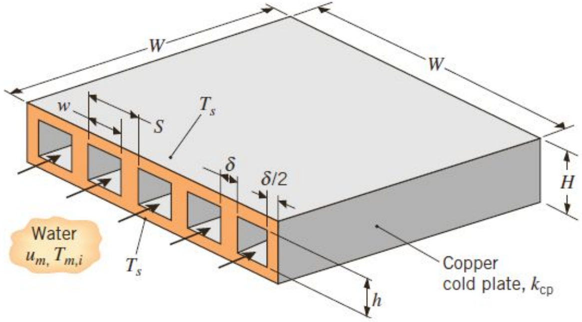
7.7 Water at 10°C flows over a 3 cm O.D. cylinder at 70°C. The velocity is 1 m/s. Evaluate $\overline{h}$.

7.27 Water at 27°C flows at 2.2 m/s in a 0.04 m I.D. thin-walled pipe. Air at 227°C flows across it at 7.6 m/s. Find the pipe wall temperature.

7.30 Water at 37°C flows at 3 m/s across at 6 cm O.D. tube that is held at 97°C. In a second configuration, 37°C water flows at an average velocity of 3 m/s through a bundle of 6 cm O.D. tubes that are held at 97°C. The bundle is staggered, with $S_T/S_L = 2$. Compare the heat transfer coefficients for the two situations.

Consider a cold plate of width $W = 100$ mm and height $H = 10$ mm, with 10 square channels of width $w = 6$ mm and a spacing of $\delta = 4$ mm between channels. Water enters the channels at a temperature of $T_{m,i} = 300$ K and a velocity of $u_m = 2$ m/s. If the top and bottom cold plate surfaces are at $T_s = 360$ K, what is the outlet water temperature and the total rate of heat transfer to the cold plate? The thermal conductivity of the copper is 400 W/m·K, while average properties of the water may be taken to be $\rho = 984$ kg/m^3, $c_p = 4184$ J/kg·K, $\mu = 489 \times 10^{-6}$ N·s/m^2, $k = 0.65$ W/m·K, and $Pr = 3.15$. Is this a good cold plate design? How could its performance be improved?



7.7 Water at 10°C flows over a 3 cm O.D. cylinder at 70°C. The velocity is 1 m/s. Evaluate $\overline{h}$.

DATA

```
T_mean=40; % °C

rho=992.2; % kg/m3
cp=4.18; % kJ/kgK
k=0.628; % W/mK
Pr=4.345; % -
mu=652.72e-6; % Pa*s
```

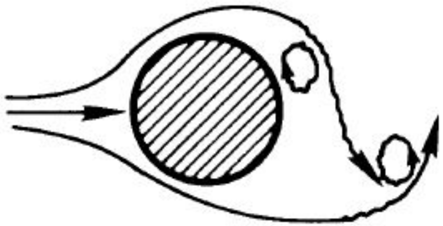
$$\overline{\mathrm{Nu}}_D = 0.3 + \frac{0.62\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1 + \left(\frac{\mathrm{Re}_D}{282{,}000}\right)^{5/8}\right]^{4/5} \qquad (7.65)$$

All properties in eqns. (7.65) to (7.68) are to be evaluated at a film temperature $T_f = (T_w + T_\infty)/2$.

Re = u*D/ni = 4.5603e+04

Nu = 257.89

h = Nu*k/D = 5398.5 W/m^2K



$150 \leqslant Re_D < 300$	Transition range to turbulence in vortex.
$300 \leqslant Re_D \lesssim 3 \times 10^5$	Vortex street is fully turbulent, and the flow field is increasingly 3-dimensional.

7.27 Water at 27°C flows at 2.2 m/s in a 0.04 m I.D. thin-walled pipe. Air at 227°C flows across it at 7.6 m/s. Find the pipe wall temperature.

WATER

T_water = 27; % °C,
properties at T_water

rho_water = 996.5; % kg/m3
cp_water = 4.181; % kJ/kgK
k_water = 0.6103; % W/mK
Pr_water = 5.85; % -
ni_water = 8.568e-7; % m2/s
mu_water = 8.5380e-04; % Pa*s

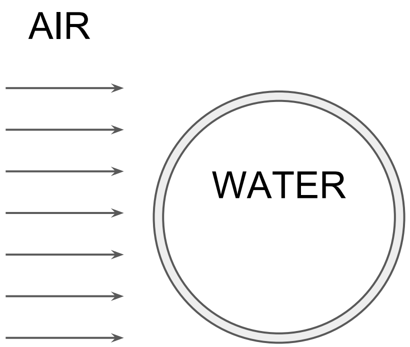
u_water=2.2; % m/s

AIR

T_air=227; % °C,
properties at T_mean = 127 °C

rho_air = 0.8821; % kg/m3
cp_air = 1.014; % kJ/kgK
k_air = 0.03328; % W/mK
Pr_air = 0.704; % -
ni_air = 2.619e-5; % m2/s

u_air=7.6; % m/s



WATER

Re_water = 1.0271e+05

$$f = (0.790 \ln Re_D - 1.64)^{-2}$$

f_water = 0.017891

$$\mathrm{Nu}_D = \frac{(f/8)(\mathrm{Re}_D - 1000)\,\mathrm{Pr}}{1 + 12.7\sqrt{f/8}\left(\mathrm{Pr}^{2/3} - 1\right)}$$

for $2300 \le \mathrm{Re}_D \le 5 \times 10^6$.

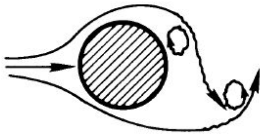
Nu_water = 566.38

h_water = 8641.5 W/m^{2}K

R_water = 1/h_water = 1.1572e-04 m^{2}K/W

AIR

Re_air = 1.1607e+04



$150 \leqslant \mathrm{Re}_D < 300$ Transition range to turbulence in vortex.

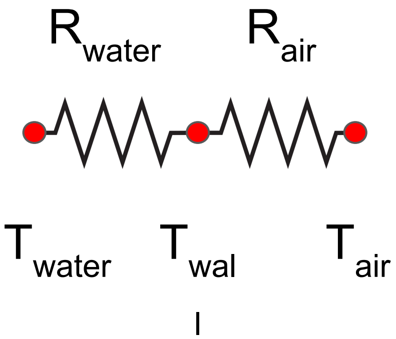
$300 \leqslant \mathrm{Re}_D \gtrsim 3 \times 10^5$ Vortex street is fully turbulent, and the flow field is increasingly 3-dimensional.

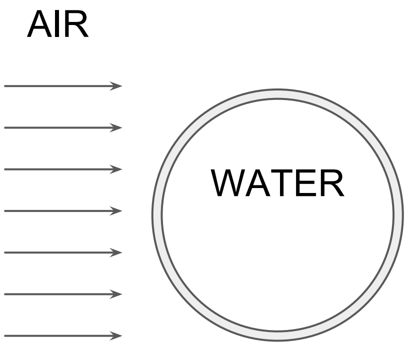
$$\overline{\mathrm{Nu}}_D = 0.3 + \frac{0.62\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1 + \left(\frac{\mathrm{Re}_D}{282{,}000}\right)^{5/8}\right]^{4/5} \qquad (7.65)$$

Nu_air = 58.055

h_air = 48.302 W/m^{2}K

R_air = 1/h_air = 0.0207 m^{2}K/W





$R\_tot = R\_water + R\_air = 0.020819\ m^2K/W$

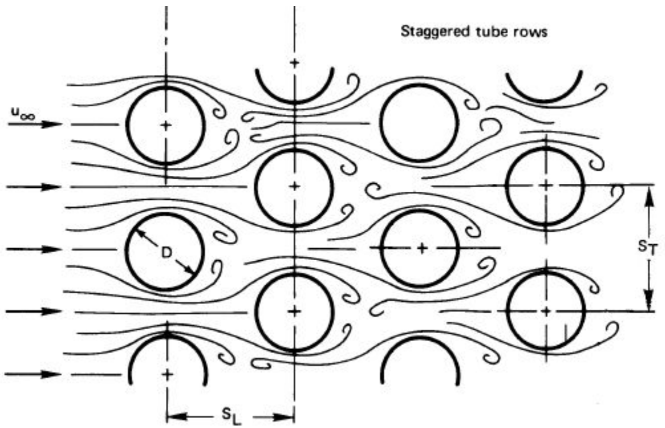
$$Q = UA(T_{air} - T_{water}) = A/R_{tot}(T_{air} - T_{water})$$

$$Q = A/R_{air}(T_{air} - T_{wall})$$

$$T_{wall} = T_{air} - R_{air}/R_{tot}*(T_{air} - T_{water}) = 28.11\ °C$$

7.30 Water at 37°C flows at 3 m/s across at 6 cm O.D. tube that is held at 97°C. In a second configuration, 37°C water flows at an average velocity of 3 m/s through a bundle of 6 cm O.D. tubes that are held at 97°C. The bundle is staggered, with $S_T/S_L = 2$. Compare the heat transfer coefficients for the two situations.

Assume: ST = 4*D; SL = 2*D.



DATA 1

Properties at 67 °C

rho=979.5; % kg/m3
cp=4.189; % kJ/kgK
k=0.6605; % W/mK
Pr=2.68; % -
ni=4.308e-7; % m2/s
Pr_wall=1.81; % -

DATA 2

Properties at 37 °C

rho=993.29; % kg/m3
cp=4.1795; % kJ/kgK
k=0.62442; % W/mK
Pr=4.63; % -
ni=6.9596e-7; % m2/s
Pr_wall=1.81; % -

ONE CYLINDER

$$\overline{\mathrm{Nu}}_D = 0.3 + \frac{0.62\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1+(0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}_D}{282{,}000}\right)^{5/8}\right]^{4/5} \tag{7.65}$$

Re_one = 4.1783e+05
Nu_one = 1011.4
h_one = 1.1134e+04 W/m^{2}K

TUBE BUNDLE

$$\overline{\mathrm{Nu}}_D = \mathrm{Pr}^{0.36}\,(\mathrm{Pr}/\mathrm{Pr}_w)^n\,\mathrm{fn}\,(\mathrm{Re}_D) \quad \text{with } n = \begin{cases} 0 & \text{for gases} \\ \frac{1}{4} & \text{for liquids} \end{cases} \tag{7.70}$$

where properties are to be evaluated at the local fluid bulk temperature, except for Pr_w, which is evaluated at the uniform tube wall temperature, T_w.

$\mathrm{Re}_D > 2 \times 10^5$:

aligned rows: $\mathrm{fn}\,(\mathrm{Re}_D) = 0.033\,\mathrm{Re}_D^{0.8}$ (7.71f)

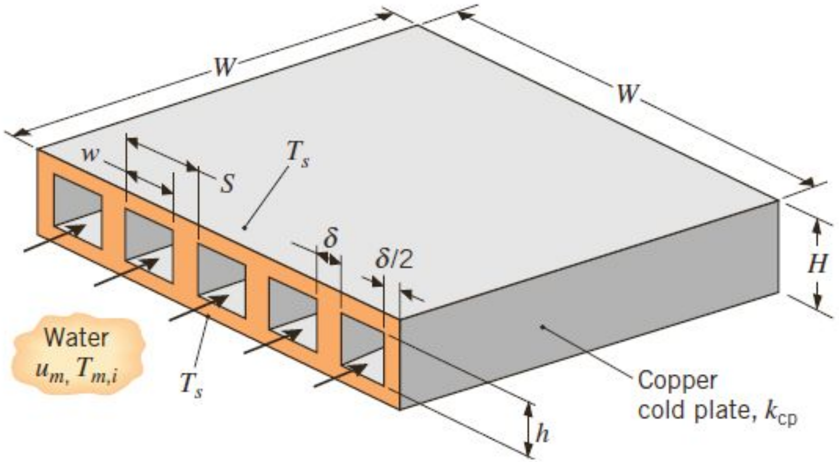
staggered rows: $\mathrm{fn}\,(\mathrm{Re}_D) = 0.031\,(S_T/S_L)^{0.2}\,\mathrm{Re}_D^{0.8}$, $\mathrm{Pr} > 1$ (7.71g)

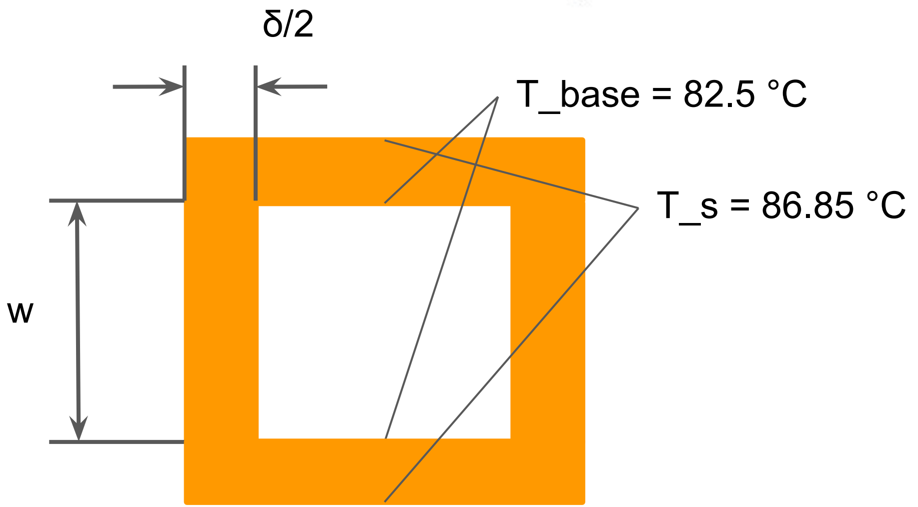
$\overline{\mathrm{Nu}}_D = 0.027\,(S_T/S_L)^{0.2}\,\mathrm{Re}_D^{0.8}$, $\mathrm{Pr} = 0.7$ (7.71h)

u_max = ST/(ST-D)*u = 4 m/s

Re_staggered = 3.4485e+05
Nu_staggered = 2105
h_staggered = 2.1907e+04 W/m^{2}K

Consider a cold plate of width $W = 100$ mm and height $H = 10$ mm, with 10 square channels of width $w = 6$ mm and a spacing of $\delta = 4$ mm between channels. Water enters the channels at a temperature of $T_{m,i} = 300$ K and a velocity of $u_m = 2$ m/s. If the top and bottom cold plate surfaces are at $T_s = 360$ K, what is the outlet water temperature and the total rate of heat transfer to the cold plate? The thermal conductivity of the copper is 400 W/m·K, while average properties of the water may be taken to be $\rho = 984$ kg/m^3, $c_p = 4184$ J/kg·K, $\mu = 489 \times 10^{-6}$ N·s/m^2, $k = 0.65$ W/m·K, and $Pr = 3.15$. Is this a good cold plate design? How could its performance be improved?





Reynolds number, friction factor and heat transfer coefficient

The hydraulic diameter, which was introduced in connection with eqn. (7.59b), provides a basis for approximating heat transfer coefficients in noncircular ducts. Recall that the hydraulic diameter is defined as

$$D_h \equiv \frac{4\,A_c}{P} \tag{7.60}$$

Dh = w

$$f = (0.790 \ln Re_D - 1.64)^{-2}$$

f = 0.024936

Δp = f*(ρ/2*u²)*L/Dh = 0.81 kPa

Mechanical power = 10*m*Δp/rho = 0.59 W

$$\boxed{\mathrm{Nu}_D = \frac{(f/8)\,(\mathrm{Re}_D - 1000)\,\mathrm{Pr}}{1 + 12.7\sqrt{f/8}\left(\mathrm{Pr}^{2/3} - 1\right)}}$$

for $2300 \le \mathrm{Re}_D \le 5 \times 10^6$.

evaluated at T_b using eqn. (7.41) or (7.43). For liquids, one then corrects by multiplying with a viscosity ratio. Over the interval $0.025 \le (\mu_b/\mu_w) \le 12.5$,

$$\mathrm{Nu}_D = \mathrm{Nu}_D\Big|_{T_b}\left(\frac{\mu_b}{\mu_w}\right)^n \qquad \text{where } n = \begin{cases} 0.11 & \text{for } T_w > T_b \\ 0.25 & \text{for } T_w < T_b \end{cases} \tag{7.44}$$

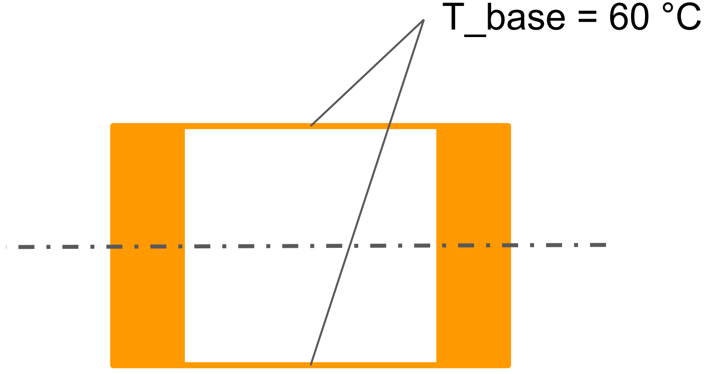
For gases and a temperatures ratio *in kelvins* within $0.27 \le (T_b/T_w) \le 2.7$,

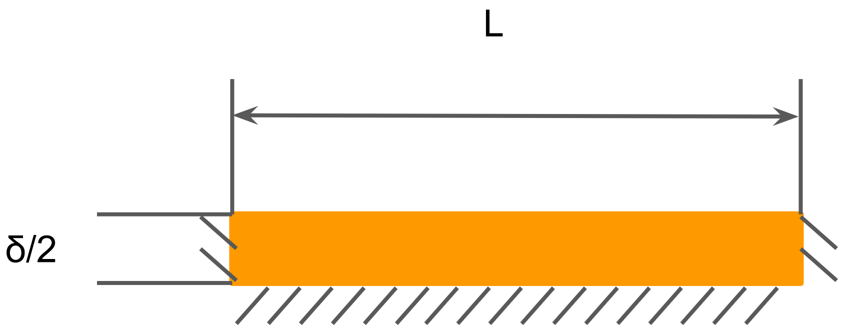
$$\mathrm{Nu}_D = \mathrm{Nu}_D\Big|_{T_b}\left(\frac{T_b}{T_w}\right)^n \qquad \text{where } n = \begin{cases} 0.47 & \text{for } T_w > T_b \\ 0 & \text{for } T_w < T_b \end{cases} \tag{7.45}$$

Nu = 130.22

h = 1.4107e+04 W/m^{2}K

Fin efficiency



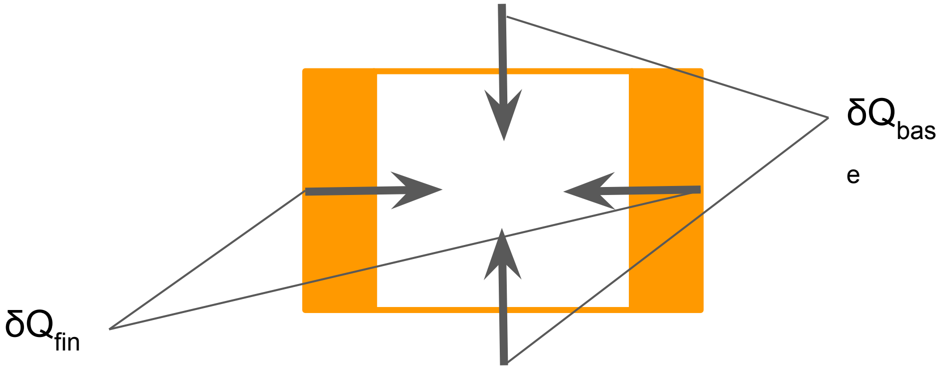


$$\eta_f = \frac{\sqrt{(\bar{h}P)(kA)}(T_0 - T_\infty)\tanh mL}{\bar{h}(PL)(T_0 - T_\infty)} = \frac{\tanh mL}{mL} \qquad (4.54)$$

$$\sqrt{\bar{h}PL^2/kA} \equiv mL$$

eta = 0.95 mL = 0.392

Energy balance



δQ_{base} =
$h*2w*dx*(T_{base}-T)$

$\delta Q_{fin} = h*2w*\boldsymbol{\eta}*dx*(T_{base}-T)$

$\delta Q_{base} + \delta Q_{fin} = m*c_p*dT$

$$\boxed{m*c_p*dT = 2*h*w*(1+\boldsymbol{\eta})*dx*(T_{base}-T)}$$

$$\boxed{\frac{T_{b_{\text{out}}} - T_{b_{\text{in}}}}{T_w - T_{b_{\text{in}}}} = 1 - \exp\left(-\frac{\overline{h}PL}{\dot{m}c_p}\right)}$$

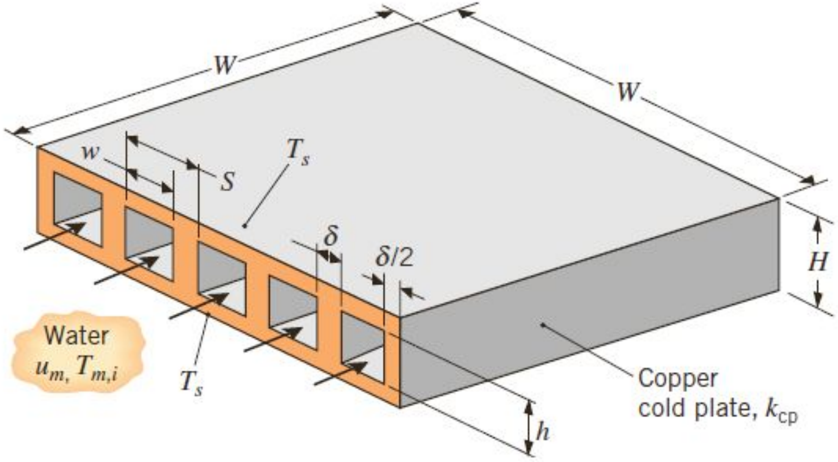
$$\int_0^L \frac{hP}{\dot{m}c_p}\,dx = -\int_{T_{b_{\text{in}}}}^{T_{b_{\text{out}}}} \frac{d(T_w - T_b)}{(T_w - T_b)}$$

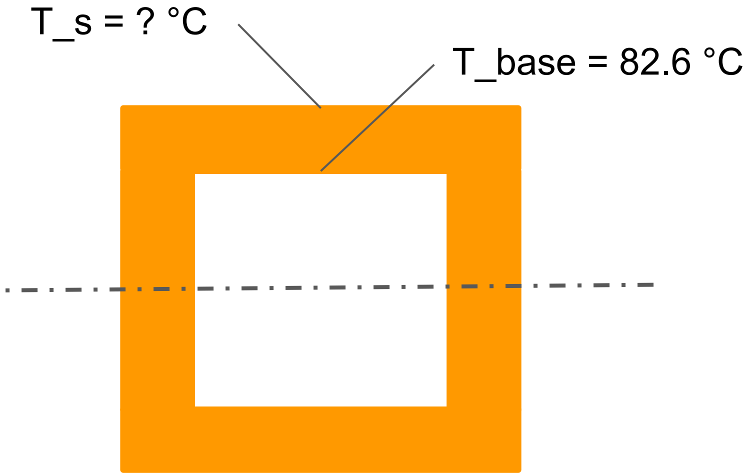
$$\frac{P}{\dot{m}c_p}\int_0^L h\,dx = -\ln\left(\frac{T_w - T_{b_{\text{out}}}}{T_w - T_{b_{\text{in}}}}\right)$$

Tout = 32.715 °C

$Q = 10*m*c_p*\Delta T = 1.739e+4$ W

Validation





$$Q/2 = k_{copper} * A_{base} / (\delta/2) * (T_s - T_{base})$$

$$T_s = 86.847\ °C = 360\ K$$