper units gives the average temperature of the duct to be

$$150 = (1.42)(0.3)\frac{(T_s - 25)^{1.25}}{0.15^{0.25}} + (1.32)(0.15)\frac{(T_s - 25)^{1.25}}{0.26^{0.25}} + (0.59)(0.15)\frac{(T_s - 25)^{1.25}}{0.26^{0.25}}$$

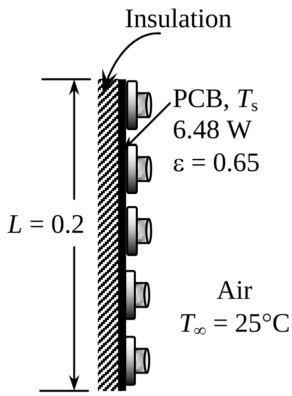
$$150 = (1.086)(T_s - 25)^{1.25} \longrightarrow T_s = \mathbf{77°C}$$

15-92 A wall-mounted circuit board containing 81 square chips is cooled by combined natural convection and radiation. The surface temperature of the chips is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer from the back side of the circuit board is negligible. **3** Temperature of surrounding surfaces is the same as the air temperature. **3** The local atmospheric pressure is 1 atm.

Analysis The natural convection heat transfer coefficient for the vertical orientation of board can be determined from (Table 15-1)

$$h_{conv} = 1.42\left(\frac{T_s - T_{fluid}}{L}\right)^{0.25}$$



Substituting it relation into the heat transfer relation gives

$$\dot{Q}_{conv} = h_{conv} A_s (T_s - T_{fluid})$$

$$= 1.42\left(\frac{T_s - T_{fluid}}{L}\right)^{0.25} A(T_s - T_{fluid}) = 1.42A\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

The rate of heat transfer from the board by radiation is

$$\dot{Q}_{rad} = \varepsilon A_s \sigma\left(T_s^{\,4} - T_{surr}^{\,4}\right)$$

Then the total heat transfer can be expressed as

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 1.42A_s\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}} + \varepsilon A_s \sigma(T_s^{\,4} - T_{surr}^{\,4})$$

where $\dot{Q}_{total} = (0.08\text{ W})\times 81 = 6.48\text{ W}$. Noting that the characteristic length is L = 0.2 m, calculating the surface area and substituting the known quantities into the above equation, the surface temperature is determined to be

$$L = 0.2\text{ m}$$

$$A_s = (0.2\text{ m})(0.2\text{ m}) = 0.04\text{ m}^2$$

$$6.48\text{ W} = (2.44)(0.04\text{ m}^2)\frac{[T_s - (25+273\text{ K})]^{1.25}}{(0.2\text{ m})^{0.25}}$$

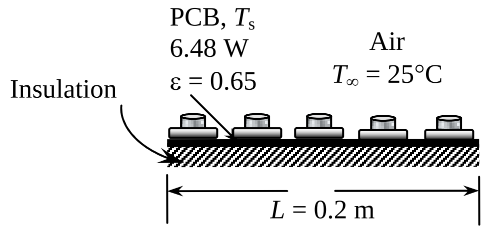
$$+(0.65)(0.04\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{K}^4)\left[T_s^{\,4} - (25+273\text{ K})^4\right]$$

$$T_s = 312.3\text{ K} = \mathbf{39.3°C}$$

15-93 A horizontal circuit board containing 81 square chips is cooled by combined natural convection and radiation. The surface temperature of the chips is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer from the back side of the circuit board is negligible. **3** Temperature of surrounding surfaces is the same as the air temperature. **3** The local atmospheric pressure is 1 atm.

Analysis (*a*) The natural convection heat transfer coefficient for the horizontal orientation of board with chips facing up can be determined from (Table 15-1)



$$h_{conv} = 1.32\left(\frac{T_s - T_{fluid}}{L}\right)^{0.25}$$

Substituting it into the heat transfer relation gives

$$\dot{Q}_{conv} = h_{conv}A_s(T_s - T_{fluid})$$
$$= 1.32\left(\frac{T_s - T_{fluid}}{L}\right)^{0.25} A_s(T_s - T_{fluid}) = 1.32A_s\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}}$$

The rate of heat transfer from the board by radiation is

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s{}^4 - T_{surr}{}^4)$$

Then the total heat transfer can be written as

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 1.32A_s\frac{(T_s - T_{fluid})^{1.25}}{L^{0.25}} + \varepsilon A_s\sigma(T_s{}^4 - T_{surr}{}^4)$$

where $\dot{Q}_{total} = (0.08\text{ W})\times 81 = 6.48\text{ W}$. Noting that the characteristic length is L = 0.2 m, calculating the surface area and substituting the known quantities into the above equation, the surface temperature is determined to be

$$L = \frac{4A_s}{p} = \frac{(4)(0.2\text{ m})(0.2\text{ m})}{(4)(0.2\text{ m})} = 0.2\text{ m}$$
$$A_s = (0.2\text{ m})(0.2\text{ m}) = 0.04\text{ m}^2$$
$$6.48\text{ W} = (1.32)(0.04\text{ m}^2)\frac{(T_s - (25+273)\text{K})^{1.25}}{(0.2\text{ m})^{0.25}}$$
$$+ (0.65)(0.04\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{K}^4)[T_s{}^4 - (25+273\text{ K})^4]$$
$$\longrightarrow T_s = 317.2\text{ K} = \mathbf{44.2°C}$$

(*b*) The solution in this case (the chips are facing down instead of up) is identical to the one above, except we must replace the constant 1.32 in the heat transfer coefficient relation by 0.59. Then the surface temperature in this case becomes

$$6.48\text{ W} = (0.59)(0.04\text{ m}^2)\frac{(T_s - (25+273)\text{K})^{1.25}}{(0.2\text{ m})^{0.25}}$$
$$+ (0.65)(0.04\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{K}^4)[T_s{}^4 - (25+273\text{ K})^4]$$
$$\longrightarrow T_s = 323.3\text{ K} = \mathbf{50.3°C}$$

Air Cooling: Forced Convection

15-94C Radiation heat transfer in forced air cooled systems is usually disregarded with no significant error since the forced convection heat transfer coefficient is usually much larger than the radiation heat transfer coefficient.

15-95C We would definitely prefer natural convection cooling whenever it is adequate in order to avoid all the problems associated with the fans such as cost, power consumption, noise, complexity, maintenance, and possible failure.

15-96C The convection heat transfer coefficient depends strongly on the average fluid velocity. Forced convection usually involves much higher fluid velocities, and thus much higher heat transfer coefficients. Consequently, forced convection cooling is much more effective.

15-97C Increasing the flow rate of air will increase the heat transfer coefficient. Then from Newton's law of cooling $\dot{Q}_{conv} = hA_s(T_s - T_{fluid})$, it becomes obvious that for a fixed amount of power, the temperature difference between the surface and the air will decrease. Therefore, the surface temperature will decrease. The exit temperature of the air will also decrease since $\dot{Q}_{conv} = \dot{m}_{air} c_p (T_{out} - T_{in})$ and the flow rate of air is increased.

15-98C Fluid flow over a body is called external flow, and flow through a confined space such as a tube or the parallel passage area between two circuit boards in an enclosure is called internal flow. A fan cooled personal computer left in windy area involves both types of flow.

15-99C For a specified power dissipation and air inlet temperature, increasing the heat transfer coefficient will decrease the surface temperature of the electronic components since, from Newton's law of cooling, $\dot{Q}_{conv} = hA_s(T_s - T_{fluid})$

15-100C A fan at a fixed speed (or fixed rpm) will deliver a fixed volume of air regardless of the altitude and pressure. But the mass flow rate of air will be less at high altitude as a result of the lower density of air. This may create serious reliability problems and catastrophic failures of electronic equipment if proper precautions are not taken. Variable speed fans which automatically increase the speed when the air density decreases are available to avoid such problems.

15-101C A fan placed at the inlet draws the air in and pressurizes the electronic box, and prevents air infiltration into the box through the cracks or other openings. Having only one location for air inlet makes it practical to install a filter at the inlet to catch all the dust and dirt before they enter the box. This allows the electronic system to operate in a clean environment. Also, the fan placed at the inlet handles cooler and thus denser air which results in a higher mass flow rate for the same volume flow rate or rpm. Being subjected to cool air has the added benefit that it increases the reliability and extends the life of the fan. The major disadvantage associated with the fan mounted at the inlet is that the heat generated by the fan and its motor is picked up by air on its way into the box, which adds to the heat load of the system.

When the fan is placed at the exit, the heat generated by the fan and its motor is immediately discarded to the atmosphere without getting blown first into the electronic box. However, the fan at the exit creates a vacuum inside the box, which draws air into the box through inlet vents as well as any cracks and openings. Therefore, the air is difficult to filter, and the dirt and dust which collects on the components undermine the reliability of the system.

15-102C The volume flow rate of air in a forced-air-cooled electronic system that has a constant speed fan is established at point where the fan static head curve and the system resistance curve intersects. Therefore, a fan will deliver a higher flow rate through a system which offers a lower flow resistance. A few PCBs added into an electronic box will increase the flow resistance and thus decrease the flow rate of air.

15-103C An undersized fan may cause the electronic system to overheat and fail. An oversized fan will definitely provide adequate cooling but it will needlessly be larger, noisier, more expensive, and will consume more power.

15-104 A hollow core PCB is cooled by forced air. The outlet temperature of the air and the highest surface temperature are to be determined.

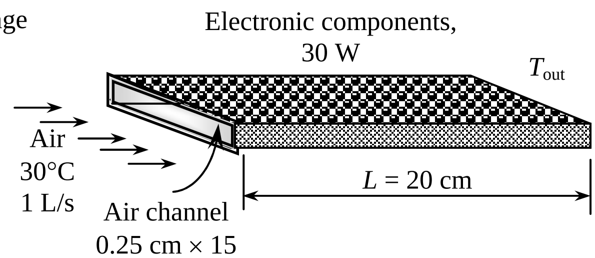
Assumptions **1** Steady operating conditions exist. **2** The inner surfaces of the duct are smooth. **3** Air is an ideal gas. **4** The local atmospheric pressure is 1 atm. **5** The entire heat generated in electronic components is removed by the air flowing through the hollow core.

Properties The air properties at the anticipated average temperature of 40°C and 1 atm (Table A-15) are

$\rho = 1.127 \text{ kg/m}^3 \qquad c_p = 1007 \text{ J/kg.°C}$

$\text{Pr} = 0.7255 \qquad k = 0.02662 \text{ W/m.°C}$

$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$



Analysis (*a*) The cross-sectional area of the channel and its hydraulic diameter are

$$A_c = (height)(width) = (0.15 \text{ m})(0.0025 \text{ m}) = 3.75 \times 10^{-4} \text{ m}^2$$

$$D_h = \frac{4A_c}{p} = \frac{(4)(3.75 \times 10^{-4} \text{ m}^2)}{(2)(0.15 \text{ m} + 0.0025 \text{ m})} = 0.00492 \text{ m}$$

The average velocity and the mass flow rate of air are

$$V = \frac{\dot{V}}{A_c} = \frac{1 \times 10^{-3} \text{ m}^3/\text{s}}{3.75 \times 10^{-4} \text{ m}^2} = 2.67 \text{ m/s}$$

$$\dot{m} = \rho \dot{V} = (1.127 \text{ kg/m}^3)(1 \times 10^{-3} \text{ m}^3/\text{s}) = 1.127 \times 10^{-3} \text{ kg/s}$$

Then the temperature of air at the exit of the hollow core becomes

$$\dot{Q} = \dot{m} c_p (T_{out} - T_{in})$$

$$T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m} c_p} = 30°\text{C} + \frac{30 \text{ W}}{(1.127 \times 10^{-3} \text{ kg/s})(1007 \text{ J/kg.°C})} = \mathbf{56.4°C}$$

(*b*) The highest surface temperature in the channel will occur near the exit, and the surface temperature there can be determined from

$$\dot{q}_{conv} = h(T_s - T_{fluid})$$

To determine heat transfer coefficient, we first need to calculate the Reynolds number,

$$\text{Re} = \frac{VD_h}{\nu} = \frac{(2.67 \text{ m/s})(0.00492 \text{ m})}{1.702 \times 10^{-5} \text{ m}^2/\text{s}} = 771.8 < 2300$$

Therefore the flow is laminar. Assuming fully developed flow, the Nusselt number for the air flow in this rectangular cross-section corresponding to the aspect ratio of $a/b = height/width = 15/0.25 = 60 \approx \infty$ is determined from Table 15-3 to be $Nu = 8.24$. Then,

$$h = \frac{k}{D_h} Nu = \frac{0.02662 \text{ W/m.°C}}{0.00492 \text{ m}}(8.24) = 44.58 \text{ W/m}^2\text{.°C}$$

The surface temperature of the hollow core near the exit is determined to be

$$T_{s,\max} = T_{out} + \frac{\dot{q}}{h} = 56.4°\text{C} + \frac{(30 \text{ W})/(0.06 \text{ m}^2)}{(44.58 \text{ W/m}^2\text{.°C})} = \mathbf{67.6°C}$$

15-105 A hollow core PCB is cooled by forced air. The outlet temperature of the air and the highest surface temperature are to be determined.

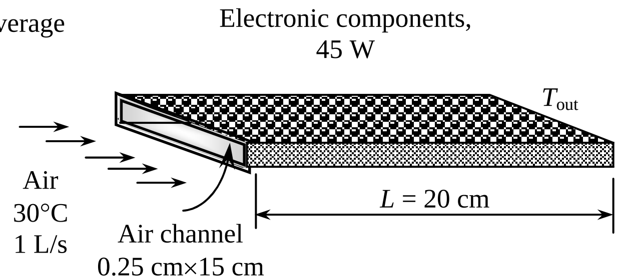
Assumptions **1** Steady operating conditions exist. **2** The inner surfaces of the duct are smooth. **3** Air is an ideal gas. **4** The local atmospheric pressure is 1 atm. **5** The entire heat generated in electronic components is removed by the air flowing through the hollow core.

Properties The air properties at the anticipated average temperature of 40°C and 1 atm (Table A-15) are

$$\rho = 1.127 \text{ kg/m}^3 \qquad c_p = 1007 \text{ J/kg.°C}$$

$$\text{Pr} = 0.7255 \qquad k = 0.02662 \text{ W/m.°C}$$

$$\nu = 1.702\times10^{-5} \text{ m}^2/\text{s}$$



Analysis (*a*) The cross-sectional area of the channel and its hydraulic diameter are

$$A_c = (height)(width) = (0.15 \text{ m})(0.0025 \text{ m}) = 3.75\times10^{-4} \text{ m}^2$$

$$D_h = \frac{4A_c}{p} = \frac{(4)(3.75\times10^{-4} \text{ m}^2)}{(2)(0.15 \text{ m} + 0.0025 \text{ m})} = 0.00492 \text{ m}$$

The average velocity and the mass flow rate of air are

$$V = \frac{\dot{V}}{A_c} = \frac{1\times10^{-3} \text{ m}^3/\text{s}}{3.75\times10^{-4} \text{ m}^2} = 2.67 \text{ m/s}$$

$$\dot{m} = \rho\dot{V} = (1.127 \text{ kg/m}^3)(1\times10^{-3} \text{ m}^3/\text{s}) = 1.127\times10^{-3} \text{ kg/s}$$

Then the temperature of air at the exit of the hollow core becomes

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in})$$

$$T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} = 30°\text{C} + \frac{45 \text{ W}}{(1.127\times10^{-3} \text{ kg/s})(1007 \text{ J/kg.°C})} = \mathbf{69.7°C}$$

(*b*) The highest surface temperature in the channel will occur near the exit, and the surface temperature there can be determined from

$$\dot{q}_{conv} = h(T_s - T_{fluid})$$

To determine heat transfer coefficient, we first need to calculate the Reynolds number,

$$\text{Re} = \frac{VD_h}{\nu} = \frac{(2.67 \text{ m/s})(0.00492 \text{ m})}{1.702\times10^{-5} \text{ m}^2/\text{s}} = 771.8 < 2300$$

Therefore the flow is laminar. Assuming fully developed flow, the Nusselt number for the air flow in this rectangular cross-section corresponding to the aspect ratio of $a/b = height/width = 15/0.25 = 60 \approx \infty$ is determined from Table 15-3 to be $Nu = 8.24$. Then,

$$h = \frac{k}{D_h}Nu = \frac{0.02662 \text{ W/m.°C}}{0.00492 \text{ m}}(8.24) = 44.58 \text{ W/m}^2.°\text{C}$$

The surface temperature of the hollow core near the exit is determined to be

$$T_{s,\max} = T_{out} + \frac{\dot{q}}{h} = 69.7°\text{C} + \frac{(45 \text{ W})/(0.06 \text{ m}^2)}{(44.58 \text{ W/m}^2.°\text{C})} = \mathbf{86.5°C}$$

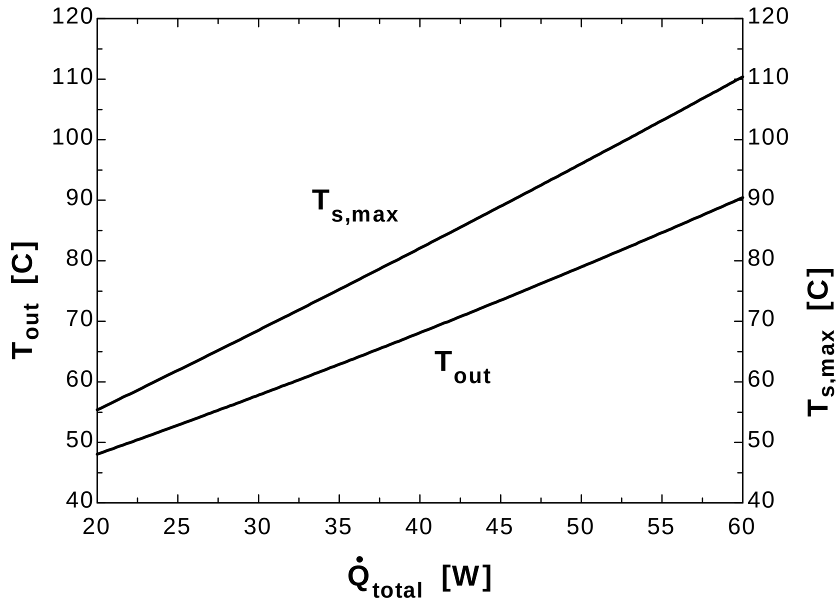
15-106 EES Prob. 15-104 is reconsidered. The effects of the power rating of the PCB and the volume flow rate of the air on the exit temperature of the air and the maximum temperature on the inner surface of the core are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

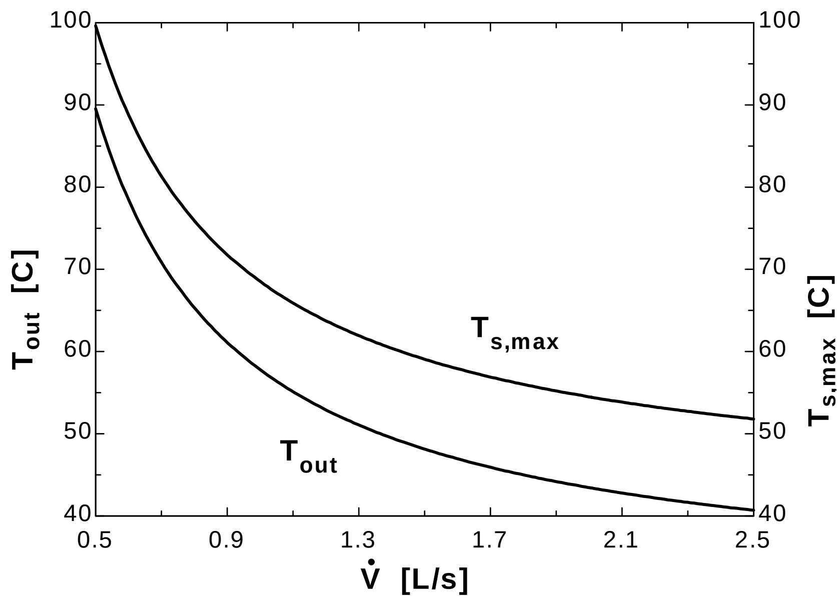
```
"GIVEN"
height=(15/100) [m]
length=(20/100) [m]
width=(0.25/100) [m]
Q_dot_total=30 [W]
T_in=30 [C]
V_dot=1 [L/s]

"PROPERTIES"
Fluid$='air'
rho=Density(Fluid$, T=T_ave, P=101.3)
c_p=CP(Fluid$, T=T_ave)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(Fluid$, T=T_ave)
Pr=Prandtl(Fluid$, T=T_ave)
mu=Viscosity(Fluid$, T=T_ave)
nu=mu/rho
T_ave=1/2*((T_in+T_out)/2+T_s_max)

"ANALYSIS"
"(a)"
A_c=height*width
p=2*(height+width)
D_h=(4*A_c/p)
Vel=(V_dot*Convert(L/s, m^3/s))/A_c
m_dot=rho*V_dot*Convert(L/s, m^3/s)
Q_dot_total=m_dot*c_p*(T_out-T_in)
"(b)"
Re=(Vel*D_h)/nu
"Re is calculated to be smaller than 2300. Therefore, the flow is laminar. From Table 15-3 of the
text"
Nusselt=8.24
h=k/D_h*Nusselt
A=2*height*length
Q_dot_total=h*A*(T_s_max-T_out)
```

Q_{total} [W]	T_{out} [C]	$T_{s, max}$ [C]
20	48.03	55.36
22	49.94	57.96
24	51.87	60.58
26	53.83	63.22
28	55.8	65.87
30	57.8	68.53
32	59.82	71.21
34	61.86	73.9
36	63.92	76.61
38	66	79.34
40	68.11	82.08
42	70.24	84.83
44	72.39	87.61
46	74.57	90.4
48	76.76	93.2
50	78.98	96.03
52	81.23	98.87
54	83.5	101.7
56	85.79	104.6
58	88.11	107.5
60	90.45	110.4

V [L/s]	T_{out} [C]	$T_{s, max}$ [C]
0.5	89.52	99.63
0.6	78.46	88.78
0.7	70.87	81.34
0.8	65.33	75.91
0.9	61.12	71.78
1	57.8	68.53
1.1	55.12	65.91
1.2	52.91	63.75
1.3	51.06	61.94
1.4	49.49	60.4
1.5	48.13	59.07
1.6	46.96	57.92
1.7	45.92	56.91
1.8	45	56.01
1.9	44.19	55.21
2	43.46	54.49
2.1	42.8	53.85
2.2	42.2	53.26
2.3	41.65	52.73
2.4	41.15	52.24
2.5	40.7	51.8

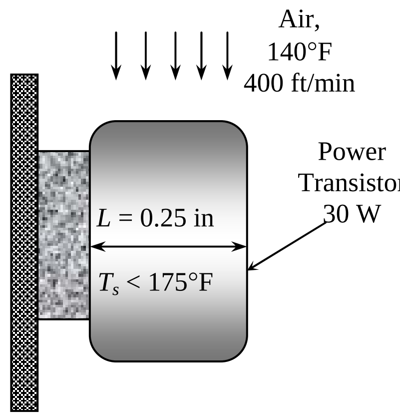


T_s,max
T_out
T_out [C]
T_s,max [C]
Q̇_total [W]



T_s,max
T_out
T_out [C]
T_s,max [C]
V̇ [L/s]

15-107E A transistor mounted on a circuit board is cooled by air flowing over it. The power dissipated when its case temperature is 175°F is to be determined.



Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas. **3** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm pressure and the film temperature of $T_f = (T_s+T_{fluid})/2 = (175+140)/2 = 157.5$°F are (Table A-15E)

$$k = 0.0166 \text{ Btu/h.ft.°F}$$
$$\nu = 0.214\times10^{-3} \text{ ft}^2/\text{s}$$
$$\text{Pr} = 0.718$$

Analysis The transistor is cooled by forced convection through its cylindrical surface as well as its flat top surface. The characteristic length for flow over a cylinder is the diameter D=0.2 in. Then,

$$\text{Re} = \frac{VD}{\nu} = \frac{(400/60 \text{ ft/s})(0.2/12 \text{ ft})}{0.214\times10^{-3} \text{ ft}^2/\text{s}} = 519$$

which falls into the range of 40-4000. Using the appropriate relation from Table 15-2, the Nusselt number and the convection heat transfer coefficient are determined to be

$$Nu = 0.683\,\text{Re}^{0.466}\,\text{Pr}^{1/3} = (0.683)(519)^{0.466}(0.718)^{1/3} = 11.3$$

$$h = \frac{k}{D}Nu = \frac{0.0166 \text{ Btu/h.ft.°F}}{(0.2/12 \text{ ft})}(11.3) = 11.2 \text{ Btu/h.ft}^2\text{.°F}$$

The transistor loses heat through its cylindrical surface as well as its circular top surface. For convenience, we take the heat transfer coefficient at the top surfaces to be the same as that of the side surface. (The alternative is to treat the top surface as a flat plate, but this will double the amount of calculations without providing much improvement in accuracy since the area of the top surface is much smaller and it is circular in shape rather than being rectangular). Then,

$$A_{cyl} = \pi DL + \pi D^2/4 = \pi(0.2/12 \text{ ft})(0.25/12 \text{ ft}) + \pi(0.2/12 \text{ ft})^2/4 = 0.00131 \text{ ft}^2$$

$$\dot{Q}_{cyl} = hA_{cyl}(T_s - T_{fluid}) = (11.2 \text{ Btu/h.ft}^2\text{.°F})(0.00131 \text{ ft}^2)(175-140)\text{°F} = 0.514 \text{ Btu/h} = \mathbf{0.15\ W}$$

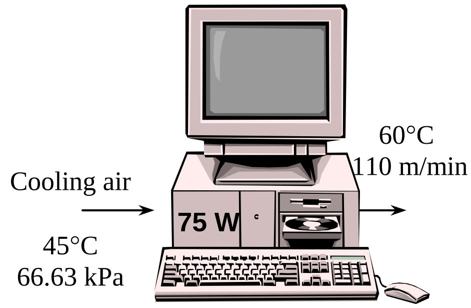
since 1 W = 3.4121 Btu/h. Therefore, the transistor can dissipate 0.15 W safely.

15-108 A desktop computer is to be cooled by a fan safely in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas.

Properties The specific heat of air at the average temperature of T_{avg} = (45+60)/2 = 52.5°C is c_p = 1007 J/kg.°C (Table A-15)

Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C. Then the required mass flow rate of air to absorb heat generated is determined to be



$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \rightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{75\text{ W}}{(1007\text{ J/kg.°C})(60-45)\text{°C}} = 0.00497\text{ kg/s} = 0.298\text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63\text{ kPa}}{(0.287\text{ kPa.m}^3\text{/kg.K})(60+273)\text{K}} = 0.6972\text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.298\text{ kg/min}}{0.6972\text{ kg/m}^3} = \mathbf{0.427\text{ m}^3\text{/min}}$$

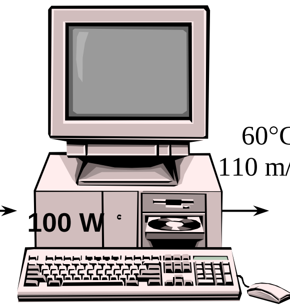
For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4}V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.427\text{ m}^3\text{/min})}{\pi(110\text{ m/min})}} = 0.070\text{ m} = \mathbf{7.0\text{ cm}}$$

15-109 A desktop computer is to be cooled by a fan safely in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas.

Properties The specific heat of air at the average temperature of T_{avg} = (45+60)/2 = 52.5°C is c_p = 1007 J/kg.°C (Table A-15)



Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C. Then the required mass flow rate of air to absorb heat generated is determined to be

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \rightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{100 \text{ W}}{(1007 \text{ J/kg.°C})(60 - 45)\text{°C}} = 0.00662 \text{ kg/s} = 0.397 \text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63 \text{ kPa}}{(0.287 \text{ kPa.m}^3\text{/kg.K})(60 + 273)\text{K}} = 0.6972 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.397 \text{ kg/min}}{0.6972 \text{ kg/m}^3} = \mathbf{0.570\ m^3/min}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

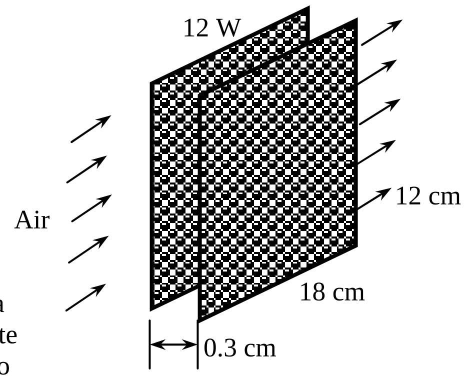
$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.570 \text{ m}^3\text{/min})}{\pi(110 \text{ m/min})}} = 0.081 \text{ m} = \mathbf{8.1\ cm}$$

15-110 A computer is cooled by a fan, and the temperature rise of air is limited to 15°C. The flow rate of air, the fraction of the temperature rise of air caused by the fan and its motor, and maximum allowable air inlet temperature are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas. **3** The local atmospheric pressure is 1 atm. **4** The entire heat generated in electronic components is removed by the air flowing through the opening between the PCBs. **5** The entire power consumed by the fan motor is transferred as heat to the cooling air.

Properties We use air properties at 1 atm and 30°C since air enters at room temperature, and the temperature rise is limited to 15°C (Table A-15)

$$\rho = 1.164 \text{ kg/m}^3$$
$$c_p = 1007 \text{ J/kg.°C}$$
$$\Pr = 0.728$$
$$k = 0.0259 \text{ W/m.°C}$$
$$\nu = 1.61\times10^{-5} \text{ m}^2/\text{s}$$



Analysis (*a*) Because of symmetry, we consider the flow area between the two adjacent PCBs only. We assume the flow rate of air through all 8 channels to be identical, and to be equal to one-eighth of the total flow rate. The total mass and volume flow rates of air through the computer are determined from

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{[(8\times12)+15]\text{ J/s}}{(1007\text{ J/kg.°C})(15\text{°C})} = 0.00735\text{ kg/s}$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.00735\text{ kg/s}}{1.164\text{ kg/m}^3} = \mathbf{0.00631\ m^3/s}$$

Noting that we have 8 PCBs and the flow area between the PCBs is 0.12 m and 0.003 m wide, the air velocity is determined to be

$$V = \frac{\dot{V}}{A_c} = \frac{(0.006819\text{ m}^3/\text{s})/8}{(0.12\text{ m})(0.003\text{ m})} = 2.37\text{ m/s}$$

(*b*) The temperature rise of air due to the 15 W of power consumed by the fan is

$$\Delta T_{air} = \frac{\dot{Q}_{fan}}{\dot{m}c_p} = \frac{15\text{ W}}{(0.00735\text{ kg/s})(1007\text{ J/kg.°C})} = 2.0\text{°C}$$

Then the fraction of temperature rise of air which is due to the heat generated by the fan becomes

$$f = \frac{2.0\text{°C}}{15\text{°C}}\times100 = \mathbf{13.5\%}$$

(*c*) To determine the surface temperature, we need to evaluate the convection heat transfer coefficient,

$$A_c = (height)(width) = (0.12\text{ m})(0.003\text{ m}) = 0.00036\text{ m}^2$$

$$D_h = \frac{4A_c}{p} = \frac{(4)(0.00036\text{ m}^2)}{(2)(0.12\text{ m} + 0.003\text{ m})} = 0.00585\text{ m}$$

$$\text{Re} = \frac{VD_h}{\nu} = \frac{(2.37\text{ m/s})(0.00585\text{ m})}{1.61\times10^{-5}\text{ m}^2/\text{s}} = 861 < 2300$$

Therefore, the flow is laminar. (Actually, the components will cause the flow to be turbulent. The laminar assumption gives conservative results). Assuming fully developed flow, the Nusselt number for the air flow through this rectangular channel corresponding to the aspect ratio $a/b = 12/0.3 = 40 \approx \infty$ is determined from Table 15-3 to be $Nu = 8.24$. Then the heat transfer coefficient becomes

$$h = \frac{k}{D_h} Nu = \frac{0.0259 \text{ W/m.°C}}{0.00585 \text{ m}} (8.24) = 36.5 \text{ W/m}^2.\text{°C}$$

Disregarding the entrance effects, the temperature difference between the surface of the PCB and the air anywhere along the channel is determined to be

$$T_s - T_{fluid} = \frac{\dot{Q}}{hA_s} = \frac{12 \text{ W}}{(36.5 \text{ W/m}^2.\text{°C})(0.12 \times 0.18 \text{ m}^2)} = 15.2\text{°C}$$

The highest air and component temperatures will occur at the exit. Therefore, in the limiting case, the component surface temperature at the exit will be 90°C. The air temperature at the exit in this case will be

$$T_{out,\max} = T_{s,\max} - \Delta T_{rise} = 90\text{°C} - 15.2\text{°C} = 74.8\text{°C}$$

Noting that the air experiences a temperature rise of 15°C between the inlet and the exit, the inlet temperature of air becomes

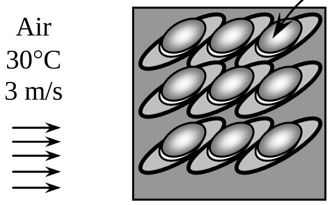
$$T_{in,\max} = T_{out,\max} - 15\text{°C} = 74.8\text{°C} - 15\text{°C} = \mathbf{59.8\text{°C}}$$

15-111 An array of power transistors is to be cooled by mounting them on a square aluminum plate and blowing air over the plate. The number of transistors that can be placed on this plate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas. **3** The local atmospheric pressure is 1 atm. **4** The entire heat generated by transistors is removed by the air flowing over the plate. **5** The heat transfer from the back side of the plate is negligible.

Properties The properties of air at the free stream temperature of 30°C are (Table A-15)

$$\rho = 1.164 \text{ kg/m}^3$$
$$c_p = 1007 \text{ J/kg.°C}$$
$$\text{Pr} = 0.728$$
$$k = 0.0259 \text{ W/m.°C}$$
$$\nu = 1.61\times10^{-5} \text{ m}^2/\text{s}$$



Analysis The plate area and the convection heat transfer coefficient are determined to be (from Table 15-2)

$$A_s = (0.2 \text{ m})(0.2 \text{ m}) = 0.04 \text{ m}^2$$

$$\text{Re} = \frac{VL}{\nu} = \frac{(3 \text{ m/s})(0.2 \text{ m})}{1.61\times10^{-5} \text{ m}^2/\text{s}} = 37{,}267$$

$$Nu = 0.664 \text{Re}^{1/2} \text{Pr}^{1/3} = (0.664)(37{,}267)^{1/2}(0.728)^{1/3} = 115.3$$

$$h = \frac{k}{L} Nu = \frac{0.0259 \text{ W/m.°C}}{0.2 \text{ m}}(115.3) = 14.9 \text{ W/m}^2.\text{°C}$$

The rate of heat transfer from the plate is

$$\dot{Q}_{conv} = hA_s(T_s - T_{fluid}) = (14.9 \text{ W/m}^2.\text{°C})(0.04 \text{ m}^2)(60-30)\text{°C} = 17.9 \text{ W}$$

Then the number of transistors that can be placed on this plate becomes

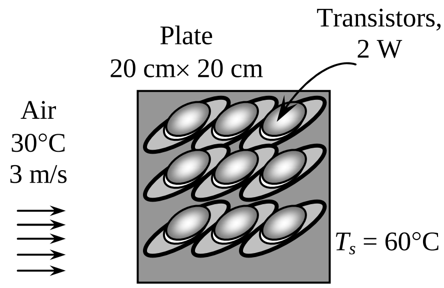
$$n = \frac{17.9 \text{ W}}{2 \text{ W}} = \mathbf{9} \text{ transistors}$$

15-112 An array of power transistors is to be cooled by mounting them on a square aluminum plate and blowing air over the plate. The number of transistors that can be placed on this plate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas. **3** The local atmospheric pressure is 83.4 kPa. **4** The entire heat generated by transistors is removed by the air flowing over the plate. **5** The heat transfer from the back side of the plate is negligible.

Properties At an elevation of 1610 m, the atmospheric pressure is 83.4 kPa or

$$P = (83.4\ \text{kPa})\frac{1\ \text{atm}}{101.325\ \text{kPa}} = 0.823\ \text{atm}$$



The properties of air at 30°C are (Table A-15)

$$\rho = 1.164\ \text{kg/m}^3$$
$$c_p = 1007\ \text{J/kg.°C}$$
$$\text{Pr} = 0.728$$
$$k = 0.0259\ \text{W/m.°C}$$
$$\nu = 1.61\times10^{-5}\ \text{m}^2/\text{s}/0.823 = 1.96\times10^{-5}\ \text{m}^2/\text{s}$$

Analysis The plate area and the convection heat transfer coefficient are determined to be (from Table 15-2)

$$A_s = (0.2\ \text{m})(0.2\ \text{m}) = 0.04\ \text{m}^2$$

$$\text{Re} = \frac{VL}{\nu} = \frac{(3\ \text{m/s})(0.2\ \text{m})}{1.96\times10^{-5}\ \text{m}^2/\text{s}} = 30{,}612$$

$$Nu = 0.664\,\text{Re}^{1/2}\,\text{Pr}^{1/3} = (0.664)(30{,}612)^{1/2}(0.728)^{1/3} = 104.5$$

$$h = \frac{k}{L}Nu = \frac{0.0259\ \text{W/m.°C}}{0.2\ \text{m}}(104.5) = 13.5\ \text{W/m}^2.\text{°C}$$

The rate of heat transfer from the plate is

$$\dot{Q}_{conv} = hA_s(T_s - T_{fluid}) = (13.5\ \text{W/m}^2.\text{°C})(0.04\ \text{m}^2)(60-30)\text{°C} = 16.2\ \text{W}$$

Then the number of transistors that can be placed on this plate becomes

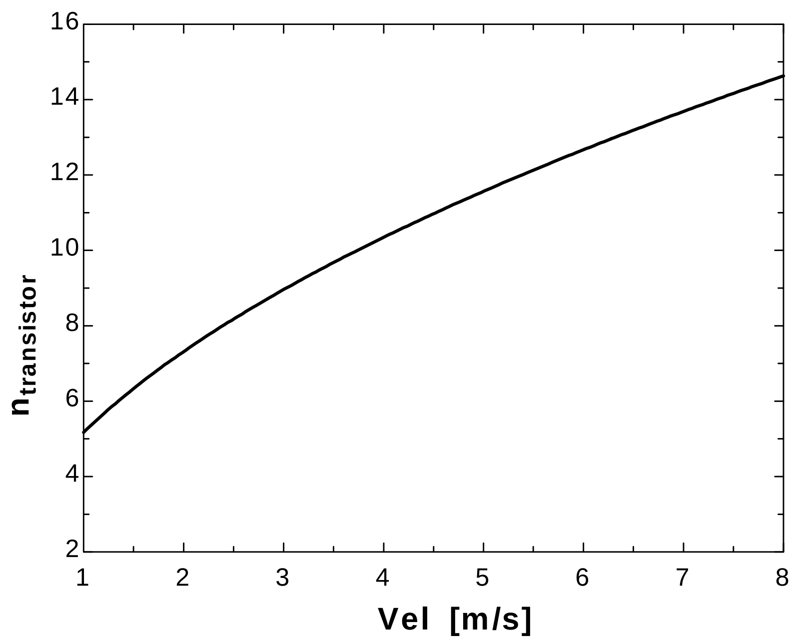
$$n = \frac{16.2\ \text{W}}{2\ \text{W}} = \mathbf{8}\ \text{transistors}$$

15-113 EES Prob. 15-111 is reconsidered. The effects effects of air velocity and the maximum plate temperature on the number of transistors are to be investigated.

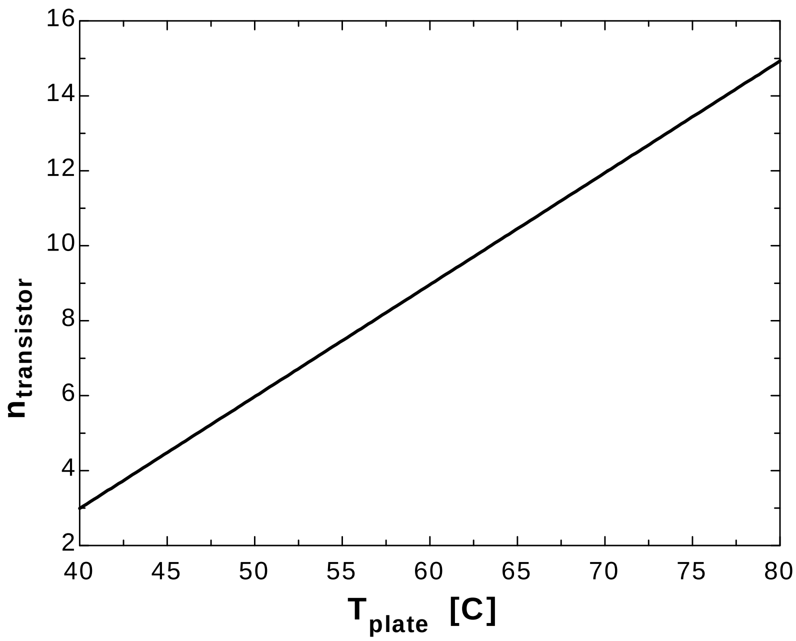
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
Q_dot=2 [W]
L=0.20 [m]
T_air=30 [C]
Vel=3 [m/s]
T_plate=60 [C]

"PROPERTIES"
Fluid$='air'
rho=Density(Fluid$, T=T_air, P=101.3)
k=Conductivity(Fluid$, T=T_air)
Pr=Prandtl(Fluid$, T=T_air)
mu=Viscosity(Fluid$, T=T_air)
nu=mu/rho

"ANALYSIS"
A=L^2
Re=(Vel*L)/nu
Nusselt=0.664*Re^0.5*Pr^(1/3)
h=k/L*Nusselt
Q_dot_conv=h*A*(T_plate-T_air)
n_transistor=Q_dot_conv/Q_dot
```

Vel [m/s]	$n_{transistor}$
1	5.173
1.5	6.335
2	7.315
2.5	8.179
3	8.96
3.5	9.677
4	10.35
4.5	10.97
5	11.57
5.5	12.13
6	12.67
6.5	13.19
7	13.69
7.5	14.17
8	14.63

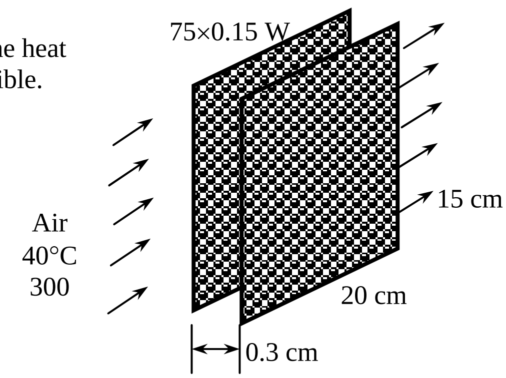


T_{plate} [C]	$n_{transistor}$
40	2.987
42.5	3.733
45	4.48
47.5	5.226
50	5.973
52.5	6.72
55	7.466
57.5	8.213
60	8.96
62.5	9.706
65	10.45
67.5	11.2
70	11.95
72.5	12.69
75	13.44
77.5	14.19
80	14.93



15-114 An enclosure containing an array of circuit boards is cooled by forced air flowing through the clearance between the tips of the components on the PCB and the back surface of the adjacent PCB. The exit temperature of the air and the highest surface temperature of the chips are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas. **3** The local atmospheric pressure is 1 atm. **4** The entire heat generated by the PCBs is removed by the air flowing through the clearance inside the enclosure. **5** The heat transfer from the back side of the circuit board is negligible.



Properties We use the properties of air at 1 atm and 40°C (Table A-15)

$$\rho = 1.127 \text{ kg/m}^3$$
$$c_p = 1007 \text{ J/kg.°C}$$
$$\text{Pr} = 0.726$$
$$k = 0.0266 \text{ W/m.°C}$$
$$\nu = 1.7\times10^{-5} \text{ m}^2/\text{s}$$

Analysis The volume and the mass flow rates of air are

$$\dot{Q} = \dot{m}c_p\Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p\Delta T} = \frac{3\text{ kJ/s}}{(4.18\text{ kJ/kg.°C})(4\text{°C})} = \mathbf{0.1794\ kg/s}$$

Then the exit temperature of air is determined from

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} = 40\text{°C} + \frac{(75\times0.15)\text{W}}{(0.00255\text{ kg/s})(1007\text{ J/kg.°C})} = \mathbf{44.4°C}$$

To determine the surface temperature, we need to calculate the convection heat transfer coefficient first,

$$A_s = (0.15\text{ m})(0.2\text{ m}) = 0.03\text{ m}^2$$

$$A_c = (0.15\text{ m})(0.003\text{m}) = 0.00045\text{ m}^2$$

$$D_h = \frac{4A_c}{p} = \frac{(4)(0.00045\text{ m}^2)}{(2)(0.15\text{ m} + 0.003\text{ m})} = 0.0059\text{ m}$$

$$\text{Re} = \frac{VD_h}{\nu} = \frac{(300/60\text{ m/s})(0.0059\text{ m})}{1.7\times10^{-5}\text{ m}^2/\text{s}} = 1735 < 2300$$

Therefore, the flow is laminar. Assuming fully developed flow, the Nusselt number for the air flow in this rectangular cross-section corresponding to the aspect ratio of $a/b = height/width = 15/0.3 = 50 \approx \infty$ is determined from Table 15-3 to be $Nu = 8.24$. Then,

$$h = \frac{k}{L}Nu = \frac{0.0266\text{ W/m.°C}}{0.0059\text{ m}}(8.24) = 37.1\text{ W/m}^2.\text{°C}$$

The highest surface temperature of the chips then becomes

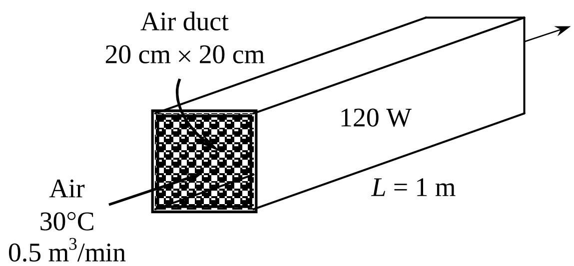
$$\dot{Q} = hA_s(T_{s,\max} - T_{fluid}) \longrightarrow T_{s,\max} = T_{air,out} + \frac{\dot{Q}}{hA_s} = 44.4\text{°C} + \frac{(75\times0.15)\text{ W}}{(37.1\text{ W/m}^2.\text{°C})(0.03\text{ m}^2)} = \mathbf{54.5°C}$$

15-115 The components of an electronic system located in a horizontal duct of rectangular cross-section are cooled by forced air flowing through the duct. The exit temperature of air and the highest component surface temperature in the duct are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas. **3** The local atmospheric pressure is 1 atm.

Properties We use the properties of air at 1 atm and 30°C (Table A-15)

$$\rho = 1.164 \text{ kg/m}^3$$
$$c_p = 1007 \text{ J/kg.°C}$$
$$\Pr = 0.728$$
$$k = 0.0259 \text{ W/m.°C}$$
$$\nu = 1.61\times10^{-5} \text{ m}^2/\text{s}$$



Analysis (*a*) The rate of heat transfer from the components to the forced air in the duct is

$$\dot{Q} = (0.80)(120 \text{ W}) = 96 \text{ W}$$

The mass flow rate of air is

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1.164 \text{ kg/m}^3)(0.5/60 \text{ m}^3\text{/s}) = 0.0097 \text{ kg/s}$$

Then the exit temperature of air is determined from

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} = 30\text{°C} + \frac{96 \text{ W}}{(0.0097 \text{ kg/s})(1007 \text{ J/kg.°C})} = \mathbf{39.8°C}$$

(*b*) The highest surface temperature can be determined from

$$\dot{Q}_{conv} = hA_s(T_s - T_{fluid})$$

But we first need to determine convection heat transfer coefficient,

$$A_s = (4)(1 \text{ m})(0.20 \text{ m}) = 0.8 \text{ m}^2$$

$$V = \frac{\dot{\boldsymbol{V}}}{A_c} = \frac{(0.5/60 \text{ m}^3\text{/s})}{(0.20 \text{ m})^2} = 0.208 \text{ m/s}$$

$$\text{Re} = \frac{VD_h}{\nu} = \frac{(0.208 \text{ m/s})(0.20 \text{ m})}{1.61\times10^{-5} \text{ m}^2\text{/s}} = 2588$$

From Table 15-2,

$$Nu = 0.102\,\text{Re}^{0.675}\,\Pr^{1/3} = (0.102)(2588)^{0.675}(0.728)^{1/3} = 18.5$$

$$h = \frac{k}{D_h}Nu = \frac{0.0259 \text{ W/m.°C}}{0.20 \text{ m}}(18.5) = 2.39 \text{ W/m}^2\text{.°C}$$

Then the highest component surface temperature in the duct becomes

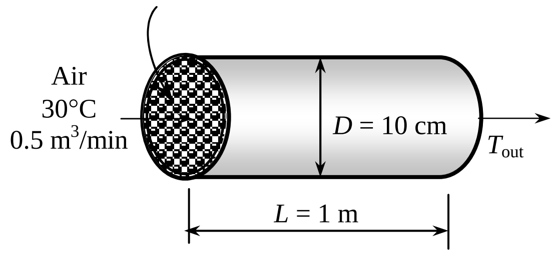
$$\dot{Q} = hA_s(T_{s,\max} - T_{air,out}) \longrightarrow T_{s,\max} = T_{air,out} + \frac{\dot{Q}}{hA_s} = 39.8\text{°C} + \frac{96 \text{ W}}{(2.39 \text{ W/m}^2\text{.°C})(0.8 \text{ m}^2)} = \mathbf{90.0°C}$$

15-116 The components of an electronic system located in a circular horizontal duct are cooled by forced air flowing through the duct. The exit temperature of air and the highest component surface temperature in the duct are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas. **3** The local atmospheric pressure is 1 atm.

Properties We use the properties of air at 1 atm and 30°C (Table A-15)

$$\rho = 1.164 \text{ kg/m}^3$$
$$c_p = 1007 \text{ J/kg.°C}$$
$$\text{Pr} = 0.728$$
$$k = 0.0259 \text{ W/m.°C}$$
$$\nu = 1.61\times10^{-5} \text{ m}^2/\text{s}$$



Analysis (*a*) The rate of heat transfer from the components to the forced air in the duct is

$$\dot{Q} = (0.80)(120 \text{ W}) = 96 \text{ W}$$

The mass flow rate of air is

$$\dot{m} = \rho\dot{V} = (1.164 \text{ kg/m}^3)(0.5/60 \text{ m}^3/\text{s}) = 0.0097 \text{ kg/s}$$

Then the exit temperature of air is determined from

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} = 30°\text{C} + \frac{96 \text{ W}}{(0.0097 \text{ kg/s})(1007 \text{ J/kg.°C})} = \mathbf{39.8°C}$$

(*b*) The highest surface temperature can be determined from

$$\dot{Q}_{conv} = hA_s(T_s - T_{fluid})$$

But we first need to determine convection heat transfer coefficient,

$$A_s = \pi DL = \pi(0.1 \text{ m})(1 \text{ m}) = 0.314 \text{ m}^2$$

$$V = \frac{\dot{V}}{A_c} = \frac{(0.5/60 \text{ m}^3/\text{s})}{\pi(0.10 \text{ m})^2/4} = 1.061 \text{ m/s}$$

$$\text{Re} = \frac{VD}{\nu} = \frac{(1.061 \text{ m/s})(0.10 \text{ m})}{1.61\times10^{-5} \text{ m}^2/\text{s}} = 6590$$

From Table 15-2,

$$Nu = 0.102\,\text{Re}^{0.675}\,\text{Pr}^{1/3} = (0.102)(6590)^{0.675}(0.728)^{1/3} = 34.7$$

$$h = \frac{k}{D_h}Nu = \frac{0.0259 \text{ W/m.°C}}{0.10 \text{ m}}(34.7) = 8.99 \text{ W/m}^2\text{.°C}$$

Then the highest component surface temperature in the duct becomes

$$\dot{Q} = hA_s(T_{s,\max} - T_{air,out}) \longrightarrow T_{s,\max} = T_{air,out} + \frac{\dot{Q}}{hA_s} = 39.8°\text{C} + \frac{96 \text{ W}}{(8.99 \text{ W/m}^2\text{.°C})(0.314 \text{ m}^2)} = \mathbf{73.8°C}$$

Liquid Cooling

15-117C When both are adequate, we would prefer forced air cooling in order to avoid the potential risks and problems associated with water cooling such as leakage, corrosion, extra weight, and condensation.

15-118C In direct cooling systems, the electronic components are in direct contact with the liquid, and thus the heat generated in the components is transferred directly to the liquid. In indirect cooling systems, however, there is no direct contact with the components. The heat generated in this case is first transferred to a medium such as a cold plate before it is removed by the liquid.

15-119C In closed loop cooling systems the liquid is recirculated while in the open loop systems the liquid is discarded after use. The heated liquid in closed loop systems is cooled in a heat exchanger, and it is recirculated through the system. In open loop systems, liquid (usually tap water) flows through the cooling system is discarded into a drain after it is heated.

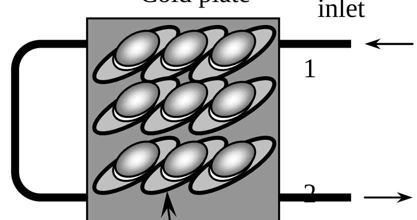
15-120C The properties of a liquid ideally suited for cooling electronic equipment include high thermal conductivity, high specific heat, low viscosity, high surface tension, high dielectric strength, chemical inertness, chemical stability, being non toxic, having low freezing and high boiling points, and low cost.

15-121 A cold plate is to be cooled by water. The mass flow rate of water, the diameter of the pipe, and the case temperature of the transistors are to be determined.

Assumptions **1** Steady operating conditions exist. **2** About 25 percent of the heat generated is dissipated from the components to the surroundings by convection and radiation.

Properties The properties of water at room temperature are ρ = 1000 kg/m^3 and c_p = 4180 J/kg.°C (Table A-9).

Analysis Noting that each of the 10 transistors dissipates 40 W of power and 75% of this power is removed by the water, the rate of heat transfer to the water is



$$\dot{Q} = (10\text{ transistors})(40\text{ W/transistor})(0.75) = 300\text{ W}$$

In order to limit the temperature rise of water to 4°C, the mass flow rate of water must be no less than

$$\dot{m} = \frac{\dot{Q}}{c_p \Delta T_{\text{rise}}} = \frac{300\text{ W}}{(4180\text{ J/kg.°C})(4°\text{C})} = 0.0179\text{ kg/s} = \mathbf{1.08\ kg/min}$$

The diameter of the pipe to maintain the velocity under 0.5 m/s is determined from

$$\dot{m} = \rho A_c V = \rho \frac{\pi D^2}{4} V$$

$$D = \sqrt{\frac{4\dot{m}}{\pi \rho V}} = \sqrt{\frac{4(0.0179\text{ kg/s})}{\pi(1000\text{ kg/m}^3)(0.5\text{ m/s})}} = 0.0068\text{ m} = \mathbf{0.68\ cm}$$

Noting that the case-to-liquid thermal resistance is 0.04 °C/W, the case temperature of the transistors is

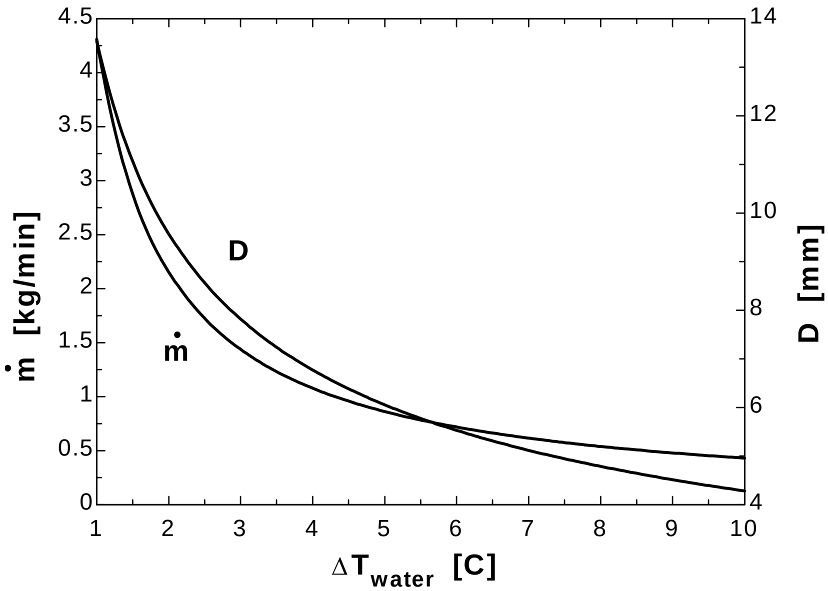
$$\dot{Q} = \frac{T_{case} - T_{liquid}}{R_{case-liquid}} \longrightarrow T_{case} = T_{liquid} + \dot{Q}R_{case-liquid} = 25°\text{C} + (300\text{ W})(0.04\text{ °C/W}) = \mathbf{37°C}$$

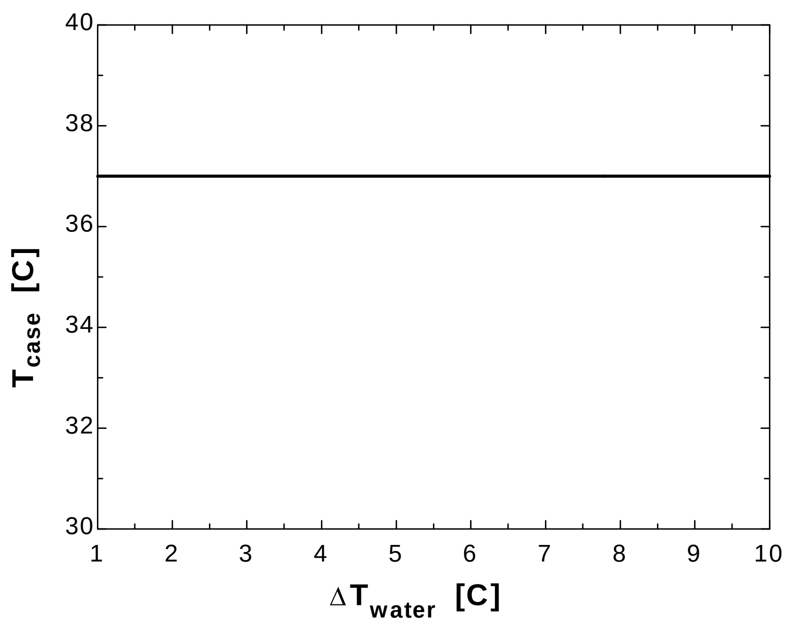
15-122 EES Prob. 15-121 is reconsidered. The effect of the maximum temperature rise of the water on the mass flow rate of water, the diameter of the pipe, and the case temperature is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

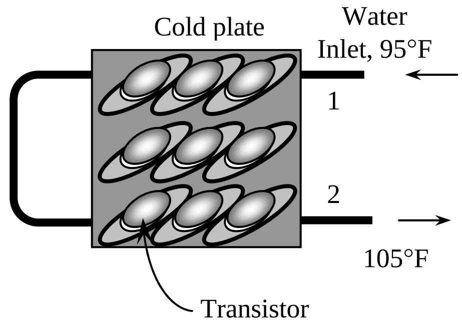
```
"GIVEN"
n_transistor=10
Q_dot=40 [W]
DELTAT_water=4 [C]
Vel=0.5 [m/s]
f_ConvRad=0.25
f_water=0.75
R_CaseLiquid=0.04 [C/W]
T_water=25 [C]

"PROPERTIES"
Fluid$='water'
rho=Density(Fluid$, T=T_water, P=101.3)
c_p=CP(Fluid$, T=T_water, P=101.3)*Convert(kJ/kg-C, J/kg-C)

"ANALYSIS"
Q_dot_total=n_transistor*Q_dot*f_water
m_dot=Q_dot_total/(c_p*DELTAT_water)*Convert(kg/s, kg/min)
m_dot*Convert(kg/min, kg/s)=rho*A*Vel
A=pi*(D*Convert(mm, m))^2/4
Q_dot_total=(T_case-T_water)/R_CaseLiquid
```

ΔT_{water} [C]	m [kg/min]	D [mm]	T_{case} [C]
1	4.31	13.54	37
1.5	2.873	11.05	37
2	2.155	9.574	37
2.5	1.724	8.563	37
3	1.437	7.817	37
3.5	1.231	7.237	37
4	1.077	6.77	37
4.5	0.9578	6.382	37
5	0.862	6.055	37
5.5	0.7836	5.773	37
6	0.7183	5.527	37
6.5	0.6631	5.31	37
7	0.6157	5.117	37
7.5	0.5747	4.944	37
8	0.5387	4.787	37
8.5	0.507	4.644	37
9	0.4789	4.513	37
9.5	0.4537	4.393	37
10	0.431	4.281	37


4.5
4
3.5
3
2.5
2
1.5
1
0.5
0
14
12
10
8
6
4
D
ṁ
ṁ [kg/min]
D [mm]
1
2
3
4
5
6
7
8
9
10
ΔT water [C]


40
38
36
34
32
30
T case [C]
1
2
3
4
5
6
7
8
9
10
ΔT water [C]

15-123E Electronic devices mounted on a cold plate is cooled by water. The amount of heat generated by the electronic devices is to be determined.



Assumptions **1** Steady operating conditions exist. **2** About 15 percent of the heat generated is dissipated from the components to the surroundings by convection and radiation.

Properties The properties of water at room temperature are ρ = 62.2 lbm/ft^3 and c_p = 0.998 Btu/lbm.°F.

Analysis The mass flow rate of water and the rate of heat removal by the water are

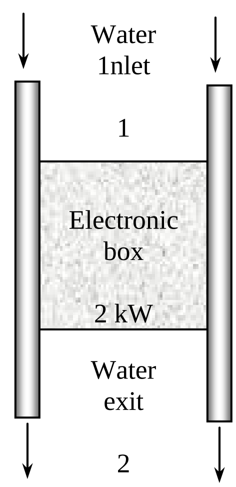
$$\dot{m} = \rho A_c V = \rho \frac{\pi D^2}{4} V = (62.2\,\text{lbm/ft}^3)\frac{\pi (0.25/12\,\text{ft})^2}{4}(60\,\text{ft/min}) = 1.272\,\text{lbm/min} = 76.33\,\text{lbm/h}$$

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = (76.33\,\text{lbm/h})(0.998\,\text{Btu/lbm.°F})(105 - 95)\text{°F} = 761.8\,\text{Btu/h}$$

which is 85 percent of the heat generated by the electronic devices. Then the total amount of heat generated by the electronic devices becomes

$$\dot{Q} = \frac{761.8\,\text{Btu/h}}{0.85} = \mathbf{896\,Btu/h} = \mathbf{263\,W}$$

15-124 A sealed electronic box is to be cooled by tap water flowing through channels on two of its sides. The mass flow rate of water and the amount of water used per year are to be determined.



Assumptions **1** Steady operating conditions exist. **2** Entire heat generated is dissipated by water.

Properties The specific heat of water at room temperature is c_p = 4180 J/kg.°C.

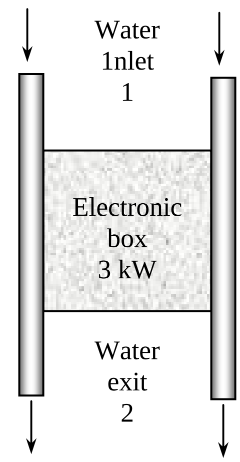
Analysis The mass flow rate of tap water flowing through the electronic box is

$$\dot{Q} = \dot{m}c_p\Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p\Delta T} = \frac{2\,\text{kJ/s}}{(4.18\,\text{kJ/kg.°C})(3\text{°C})} = \mathbf{0.1595\,kg/s}$$

Therefore, 0.1595 kg water is needed per second to cool this electronic box. Then the amount of cooling water used per year becomes

$$m = \dot{m}\Delta t = (0.1595\,\text{kg/s})(365\,\text{days/yr} \times 24\,\text{h/day} \times 3600\,\text{s/h})$$
$$= 5{,}030{,}000\,\text{kg/yr} = \mathbf{5030\,tons/yr}$$

15-125 A sealed electronic box is to be cooled by tap water flowing through channels on two of its sides. The mass flow rate of water and the amount of water used per year are to be determined.



Assumptions **1** Steady operating conditions exist. **2** Entire heat generated is dissipated by water.

Properties The specific heat of water at room temperature is c_p = 4180 J/kg.°C.

Analysis The mass flow rate of tap water flowing through the electronic box is

$$\dot{Q} = \dot{m}c_p \Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p \Delta T} = \frac{3\,\text{kJ/s}}{(4.18\,\text{kJ/kg.°C})(3°\text{C})} = \mathbf{0.2392\,kg/s}$$

Therefore, 0.2392 kg water is needed per second to cool this electronic box. Then the amount of cooling water used per year becomes

$$m = \dot{m}\Delta t = (0.2392\,\text{kg/s})(365\,\text{days/yr} \times 24\,\text{h/day} \times 3600\,\text{s/h})$$
$$= 7{,}544{,}500\,\text{kg/yr} = \mathbf{7545\,tons/yr}$$

Immersion Cooling

15-126C The desirable characteristics of a dielectric liquid used in immersion cooling of electronic devices are non-flammability, being chemically inert, compatibility with materials used in electronic equipment, and low boiling and freezing points.

15-127C An open loop immersion cooling system involves an external reservoir which supplies liquid continually to the electronic enclosure. The vapor generated inside is allowed to escape to the atmosphere. A pressure relief valve on the vapor vent line keeps the pressure and thus the temperature inside the enclosure at a preset value. In a closed loop immersion system, the vapor is condensed and returned to the electronic enclosure instead of being purged into the atmosphere.

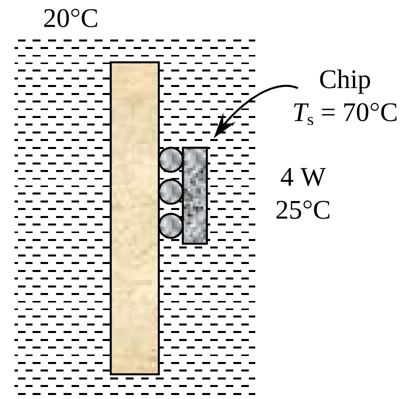
15-128C In external immersion cooling systems, the vapor is condensed outside the enclosure whereas in internal immersion cooling systems the vapor is condensed inside the enclosure by circulating a cooling fluid through the vapor. Therefore, in condenser is built into the enclosure in internal immersion cooling systems whereas it is placed outside in external immersion cooling systems.

15-129C The heat transfer coefficient is much greater in the boiling heat transfer than it is in the forced air or liquid cooling. Therefore, in the cooling of high-power electronic devices, boiling heat transfer is used to achieve high cooling rates with minimal temperature differences.

15-130 A logic chip is to be cooled by immersion in a dielectric fluid. The minimum heat transfer coefficient and the type of cooling mechanism are to be determined.

Assumptions Steady operating conditions exist.

Analysis The average heat transfer coefficient over the surface of the chip is determined from Newton's law of cooling to be



$$\dot{Q} = hA_s(T_{chip} - T_{fluid})$$

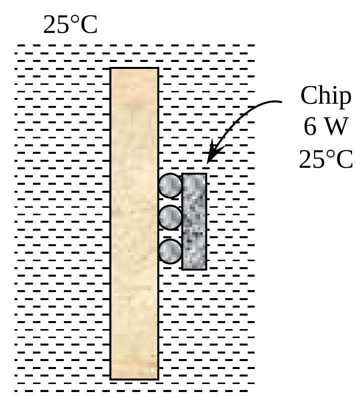
$$h = \frac{\dot{Q}}{A_s(T_{chip} - T_{fluid})}$$

$$= \frac{4\text{ W}}{(0.3\times10^{-4}\text{ m}^2)(70-20)°\text{C}} = \mathbf{2667\ W/m^2.°C}$$

which is rather high. An examination of Fig. 15-62 reveals that we can obtain such heat transfer coefficients with the boiling of fluorocarbon fluids. Therefore, a suitable cooling technique in this case is immersion cooling in such a fluid.

15-131 A chip is cooled by boiling in a dielectric fluid. The surface temperature of the chip is to be determined.

Assumptions The boiling curve in Fig. 15-63 is prepared for a chip having a surface area of 0.457 cm^2 being cooled in FC86 maintained at 5°C. The chart can be used for similar cases with reasonable accuracy.



Analysis The heat flux in this case is

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{6\text{ W}}{0.5\text{ cm}^2} = 12\text{ W/cm}^2$$

The temperature of the chip surface corresponding to this heat flux is determined from Fig. 15-63 to be

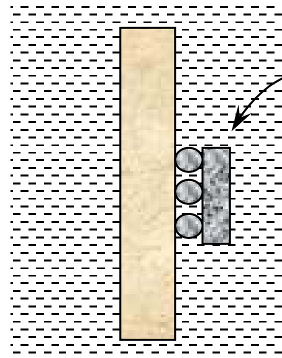
$$T_{chip} - T_{fluid} = 57°\text{C} \to T_{chip} = (T_{fluid} + 57)°\text{C} = (25+57)°\text{C} = \mathbf{82°C}$$

15-132 A logic chip is cooled by immersion in a dielectric fluid. The heat flux and the heat transfer coefficient on the surface of the chip and the thermal resistance between the surface of the chip and the cooling medium are to be determined.

Assumptions Steady operating conditions exist.

Analysis (*a*) The heat flux on the surface of the chip is

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{3.5\text{ W}}{0.8\text{ cm}^2} = \mathbf{4.375\ W/cm^2}$$



(*b*) The heat transfer coefficient on the surface of the chip is

$$\dot{Q} = hA_s(T_{chip} - T_{fluid})$$

$$h = \frac{\dot{Q}}{A_s(T_{chip} - T_{fluid})}$$

$$= \frac{3.5\text{ W}}{(0.8\times10^{-4}\text{ m}^2)(95-50)°\text{C}} = \mathbf{972\ W/m^2.°C}$$

(*c*) The thermal resistance between the surface of the chip and the cooling medium is

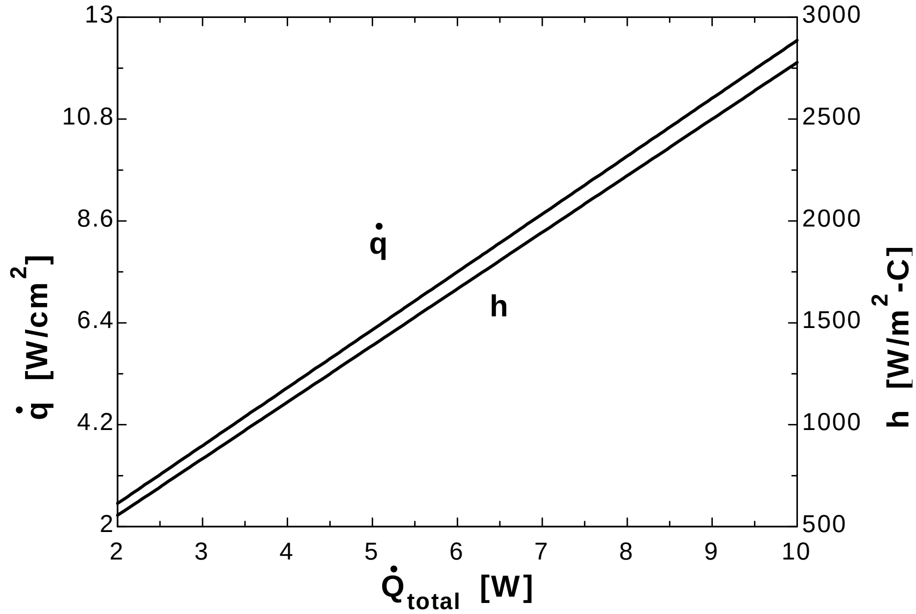
$$\dot{Q} = \frac{T_{chip} - T_{fluid}}{R_{chip-fluid}} \longrightarrow R_{chip-fluid} = \frac{T_{chip} - T_{fluid}}{\dot{Q}} = \frac{(95-50)°\text{C}}{3.5\text{ W}} = \mathbf{12.9°C/W}$$

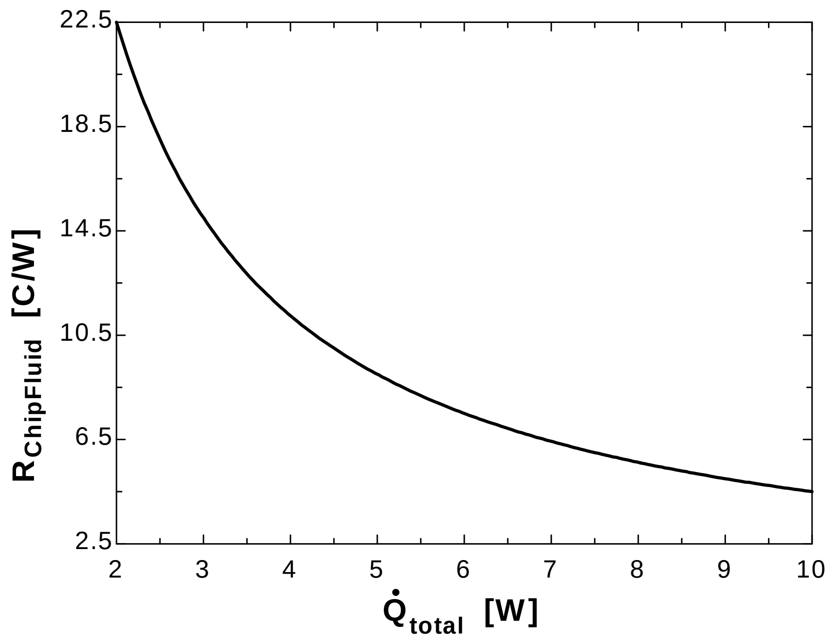
15-133 EES Prob. 15-132 is reconsidered. The effect of chip power on the heat flux, the heat transfer coefficient, and the convection resistance on chip surface is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
Q_dot_total=3.5 [W]
T_ambient=50 [C]
T_chip=95 [C]
A=0.8 [cm^2]

"ANALYSIS"
q_dot=Q_dot_total/A
Q_dot_total=h*A*Convert(cm^2, m^2)*(T_chip-T_ambient)
Q_dot_total=(T_chip-T_ambient)/R_ChipFluid
```

Q_{total} [W]	q [W/cm^2]	h [W/m^2-C]	$R_{ChipFluid}$ [C/W]
2	2.5	555.6	22.5
2.5	3.125	694.4	18
3	3.75	833.3	15
3.5	4.375	972.2	12.86
4	5	1111	11.25
4.5	5.625	1250	10
5	6.25	1389	9
5.5	6.875	1528	8.182
6	7.5	1667	7.5
6.5	8.125	1806	6.923
7	8.75	1944	6.429
7.5	9.375	2083	6
8	10	2222	5.625
8.5	10.63	2361	5.294
9	11.25	2500	5
9.5	11.88	2639	4.737
10	12.5	2778	4.5


13
10.8
8.6
6.4
4.2
2
3000
2500
2000
1500
1000
500
2
3
4
5
6
7
8
9
10
q̇
h
q̇ [W/cm^2]
h [W/m^2-C]
Q̇$_{total}$ [W]


22.5
18.5
14.5
10.5
6.5
2.5
2
3
4
5
6
7
8
9
10
R$_{ChipFluid}$ [C/W]
Q̇$_{total}$ [W]

15-134 A computer chip is to be cooled by immersion in a dielectric fluid. The minimum heat transfer coefficient and the appropriate type of cooling mechanism are to be determined.

Assumptions Steady operating conditions exist.

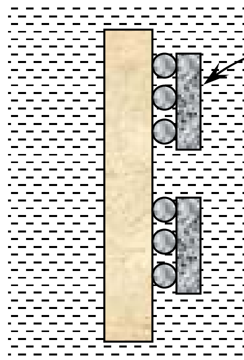
Analysis The average heat transfer coefficient over the surface of the chip is determined from Newton's law of cooling to be

$$\dot{Q} = hA_s(T_{chip} - T_{fluid})$$

$$h = \frac{\dot{Q}}{A_s(T_{chip} - T_{fluid})} = \frac{5\text{ W}}{(0.4\times10^{-4}\text{ m}^2)(55-10)^\circ\text{C}}$$

$$= \mathbf{2778\ W/m^2.^\circ C}$$

which is rather high. An examination of Fig. 15-62 reveals that we can obtain such heat transfer coefficients with the boiling of fluorocarbon fluids. Therefore, a suitable cooling technique in this case is immersion cooling in such a fluid.



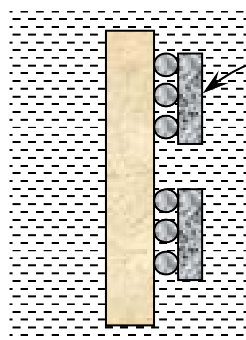
15-135 A chip is cooled by boiling in a dielectric fluid. The surface temperature of the chip is to be determined.

Assumptions The boiling curve in Fig. 15-63 is prepared for a chip having a surface area of 0.457 cm^2 being cooled in FC86 maintained at 5°C. The chart can be used for similar cases with reasonable accuracy.

Analysis The heat flux in this case is

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{3\text{ W}}{0.2\text{ cm}^2} = 15\text{ W/cm}^2$$

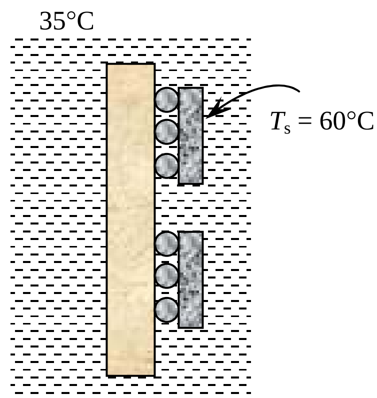
The temperature of the chip surface corresponding to this value is determined from Fig. 15-63 to be



$$T_{chip} - T_{fluid} = 63^\circ\text{C} \longrightarrow T_{chip} = (T_{fluid} + 63)^\circ\text{C} = (45+63)^\circ\text{C} = \mathbf{108^\circ C}$$

15-136 A chip is cooled by boiling in a dielectric fluid. The maximum power that the chip can dissipate safely is to be determined.

Assumptions The boiling curve in Fig. 15-63 is prepared for a chip having a surface area of 0.457 cm^2 being cooled in FC86 maintained at 5°C. The chart can be used for similar cases with reasonable accuracy.



Analysis The temperature difference between the chip surface and the liquid is

$$T_{chip} - T_{fluid} = (60 - 35)°\text{C} = 25°\text{C}$$

Using this value, the heat flux can be determined from Fig. 15-63 to be

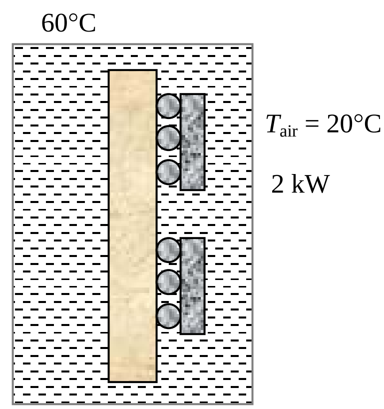
$$\dot{q} = 3.3\ \text{W/cm}^2$$

Then the maximum power that the chip can dissipate safely becomes

$$\dot{Q} = \dot{q}A_s = (3.3\ \text{W/cm}^2)(0.3\ \text{cm}^2) = \mathbf{0.99\ W}$$

15-137 An electronic device is to be cooled by immersion in a dielectric fluid. It is to be determined if the heat generated inside can be dissipated to the ambient air by natural convection and radiation as well as the heat transfer coefficient at the surface of the electronic device.

Assumptions Steady operating conditions exist.



Analysis Assuming the surfaces of the cubic enclosure to be at the temperature of the boiling dielectric fluid at 60 °C, the rate at which heat can be dissipated to the ambient air at 20 °C by combined natural convection and radiation is determined from

$$\dot{Q} = hA_s(T_s - T_{air}) = h(6a^2)(T_s - T_{air})$$
$$= (10\ \text{W/m}^2.°\text{C})[6(1\ \text{m})^2](60 - 20)°\text{C} = 2400\ \text{W} = \mathbf{2.4\ kW}$$

Therefore, the heat generated inside the cubic enclosure can be dissipated by natural convection and radiation. The heat transfer coefficient at the surface of the electronic device is

$$\dot{Q} = hA_s(T_s - T_{fluid}) \longrightarrow h = \frac{\dot{Q}}{A_s(T_s - T_{fluid})} = \frac{2000\ \text{W}}{(0.012\ \text{m}^2)(80 - 60)°\text{C}} = \mathbf{8333\ W/m^2.°C}$$

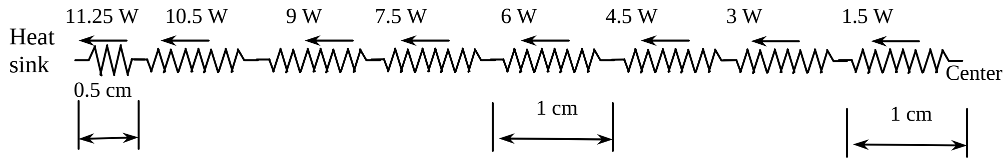
Review Problems

15-138C For most effective cooling, (1) the transistors must be mounted directly over the cooling lines, (2) the thermal contact resistance between the transistors and the cold plate must be minimized by attaching them tightly with a thermal grease, and (3) the thickness of the plates and the tubes should be as small as possible to minimize the thermal resistance between the transistors and the tubes.

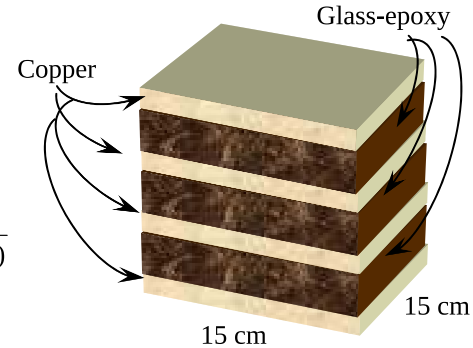
15-139C There is no such thing as heat rising. Only heated fluid rises because of lower density due to buoyancy. Heat conduction in a solid is due to the molecular vibrations and electron movement, and gravitational force has no effect on it. Therefore, the orientation of the bar is irrelevant.

15-140 A multilayer circuit board consisting of four layers of copper and three layers of glass-epoxy sandwiched together is considered. The magnitude and location of the maximum temperature that occurs in the PCB are to be determined.

Assumptions **1** Steady operating conditions exist **2** Thermal properties are constant. **3** There is no direct heat dissipation from the surface of the PCB, and thus all the heat generated is conducted by the PCB to the heat sink.



Analysis The effective thermal conductivity of the board is determined from



$$(k_1 t_1)_{copper} = 4[(386\text{ W/m.°C})(0.0001\text{ m})] = 0.1544\text{ W/°C}$$

$$(k_2 t_2)_{epoxy} = 3[(0.26\text{ W/m.°C})(0.0005\text{ m})] = 0.00039\text{ W/°C}$$

$$k_{eff} = \frac{(k_1 t_1)_{copper} + (k_2 t_2)_{epoxy}}{t_1 + t_2} = \frac{(0.1544 + 0.00039)\text{W/°C}}{4(0.0001\text{ m}) + 3(0.0005\text{ m})}$$
$$= 81.5\text{ W/m.°C}$$

The maximum temperature will occur in the middle of the board which is farthest away from the heat sink. We consider half of the board because of symmetry, and divide the region in 1-cm thick strips, starting at the mid-plane. Then from Fourier's law, the temperature difference across a strip can be determined from

$$\dot{Q} = k_{eff} A \frac{\Delta T}{L} \longrightarrow \Delta T = \frac{\dot{Q}L}{k_{eff} A}$$

where L = 1 cm = 0.01 m (except it is 0.5 cm for the strip attached to the heat sink), and the heat transfer area for all the strips is

$$A = (0.15\text{ m})[4(0.0001\text{ m}) + 3(0.0005\text{ m})] = 0.000285\text{ m}^2$$

Then the temperature at the center of the board is determined by adding the temperature differences across all the strips as

$$\Delta T_{\text{center-heat sink}} = \sum \frac{\dot{Q}L}{k_{eff} A} = \frac{(\dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 + \dot{Q}_4 + \dot{Q}_5 + \dot{Q}_6 + \dot{Q}_7 + \dot{Q}_8 / 2)L}{k_{eff} A}$$
$$= \frac{(1.5 + 3 + 4.5 + 6 + 7.5 + 9 + 10.5 + 11.25/2\text{ W})(0.01\text{ m})}{(81.5\text{ W/m.°C})(0.000285\text{ m}^2)} = 20.5\text{°C}$$

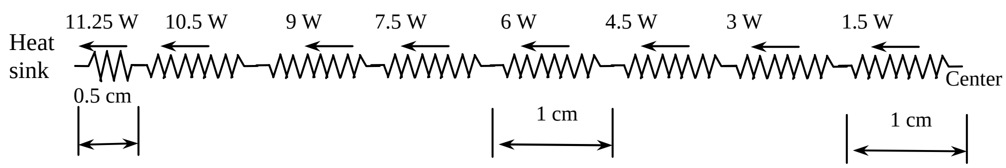
and $$T_{\text{center}} = T_{\text{heat sink}} + \Delta T_{\text{center-heat sink}} = 35\text{°C} + 20.5\text{°C} = \mathbf{55.5°C}$$

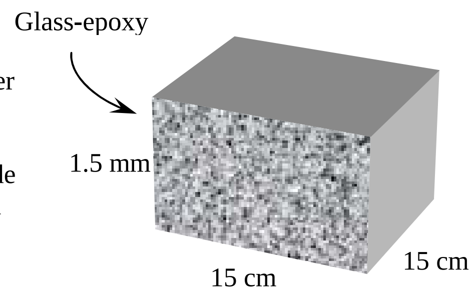
Discussion This problem can also be solved approximately by using the "average" heat transfer rate for the entire half board, and treating it as a constant. The heat transfer rate in each half changes from 0 at the center to 22.5/2 = 11.25 W at the heat sink, with an average of 11.25/2 = 5.625 W. Then the center temperature becomes

$$\dot{Q}_{avg} \cong k_{eff} A \frac{T_1 - T_2}{L} \longrightarrow T_{\text{center}} \cong T_{\text{heat sink}} + \frac{\dot{Q}_{ave} L}{k_{eff} A} = 35\text{°C} + \frac{(5.625\text{ W})(0.075\text{ m})}{(81.5\text{ W/m.°C})(0.000285\text{ m}^2)} = \mathbf{53.2°C}$$

15-141 A circuit board consisting of a single layer of glass-epoxy is considered. The magnitude and location of the maximum temperature that occurs in the PCB are to be determined.

Assumptions **1** Steady operating conditions exist **2** Thermal properties are constant. **3** There is no direct heat dissipation from the surface of the PCB, and thus all the heat generated is conducted by the PCB to the heat sink.





Analysis In this case the board consists of a 1.5-mm thick layer of epoxy. Again the maximum temperature will occur in the middle of the board which is farthest away from the heat sink. We consider half of the board because of symmetry, and divide the region in 1-cm thick strips, starting at the mid-plane. Then from Fourier's law, the temperature difference across a strip can be determined from

$$\dot{Q} = k_{eff} A \frac{\Delta T}{L} \longrightarrow \Delta T = \frac{\dot{Q}L}{k_{eff} A}$$

where L = 1 cm = 0.01 m (except it is 0.5 cm for the strip attached to the heat sink), and the heat transfer area for all the strips is

$$A = (0.15\,\text{m})(0.0015\,\text{m}) = 0.000225\,\text{m}^2$$

Then the temperature at the center of the board is determined by adding the temperature differences across all the strips as

$$\Delta T_{\text{center-heat sink}} = \sum \frac{\dot{Q}L}{k_{eff} A} = \frac{(\dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 + \dot{Q}_4 + \dot{Q}_5 + \dot{Q}_6 + \dot{Q}_7 + \dot{Q}_8 / 2)L}{k_{eff} A}$$

$$= \frac{(1.5 + 3 + 4.5 + 6 + 7.5 + 9 + 10.5 + 11.25/2\text{ W})(0.01\,\text{m})}{(0.26\text{ W/m.°C})(0.000225\,\text{m}^2)} = 8141°\text{C}$$

and $T_{\text{center}} = T_{\text{heat sink}} + \Delta T_{\text{center-heat sink}} = 35°\text{C} + 8141°\text{C} = \mathbf{8176°C}$

Discussion This problem can also be solved approximately by using the "average" heat transfer rate for the entire half board, and treating it as a constant. The heat transfer rate in each half changes from 0 at the center to 22.5/2 = 11.25 W at the heat sink, with an average of 11.25/2 = 5.625 W. Then the center temperature becomes

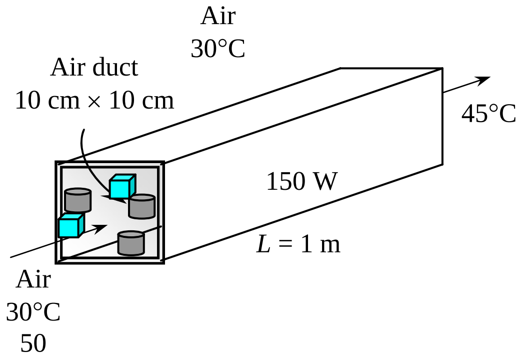
$$\dot{Q}_{avg} \cong k_{eff} A \frac{T_1 - T_2}{L} \longrightarrow T_{\text{center}} \cong T_{\text{heat sink}} + \frac{\dot{Q}_{avg} L}{k_{eff} A} = 35°\text{C} + \frac{(5.625\text{ W})(0.075\text{ m})}{(0.26\text{ W/m.°C})(0.000225\,\text{m}^2)} = \mathbf{7247°C}$$

15-142 The components of an electronic system located in a horizontal duct of rectangular cross-section are cooled by forced air flowing through the duct. The heat transfer from the outer surfaces of the duct by natural convection, the average temperature of the duct and the highest component surface temperature in the duct are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas. **3** The local atmospheric pressure is 1 atm.

Properties We use the properties of air at (30+45)/2 = 37.5°C are (Table A-15)

$$\rho = 1.136\ \text{kg/m}^3$$
$$c_p = 1007\ \text{J/kg.°C}$$
$$\text{Pr} = 0.726$$
$$k = 0.0264\ \text{W/m.°C}$$
$$\nu = 1.68\times10^{-5}\ \text{m}^2/\text{s}$$



Analysis (*a*) The volume and the mass flow rates of air are

$$\dot{V} = A_c V = (0.1\,\text{m})(0.1\,\text{m})(50/60\,\text{m/s}) = 0.008333\,\text{m}^3/\text{s}$$
$$\dot{m} = \rho\dot{V} = (1.136\,\text{kg/m}^3)(0.008333\,\text{m}^3/\text{s}) = 0.009466\,\text{kg/s}$$

The rate of heat transfer to the air flowing through the duct is

$$\dot{Q}_{forced\ conv} = \dot{m}c_p(T_{in} - T_{out}) = (0.009466\ \text{kg/s})(1007\ \text{J/kg.°C})(45 - 30)\text{°C} = 143.0\ \text{W}$$

Then the rate of heat loss from the outer surfaces of the duct to the ambient air by natural convection becomes

$$\dot{Q}_{conv} = \dot{Q}_{total} - \dot{Q}_{forced\ conv} = 150\ \text{W} - 143\ \text{W} = \mathbf{57\ W}$$

(*b*) The average surface temperature can be determined from

$$\dot{Q}_{conv} = hA_s(T_{s,duct} - T_{\substack{ambient\\air}})$$

But we first need to determine convection heat transfer coefficient. Using the Nusselt number relation from Table 15-2,

$$A_s = (4)(0.1\,\text{m})(1\,\text{m}) = 0.4\,\text{m}^2$$
$$\text{Re} = \frac{VD_h}{\nu} = \frac{(50/60\ \text{m/s})(0.1\,\text{m})}{1.68\times10^{-5}\ \text{m}^2/\text{s}} = 4960$$
$$Nu = 0.102\,\text{Re}^{0.675}\,\text{Pr}^{1/3} = (0.102)(4960)^{0.675}(0.726)^{1/3} = 28.6$$
$$h = \frac{k}{D_h}Nu = \frac{0.0264\ \text{W/m.°C}}{0.1\,\text{m}}(28.6) = 7.56\ \text{W/m}^2.\text{°C}$$

Then the average surface temperature of the duct becomes

$$\dot{Q}_{conv} = hA_s(T_s - T_{ambient}) \longrightarrow T_s = T_{ambient} + \frac{\dot{Q}_{conv}}{hA_s} = 30\text{°C} + \frac{57\ \text{W}}{(7.56\ \text{W/m}^2.\text{°C})(0.4\,\text{m}^2)} = \mathbf{48.9°C}$$

(*c*) The highest component surface temperature will occur at the exit of the duct. From Newton's law relation at the exit,

$$\dot{q}_{conv} = h(T_{s,\max} - T_{air,exit}) \longrightarrow T_{s,\max} = T_{air,exit} + \frac{\dot{Q}_{conv}/A_s}{h} = 45\text{°C} + \frac{57\ \text{W}}{(7.56\ \text{W/m}^2.\text{°C})(0.4\,\text{m}^2)} = \mathbf{63.8°C}$$

15-143 Two power transistors are cooled by mounting them on the two sides of an aluminum bracket that is attached to a liquid-cooled plate. The temperature of the transistor case and the fraction of heat dissipation to the ambient air by natural convection and to the cold plate by conduction are to be determined.

Assumptions **1** Steady operating conditions exist **2** Conduction heat transfer is one-dimensional. **3** We assume the ambient temperature is 25°C.

Analysis The rate of heat transfer by conduction is

$$\dot{Q}_{conduction} = (0.80)(12\text{ W}) = 9.6\text{ W}$$

Assuming heat conduction in the plate to be one-dimensional for simplicity, the thermal resistance of the aluminum plate and epoxy adhesive are

$$R_{alu\min um} = \frac{L}{kA} = \frac{0.02\text{ m}}{(237\text{ W/m.°C})(0.003\text{ m})(0.03\text{ m})} = 0.938\text{°C/W}$$

$$R_{epoxy} = \frac{L}{kA} = \frac{0.0002\text{ m}}{(1.8\text{ W/m.°C})(0.003\text{ m})(0.03\text{ m})} = 1.235\text{°C/W}$$

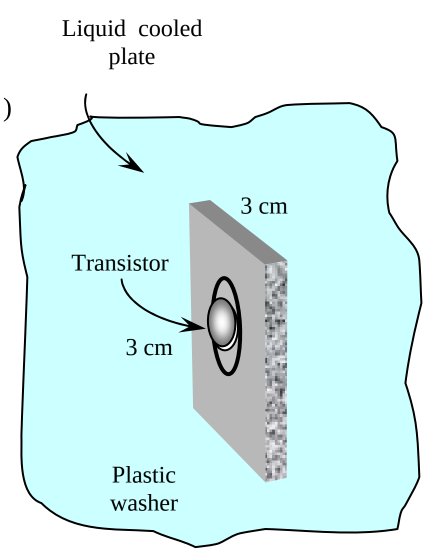
The total thermal resistance of the plate and the epoxy is

$$R_{plate+epoxy} = R_{epoxy} + R_{plate} = 1.235 + 0.938 = 2.173\text{°C/W}$$

Heat generated by the transistors is conducted to the plate, and then it is dissipated to the cold plate by conduction, and to the ambient air by convection. Denoting the plate temperature where the transistors are connected as $T_{s,max}$ and using the heat transfer coefficient relation from Table 15-1 for a vertical plate, the total heat transfer from the plate can be expressed as

$$\dot{Q}_{total} = \dot{Q}_{cond} + \dot{Q}_{conv} = \frac{\Delta T_{plate}}{R_{plate+epoxy}} + hA_{side}(T_{s,ave} - T_{air})$$

$$= \frac{T_{s,\max} - T_{cold\ plate}}{R_{plate+epoxy}} + 1.42\left(\frac{(T_{s,ave} - T_{air})}{L}\right)^{0.25} A_{side}(T_{s,\max} - T_{air})$$



where $T_{s,\,ave} = (T_{s,\,max} + T_{cold\ plate})/2$, L 0.03 m, and $A_{side} = 2(0.03\text{ m})(0.03\text{ m}) = 0.00018\text{ m}^2$. Substituting the known quantities gives

$$20\text{ W} = \frac{T_{s,\max} - 40}{2.173\text{°C/W}} + 1.42(0.00018)\frac{[(T_{s,\max} + 40)/2 - 25]^{1.25}}{(0.03)^{0.25}}$$

Solving for $T_{s,\,max}$ gives

$$T_{s,\,max} = 83.3\text{°C}$$

Then the rate of heat transfer by natural convection becomes

$$\dot{Q}_{conv} = 1.42(0.00018)\frac{[(83.3 + 40)/2 - 25]^{1.25}}{(0.03)^{0.25}} = 0.055\text{ W}$$

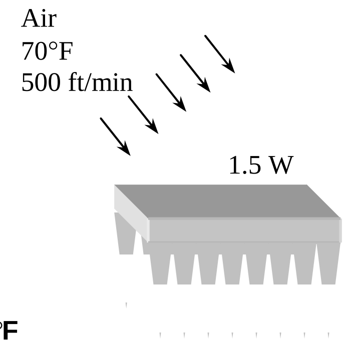
which is 0.055/20 = 0.00275 or **0.3%** of the total heat dissipated. The remaining **99.7%** of the heat is transferred by conduction. Therefore, heat transfer by natural convection is negligible. Then the surface temperature of the transistor case becomes

$$T_{case} = T_{s,\max} + \dot{Q}R_{\text{plastic washer}} = 83.3\text{°C} + (10\text{ W})(2\text{°C/W}) = \mathbf{103.3\text{°C}}$$

15-144E A plastic DIP with 24 leads is cooled by forced air. Using data supplied by the manufacturer, the junction temperature is to be determined for two cases.

Assumptions Steady operating conditions exist.

Analysis The junction-to-ambient thermal resistance of the device with 24 leads corresponding to an air velocity of 500 × 0.3048 = 152.4 m/min is determined from Fig. 15-23 to be



$$R_{junction-ambient} = 50\,°\text{C/W} = 50\times 1.8 = 90\,°\text{F/W}$$

Then the junction temperature becomes

$$\dot{Q} = \frac{T_{junction} - T_{ambient}}{R_{junction-ambient}} \longrightarrow T_{junction} = T_{ambient} + \dot{Q}R_{junction-ambient}$$

$$= 70°\text{F} + (1.5\text{ W})(90°\text{F/W}) = \mathbf{205°F}$$

When the fan fails the total thermal resistance is determined from Fig. 15-23 by reading the value at the intersection of the curve at the vertical axis to be

$$R_{junction-ambient} = 66\,°\text{C/W} = 66\times 1.8 = 118.8°\text{F/W}$$

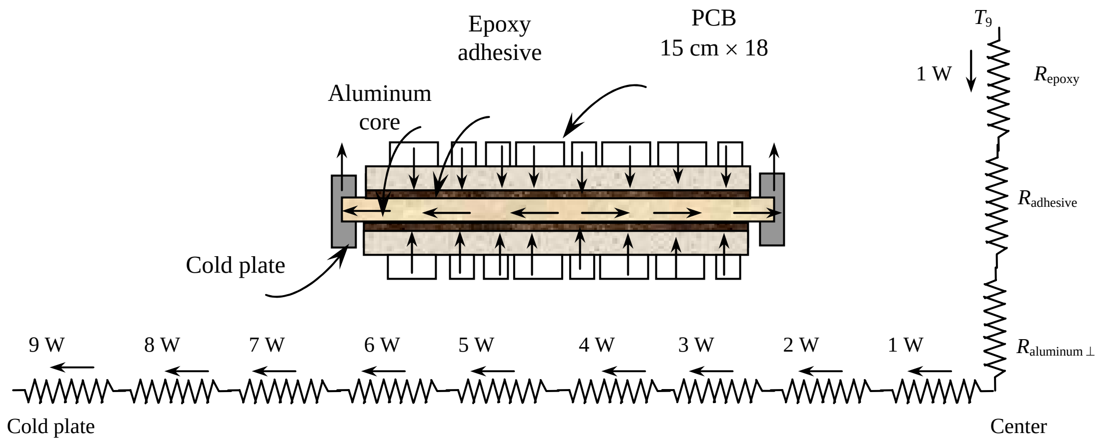
which yields

$$\dot{Q} = \frac{T_{junction} - T_{ambient}}{R_{junction-ambient}} \longrightarrow T_{junction} = T_{ambient} + \dot{Q}R_{junction-ambient}$$

$$= 70°\text{F} + (1.5\text{ W})(118.8\,°\text{F/W}) = \mathbf{248°F}$$

15-145 A circuit board is to be conduction-cooled by aluminum core plate sandwiched between two epoxy laminates. The maximum temperature rise between the center and the sides of the PCB is to be determined.

Assumptions **1** Steady operating conditions exist **2** Thermal properties are constant. **3** There is no direct heat dissipation from the surface of the PCB., and thus all the heat generated is conducted by the PCB to the heat sink.



Analysis Using the half thickness of the aluminum frame because of symmetry, the thermal resistances against heat flow in the vertical direction for a 1-cm wide strip are

$$R_{\text{aluminum},\perp} = \frac{L}{kA} = \frac{0.0006\text{ m}}{(237\text{ W/m.°C})(0.01\text{ m})(0.15\text{ m})} = 0.00169\text{ °C/W}$$

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.0005\text{ m}}{(0.26\text{ W/m.°C})(0.01\text{ m})(0.15\text{ m})} = 1.28205\text{ °C/W}$$

$$R_{\text{adhesive}} = \frac{L}{kA} = \frac{0.0001\text{ m}}{(1.8\text{ W/m.°C})(0.01\text{ m})(0.15\text{ m})} = 0.03703\text{ °C/W}$$

$$R_{\text{vertical}} = R_{\text{aluminum},\perp} + R_{\text{epoxy}} + R_{\text{adhesive}} = 0.00169 + 1.28205 + 0.03704 = 1.321\text{ °C/W}$$

We assume heat conduction along the epoxy and adhesive in the horizontal direction to be negligible, and heat to be conduction to the heat sink along the aluminum frame. The thermal resistance of the aluminum frame against heat conduction in the horizontal direction for a 1-cm long strip is

$$R_{\text{frame}} = R_{\text{aluminum},\parallel} = \frac{L}{kA} = \frac{0.01\text{ m}}{(237\text{ W/m.°C})(0.0012\text{ m})(0.18\text{ m})} = 0.1953\text{ °C/W}$$

The temperature difference across a strip is determined from

$$\Delta T = \dot{Q}R_{\text{aluminum},\parallel}$$

The maximum temperature rise across the 9 cm distance between the center and the sides of the board is determined by adding the temperature differences across all the strips as

$$\Delta T_{\text{horizontal}} = \sum \dot{Q}R_{\text{aluminum},\parallel} = (\dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 + \dot{Q}_4 + \dot{Q}_5 + \dot{Q}_6 + \dot{Q}_7 + \dot{Q}_8\, 2)R_{\text{aluminum},\parallel}$$
$$= (1+2+3+4+5+6+7+8+9\text{ W})(0.1953\text{ °C/W}) = 8.8\text{°C}$$

The temperature difference between the center of the aluminum core and the outer surface of the PCB is determined similarly to be

$$\Delta T_{\text{vertical}} = \sum \dot{Q}R_{\text{vertical},\perp} = (1\text{ W})(1.321\text{ °C/W}) = 1.3\text{°C}$$

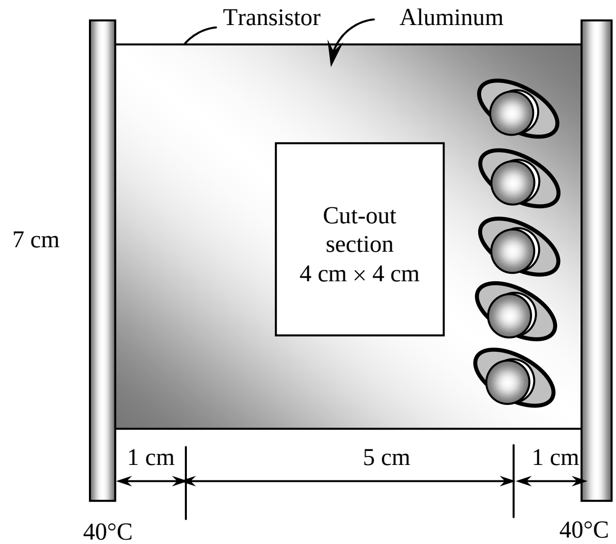
Then the maximum temperature rise across the 9-cm distance between the center and the sides of the PCB becomes

$$\Delta T_{\max} = \Delta T_{\text{horizontal}} + \Delta T_{\text{vertical}} = 8.8 + 1.3 = \mathbf{10.1°C}$$

15-146 Ten power transistors attached to an aluminum plate are cooled from two sides of the plate by liquid. The temperature rise between the transistors and the heat sink is to be determined.

Assumptions **1** Steady operating conditions exist **2** Thermal properties are constant.

Analysis We consider only half of the plate because of symmetry. It is stated that 70% of the heat generated is conducted through the aluminum plate, and this heat will be conducted across the 1-cm wide section between the transistors and the cooled edge of the plate. (Note that the mid section of the plate will essentially be isothermal and thus there will be no significant heat transfer towards the midsection). The rate of heat conduction to each side is of the plate is



$$\dot{Q}_{\text{cond,1-side}} = 0.70 \times (10\text{ W}) = 7\text{ W}$$

Then the temperature rise across the 1-cm wide section of the plate can be determined from

$$\dot{Q}_{\text{cond.1-side}} = kA\frac{(\Delta T)_{\text{plate}}}{L}$$

Solving for $(\Delta T)_{\text{plate}}$ and substituting gives

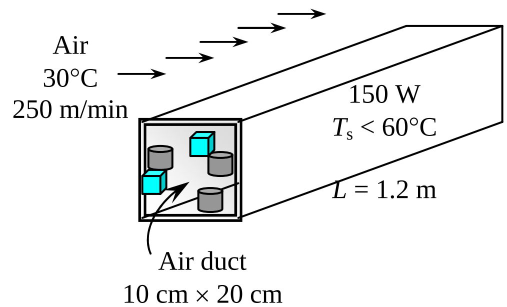
$$(\Delta T)_{\text{plate}} = \frac{\dot{Q}_{\text{cond.1-side}}L}{kA} = \frac{(7\text{ W})(0.01\text{ m})}{(237\text{ W/m.°C})(0.07\times 0.002\text{ m}^2)} = \mathbf{2.1°C}$$

15-147 The components of an electronic system located in a horizontal duct are cooled by air flowing over the duct. The total power rating of the electronic devices that can be mounted in the duct is to be determined for two cases.

Assumptions **1** Steady operating conditions exist **2** Thermal properties of air are constant. **3** The local atmospheric pressure is 1 atm.

Properties The properties of air at the film temperature of (30+60)/2 = 45°C are (Table A-15)

$$\text{Pr} = 0.724$$
$$k = 0.0270 \text{ W/m.°C}$$
$$\nu = 1.75\times10^{-5} \text{ m}^2/\text{s}$$



Analysis The surface area of the duct is

$$A_s = 2\{[(1.2\text{ m})(0.1\text{m})] + [(1.2\text{ m})(0.2\text{ m})]\} = 0.72\text{ m}^2$$

The duct is oriented such that air strikes the 10 cm high side normally. Using the Nusselt number relation from Table 15-2 for a 10-cm by 10-cm cross-section square as an approximation, the heat transfer coefficient is determined to be

$$\text{Re} = \frac{VD}{\nu} = \frac{(250/60\text{ m/s})(0.1\text{m})}{1.75\times10^{-5}\text{ m}^2/\text{s}} = 23{,}810$$

$$Nu = 0.102\,\text{Re}^{0.675}\,\text{Pr}^{1/3} = (0.102)(23{,}810)^{0.675}(0.724)^{1/3} = 82.4$$

$$h = \frac{k}{D_h}Nu = \frac{0.0270\text{ W/m.°C}}{0.1\text{ m}}(82.4) = 22.3\text{ W/m}^2.\text{°C}$$

Then the heat transfer rate (and thus the power rating of the components inside) in this orientation is determined from

$$\dot{Q} = hA_s(T_s - T_{fluid}) = (22.3\text{ W/m}^2.\text{°C})(0.72\text{ m}^2)(60-30)\text{°C} = \mathbf{481\ W}$$

We now consider the duct oriented such that air strikes the 20 cm high side normally. Using the Nusselt number relation from Table 15-2 for a 20-cm by 20-cm cross-section square as an approximation, the heat transfer coefficient is determined to be

$$\text{Re} = \frac{VD}{\nu} = \frac{(250/60\text{ m/s})(0.2\text{ m})}{1.75\times10^{-5}\text{ m}^2/\text{s}} = 47{,}619$$

$$Nu = 0.102\,\text{Re}^{0.675}\,\text{Pr}^{1/3} = (0.102)(47{,}619)^{0.675}(0.724)^{1/3} = 131.6$$

$$h = \frac{k}{D_h}Nu = \frac{0.0270\text{ W/m.°C}}{0.2\text{ m}}(131.6) = 17.8\text{ W/m}^2.\text{°C}$$

Then the heat transfer rate (and thus the power rating of the components inside) in this orientation is determined from

$$\dot{Q} = hA_s(T_s - T_{fluid}) = (17.8\text{ W/m}^2.\text{°C})(0.72\text{ m}^2)(60-30)\text{°C} = \mathbf{384\ W}$$

15-148 The components of an electronic system located in a horizontal duct are cooled by air flowing over the duct. The total power rating of the electronic devices that can be mounted in the duct is to be determined for two cases.

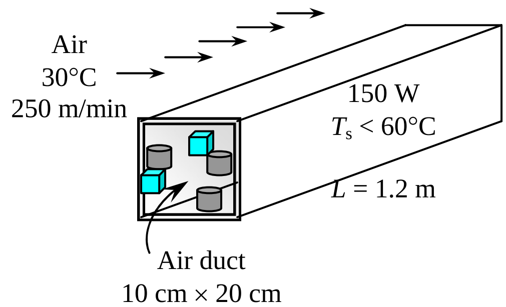
Assumptions **1** Steady operating conditions exist **2** Thermal properties of air are constant. **3** The local atmospheric pressure is 1 atm.

Properties The properties of air at the film temperature of (30+60)/2 = 45°C and 54.05 kPa are (Table A-15)

$$\mathrm{Pr} = 0.724$$

$$k = 0.0270 \text{ W/m.°C}$$

$$\nu = \frac{1.75\times10^{-5} \text{ m}^2/\text{s}}{54.05/101.325} = 3.28\times10^{-5} \text{ m}^2/\text{s}$$



Analysis The surface area of the duct is

$$A_s = 2\{[(1.2\text{ m})(0.1\text{ m})] + [(1.2\text{ m})(0.2\text{ m})]\} = 0.72\text{ m}^2$$

The duct is oriented such that air strikes the 10 cm high side normally. Using the Nusselt number relation from Table 15-2 for a 10-cm by 10-cm cross-section square as an approximation, the heat transfer coefficient is determined to be

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{(250/60\text{ m/s})(0.1\text{ m})}{3.28\times10^{-5}\text{ m}^2/\text{s}} = 12{,}703$$

$$Nu = 0.102\,\mathrm{Re}^{0.675}\,\mathrm{Pr}^{1/3} = (0.102)(12{,}703)^{0.675}(0.724)^{1/3} = 53.9$$

$$h = \frac{k}{D_h} Nu = \frac{0.0270\text{ W/m.°C}}{0.1\text{ m}}(53.9) = 14.6\text{ W/m}^2\text{.°C}$$

Then the heat transfer rate (and thus the power rating of the components inside) in this orientation is determined from

$$\dot{Q} = hA_s(T_s - T_{fluid}) = (14.6\text{ W/m}^2\text{.°C})(0.72\text{ m}^2)(60-30)\text{°C} = \mathbf{315\ W}$$

We now consider the duct oriented such that air strikes the 20 cm high side normally. Using the Nusselt number relation from Table 15-2 for a 20-cm by 20-cm cross-section square as an approximation, the heat transfer coefficient is determined to be

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{(250/60\text{ m/s})(0.2\text{ m})}{3.28\times10^{-5}\text{ m}^2/\text{s}} = 25{,}407$$

$$Nu = 0.102\,\mathrm{Re}^{0.675}\,\mathrm{Pr}^{1/3} = (0.102)(25{,}407)^{0.675}(0.724)^{1/3} = 86.1$$

$$h = \frac{k}{D_h} Nu = \frac{0.0270\text{ W/m.°C}}{0.2\text{ m}}(86.1) = 11.6\text{ W/m}^2\text{.°C}$$

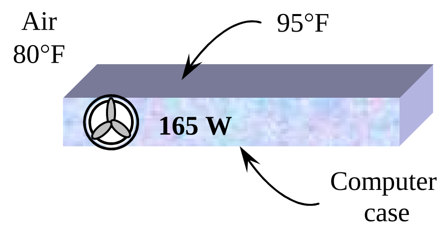
Then the heat transfer rate (and thus the power rating of the components inside) in this orientation is determined from

$$\dot{Q} = hA_s(T_s - T_{fluid}) = (11.6\text{ W/m}^2\text{.°C})(0.72\text{ m}^2)(60-30)\text{°C} = \mathbf{251\ W}$$

15-149E A computer is cooled by a fan blowing air into the computer enclosure. The fraction of heat lost from the outer surfaces of the computer case is to be determined.

Assumptions **1** Steady operating conditions exist **2** Thermal properties of air are constant. **3** The local atmospheric pressure is 1 atm.

Analysis Using the proper relation from Table 15-1, the heat transfer coefficient and the rate of natural convection heat transfer from the vertical side surfaces are determined to be



$$L = \frac{6}{12}\text{ ft}$$

$$A_{side} = (2)\left(\frac{20}{12}\text{ ft} + \frac{24}{12}\text{ ft}\right)\left(\frac{6}{12}\text{ ft}\right) = 3.67\text{ ft}^2$$

$$h_{conv,side} = 0.29\left(\frac{\Delta T}{L}\right)^{0.25} = 0.29\left(\frac{95-80}{6/12}\right)^{0.25} = 0.679\text{ Btu/h.ft}^2.°\text{F}$$

$$\dot{Q}_{conv,side} = h_{conv,side}A_{side}(T_s - T_{fluid}) = (0.679\text{ Btu/h.ft}^2.°\text{F})(3.67\text{ ft}^2)(95-80)°\text{F} = 37.4\text{ Btu/h}$$

Similarly, the rate of heat transfer from the horizontal top surface by natural convection is determined to be

$$L = \frac{4A_{top}}{p} = \frac{4\left(\frac{20}{12}\text{ ft}\right)\left(\frac{24}{12}\text{ ft}\right)}{2\left[\left(\frac{20}{12}\text{ ft}\right)+\left(\frac{24}{12}\text{ ft}\right)\right]} = 1.82\text{ ft}$$

$$A_{top} = \left(\frac{20}{12}\text{ ft}\right)\left(\frac{24}{12}\text{ ft}\right) = 3.33\text{ ft}^2$$

$$h_{conv,top} = 0.27\left(\frac{\Delta T}{L}\right)^{0.25} = 0.27\left(\frac{95-80}{1.82}\right)^{0.25} = 0.457\text{ Btu/h.ft}^2.°\text{F}$$

$$\dot{Q}_{conv,top} = h_{conv,top}A_{top}(T_s - T_{fluid}) = (0.457\text{ Btu/h.ft}^2.°\text{F})(3.33\text{ ft}^2)(95-80)°\text{F} = 22.8\text{ Btu/h}$$

The rate of heat transfer from the outer surfaces of the computer case by radiation is

$$\begin{aligned}\dot{Q}_{rad} &= \varepsilon A_s \sigma(T_s{}^4 - T_{surr}{}^4) \\ &= (0.85)(3.67\text{ ft}^2 + 3.33\text{ ft}^2)(0.1714\times10^{-8}\text{ Btu/h.ft}^2.\text{R}^4)[(95+460\text{ R})^4 - (80+460\text{ R})^4] \\ &= 100.4\text{ Btu/h}\end{aligned}$$

Then the total rate of heat transfer from the outer surfaces of the computer case becomes

$$\dot{Q}_{total} = \dot{Q}_{conv,side} + \dot{Q}_{conv,top} + \dot{Q}_{rad} = 37.4 + 22.8 + 100.4 = 160.6\text{ Btu/h}$$

Therefore, the fraction of the heat loss from the outer surfaces of the computer case is

$$f = \frac{(160.6/3.41214)\text{ W}}{165\text{ W}} = 0.285 = \mathbf{28.5\%}$$

15-150 . . . 15-152 Design and Essay Problems