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.

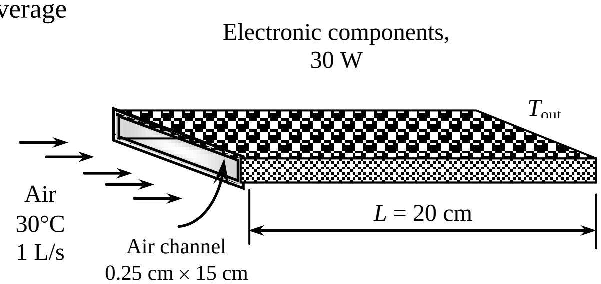
15-103C An undersized fan may cause the electronic system to overheat and fail. An oversized fan will definitely provide adequate cooling but it will needlessly be larger, noisier, more expensive, and will consume more power.

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

$$\rho = 1.127\ \text{kg/m}^3 \qquad C_p = 1007\ \text{J/kg.°C}$$
$$\text{Pr} = 0.7255 \qquad k = 0.02662\ \text{W/m.°C}$$
$$\nu = 1.702\times10^{-5}\ \text{m}^2/\text{s}$$



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

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

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

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

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

$$T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}C_p} = 30°\text{C} + \frac{30\ \text{W}}{(1.127\times10^{-3}\ \text{kg/s})(1007\ \text{J/kg.°C})} = \mathbf{56.4°C}$$

(*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,

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

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

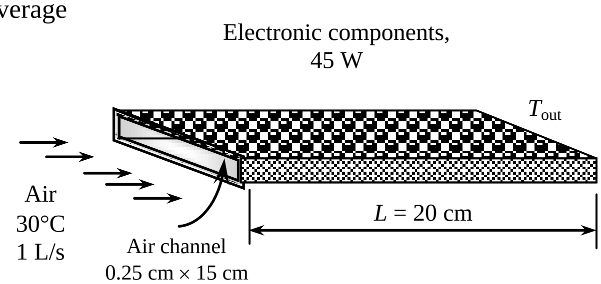
$$T_{s,\max} = T_{out} + \frac{\dot{q}}{h} = 56.4°\text{C} + \frac{(30\ \text{W})/(0.06\text{m}^2)}{(44.58\text{W/m}^2.°\text{C})} = \mathbf{67.6°C}$$

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

$$\rho = 1.127\ \text{kg/m}^3 \qquad C_p = 1007\ \text{J/kg.°C}$$
$$\Pr = 0.7255 \qquad k = 0.02662\ \text{W/m.°C}$$
$$\nu = 1.702\times10^{-5}\ \text{m}^2/\text{s}$$



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

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

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

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,

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

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 "!PROBLEM 15-106"

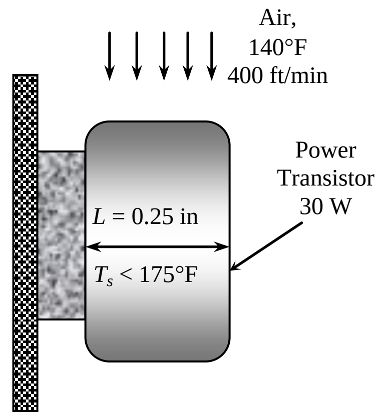
```
"GIVEN"
height=15/100 "[m]"
length=20/100 "[m]"
width=0.25/100 "[m]"
Q_dot_total=30 "[W], parameter to be varied"
T_in=30 "[C]"
"V_dot=1 [L/s], parameter to be varied"

"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

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)

$$k = 0.0166 \text{ Btu/h.ft.°F}$$
$$\nu = 0.214 \times 10^{-3} \text{ ft}^2/\text{s}$$
$$\Pr = 0.718$$

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,

$$\text{Re} = \frac{\mathbf{V}D}{\nu} = \frac{(400/60 \text{ ft/s})(0.2/12 \text{ ft})}{0.214 \times 10^{-3} \text{ ft}^2/\text{s}} = 519$$

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

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

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

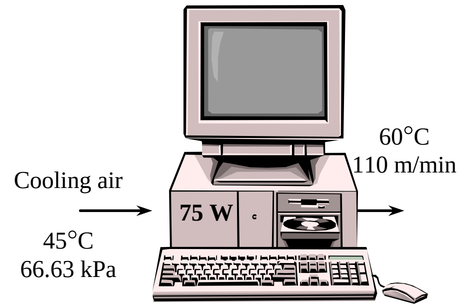
The transistor loses heat through its cylindrical surface as well as its circular top surface. For convenience, we take the heat transfer coefficient at the top surfaces to be the same as that of the side surface. (The alternative is to treat the top surface as a flat plate, but this will double the 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_{ave} = (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

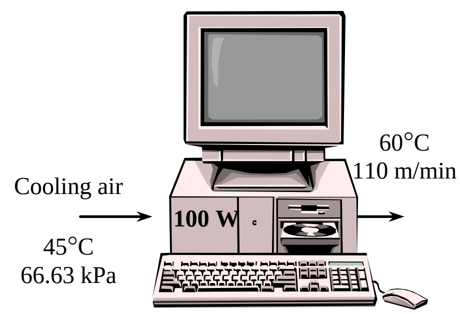
$$\rho = \frac{P}{RT} = \frac{66.63\,\text{kPa}}{(0.287\,\text{kPa.m}^3\text{/kg.K})(60 + 273)\text{K}} = 0.6972\,\text{kg/m}^3$$

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

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

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

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_{ave} = (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

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

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

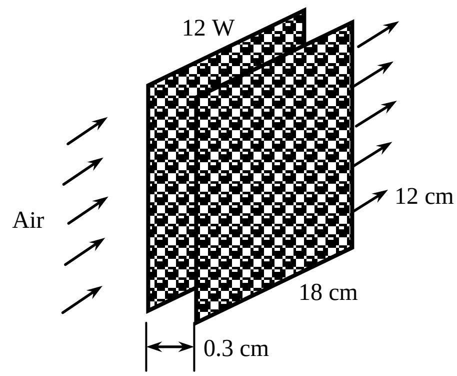
For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

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

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)



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

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

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

(*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

$$f = \frac{2.0°\text{C}}{15°\text{C}} \times 100 = \mathbf{13.5\%}$$

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

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

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

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

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

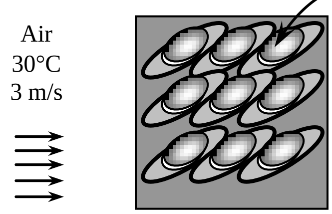
$$T_{in,\max} = T_{out,\max} - 15\text{°C} = 74.8\text{°C} - 15\text{°C} = \mathbf{59.8\text{°C}}$$

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)

$$\rho = 1.164 \text{ kg/m}^3$$
$$C_p = 1007 \text{J/kg.°C}$$
$$\text{Pr} = 0.728$$
$$k = 0.0259 \text{W/m.°C}$$
$$\nu = 1.61 \times 10^{-5} \text{ m}^2/\text{s}$$



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

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

$$\text{Re} = \frac{\mathbf{V}L}{\nu} = \frac{(3 \text{ m/s})(0.2 \text{ m})}{1.61 \times 10^{-5} \text{ m}^2/\text{s}} = 37{,}267$$

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

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

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.

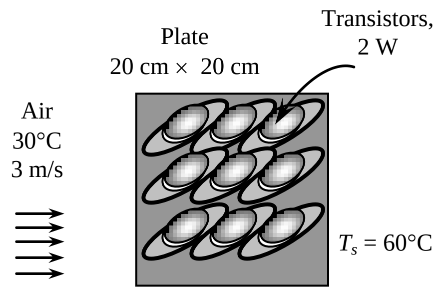
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas. **3** The local atmospheric pressure is 83.4 kPa. **4** The entire heat generated by transistors is removed by the air flowing over the plate. **5** The heat transfer from the back side of the plate is negligible.

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)

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

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

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

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

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 "!PROBLEM 15-113"

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

"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

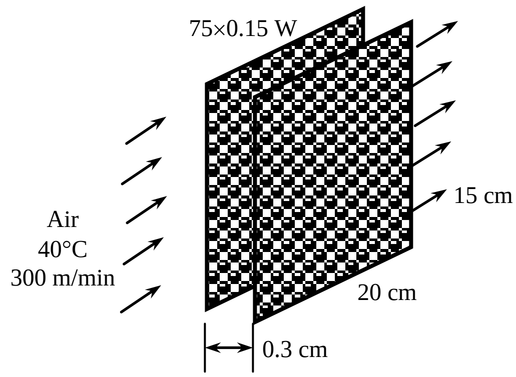
72.5	12.69
75	13.44
77.5	14.19
80	14.93

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

$$\dot{Q} = \dot{m}C_p\Delta T$$

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

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,

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

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

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

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

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

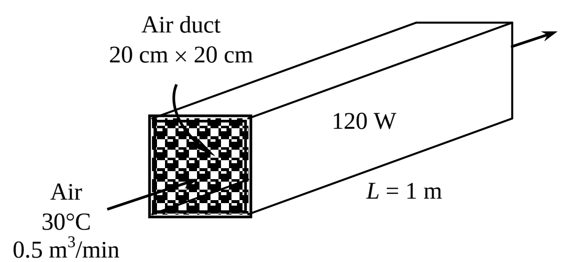
$$\dot{Q} = hA_s(T_{s,\max} - T_{fluid}) \longrightarrow T_{s,\max} = T_{air,out} + \frac{\dot{Q}}{hA_s} = 44.4°\text{C} + \frac{(75\times0.15)\ \text{W}}{(37.1\text{W/m}^2.\text{°C})(0.03\text{m}^2)} = \mathbf{54.5°C}$$

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)

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



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,

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

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

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

From Table 15-2,

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

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

Then the highest component surface temperature in the duct becomes

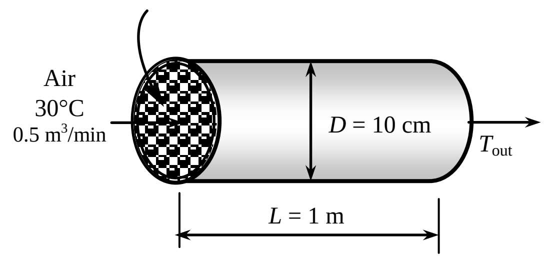
$$\dot{Q} = hA_s(T_{s,\max} - T_{air,out}) \longrightarrow T_{s,\max} = T_{air,out} + \frac{\dot{Q}}{hA_s} = 39.8°\text{C} + \frac{96\ \text{W}}{(2.39\ \text{W/m}^2.°\text{C})(0.8\ \text{m}^2)} = \mathbf{90.0°C}$$

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,

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

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

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

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.

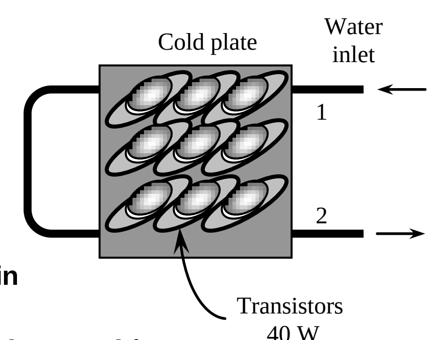
15-120C The properties of a liquid ideally suited for cooling electronic equipment include high thermal conductivity, high specific heat, low viscosity, high surface tension, high dielectric strength, chemical inertness, chemical stability, being non toxic, having low freezing and high boiling points, 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 $4\,°\text{C}$, 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

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

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

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

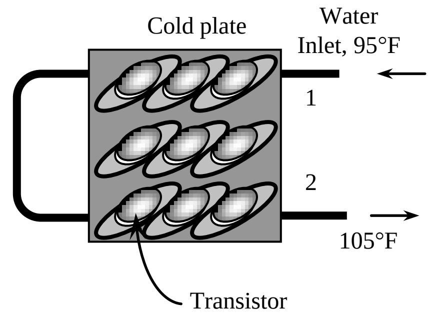
$$\dot{Q} = \frac{T_{case} - T_{liquid}}{R_{case-liquid}} \longrightarrow T_{case} = T_{liquid} + \dot{Q} R_{case-liquid} = 25°\text{C} + (300\,\text{W})(0.04°\text{C/W}) = \mathbf{37°C}$$

15-122 "!PROBLEM 15-122"

```
"GIVEN"
n_transistor=10
Q_dot=40 "[W]"
"DELTAT_water=4 [C], parameter to be varied"
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

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

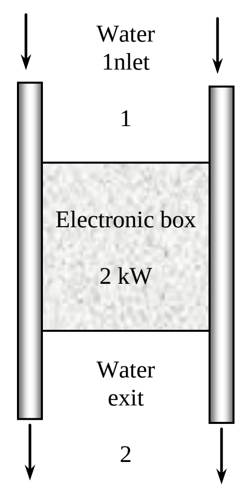
$$\dot{m} = \rho A_c \mathbf{V} = \rho \frac{\pi D^2}{4} \mathbf{V} = (62.2\,\text{lbm/ft}^3) \frac{\pi (0.25/12\,\text{ft})^2}{4} (60\,\text{ft/min}) = 1.272\,\text{lbm/min} = 76.33\,\text{lbm/h}$$

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

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.

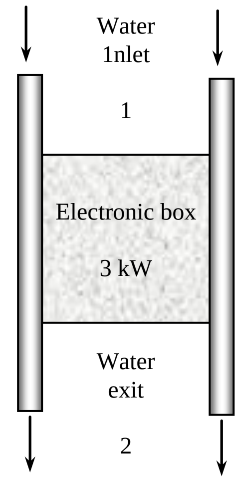
Analysis The mass flow rate of tap water flowing through the electronic box is

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

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

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

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.

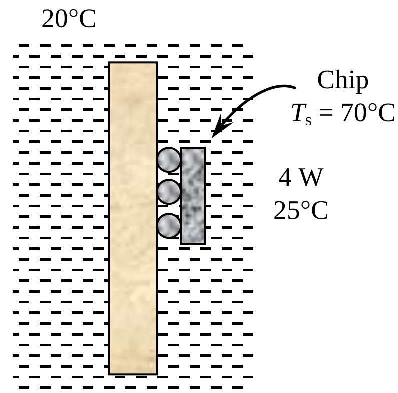
15-130 A logic chip is to be cooled by immersion in a dielectric fluid. The minimum heat transfer coefficient and the type of cooling mechanism are to be determined.

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

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

$$h = \frac{\dot{Q}}{A_s(T_{chip} - T_{fluid})} = \frac{4\text{ W}}{(0.3\times10^{-4}\text{ m}^2)(70-20)°\text{C}} = \mathbf{2667\ W/m^2.°C}$$



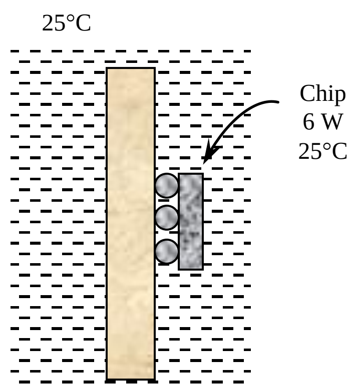
which is rather high. An examination of Fig. 15-62 reveals that we can obtain such heat transfer coefficients with the boiling of fluorocarbon fluids. Therefore, a suitable cooling technique in this case is immersion cooling in such a fluid.

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

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

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.

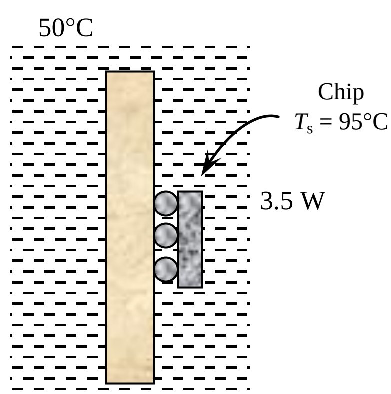
Analysis (*a*) The heat flux on the surface of the chip is

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

(*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 "!PROBLEM 15-133"

```
"GIVEN"
"Q_dot_total=3.5 [W], parameter to be varied"
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

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.

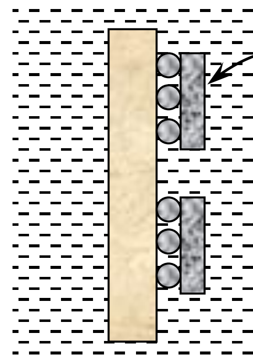
Analysis The average heat transfer coefficient over the surface of the chip is determined from Newton's law of cooling to be

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

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

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

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.

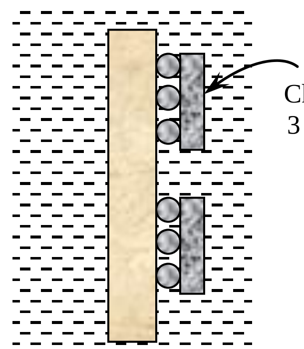


15-135 A chip is cooled by boiling in a dielectric fluid. The surface temperature of the chip is to be determined.

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

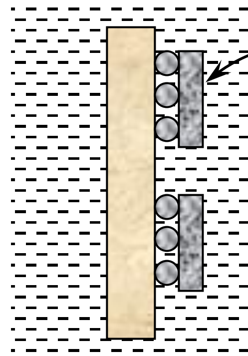
$$T_{chip} - T_{fluid} = 63^\circ\text{C} \longrightarrow T_{chip} = (T_{fluid} + 63)^\circ\text{C} = (45+63)^\circ\text{C} = \mathbf{108^\circ C}$$

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

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

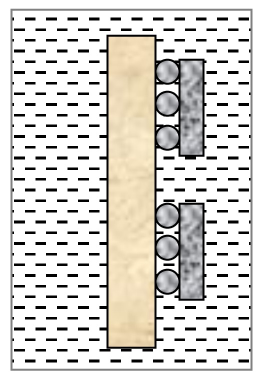
Then the maximum power that the chip can dissipate safely becomes

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

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 $60\,°\text{C}$, the rate at which heat can be dissipated to the ambient air at $20\,°\text{C}$ by combined natural convection and radiation is determined from



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

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.