IIA GA3 Heat Exchanger Project

Introductory Details for Worked Example

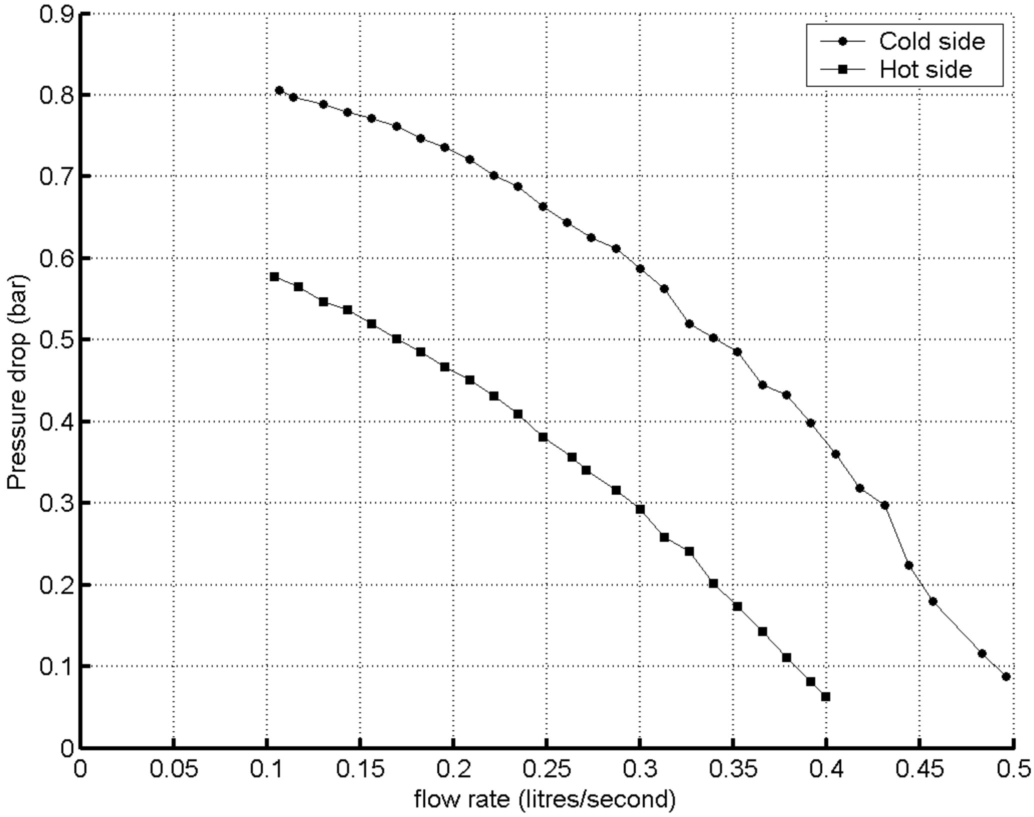


Figure W1: Characteristics of cold and hot circuit of CUED test rig for this worked example.

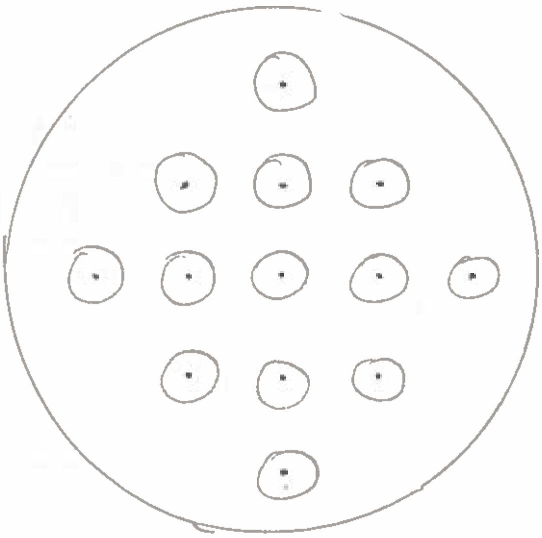
Note: A few minor errors have been corrected in the equations listed in the Shell and Tube Heat Exchanger document. The associated corrections have not yet been implemented in this worked example.

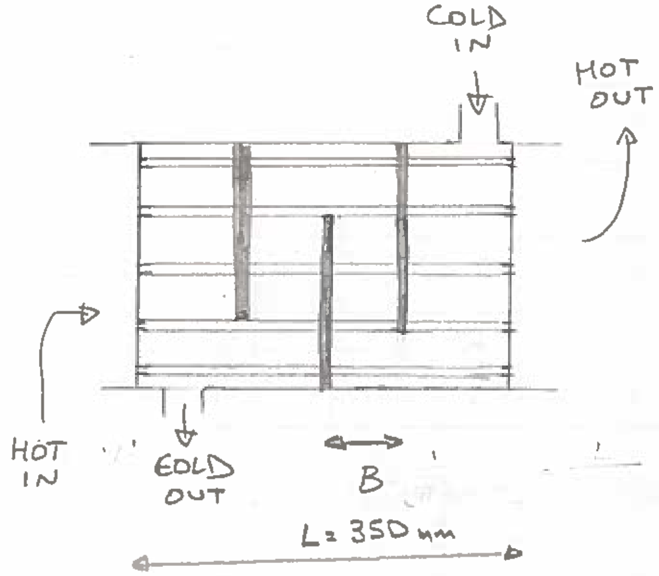
Some comments have been added.

JPL 03May2018.

HEAT EXCHANGER DESIGN PROJECT

WORKED EXAMPLE





(NOT TO SCALE)

CONFIGURATION

- I have 13 tubes, in square pitch, $Y = 14$ mm
- One shell - one tube pass
- Number of baffles = 9

$$\Rightarrow B = \frac{L}{9+1} = 35 \text{ mm}$$

Properties @ 40°C
$C_p = 4179$ J/kgK
$k = 0.632$ W/mK
$\mu = 6.51 \times 10^{-4}$ kg/ms
$Pr = 4.31$
$\rho = 990.1$
$k_{tube} = 386$ W/mK

- Counterflow arrangement (for one pass, this is pure counterflow).

HYDRAULIC DESIGN

- Say $\dot{m}_h = 0.45$ kg/s $\Rightarrow$ $\dot{m}_{tube} = \frac{\dot{m}_h}{13} = 0.0346$ kg/s

$$V_{tube} = \frac{\dot{m}_{tube}}{\rho \frac{\pi d_i^2}{4}} = 1.237 \text{ m/s}$$

$$Re_t = \frac{V_{tube}\, \rho\, d_i}{\mu} = 11280$$

$$V_{nozzle,hot} = \frac{\dot{m}_h}{\rho \frac{\pi d_{noz}^2}{4}} = 1.603 \text{ m/s}$$

- $\sigma = \dfrac{13 \cdot \frac{\pi}{4} d_i^2}{\frac{\pi}{4} D_{sh}^2} = 0.11 \Rightarrow K_c \approx 0.45$, $K_e \approx 0.8$ (from Fig 7)

- $\Delta P_{tube} = f \cdot \frac{L}{d_i} \cdot \frac{1}{2} \rho V_t^2$, $f = 0.0304$ (Holman, p. 290) (or p. 289, Eq 6-8)

(For those of you who want to code these eqtns, you may use $f = (1.82 \log_{10} Re - 1.64)^{-2}$ to acceptable accuracy)

$\Rightarrow \Delta P_{tube} =$ 1.343 kPa

- $\Delta P_{in+out} = \rho V_t^2 (K_c + K_e) = 1.89$ kPa half factor missing
- $\Delta P_{nozzles} = \rho V_{noz}^2 = 2.54$ kPa twice times half factor missing

$\Rightarrow \Delta P_{hot,\,overall} \simeq 0.06$ bar

– Now look at Fig 6: At $\dot{m}_h = 0.45$ kg/s, $\Delta P_{hot} \sim 0.15$ bar

so we have to increase $\dot{m}_h$ slightly.

With $\dot{m}_h = 0.48$ kg/s, $\Delta P_{hot} \approx 0.07$ bar, which is close enough to the curve of Fig 6 $\Rightarrow$ $\underline{\underline{\dot{m}_h = 0.48 \text{ kg/s}}}$.

- Say $\dot{m}_c = 0.5$ kg/s

A_{sh} (Eqtn 6) $= \frac{D_{sh}}{Y}(Y - d_o) \cdot B = \frac{64 \times 10^{-3}}{14 \times 10^{-3}} (14 - 8) \times 10^{-3} \cdot 35 \times 10^{-3} = 9.6 \times 10^{-4}\ m^2$

$V_{sh} = \frac{\dot{m}_c}{\rho A_{sh}} \Rightarrow V_{sh} = 0.526$ m/s

Shell side velocity ought to be a representative velocity of the flow transverse to the tubes.
Vsh*(width-tube blockage)*rho=mass flow rate

$Re_{sh} = \frac{V_{sh} \cdot d_o \cdot \rho}{\mu} = 6400$

$\Rightarrow \Delta P_{shell} = 4 \cdot \underset{\substack{\uparrow \\ \text{square} \\ \text{pitch}}}{0.34} \cdot (6400)^{-0.15} \cdot 13 \cdot \rho V_{sh}^2 = 1.3$ kPa

$\Delta P_{cold,\,nozzle} = \rho V_{noz,c}^2 = 3.14$ kPa

$\Rightarrow \Delta P_{shell,\,overall} \simeq 0.04$ bar

Low, therefore $\underline{\underline{\dot{m}_c = 0.5 \text{ kg/s}}}$ is OK.

THERMAL DESIGN

- Once my mass flow rates have been fixed, I must calculate the outlet temperatures for my design.

So $\quad Q = \dot{m}_c C_p \left(T_{c,out} - T_{c,in} \right) \qquad (\mathrm{I})$

$\quad\quad = \dot{m}_h C_p \left(T_{h,in} - T_{h,out} \right) \qquad (\mathrm{II})$

$\quad\quad = U \cdot A \cdot \Delta T_{lm} \qquad (\mathrm{III})$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}, \quad \Delta T_1 = T_{h,in} - T_{c,out}, \quad \Delta T_2 = T_{h,out} - T_{c,in} \quad \text{(for counterflow)}$$

- Eqtn 3 :

$$U = \frac{1}{\frac{1}{h_i} + \frac{d_i \ln(d_o / d_i)}{2 k_{tube}} + \frac{d_i}{d_o h_o}}$$

$$h_i = \frac{Nu_i \cdot k}{d_i}$$

$$Nu_i = 0.023 \cdot Re^{0.8} \cdot Pr^{0.3} = 66$$

(12036 for $\dot{m}_h = 0.48$ kg/s)

$$\Rightarrow h_i = 6950 \ \mathrm{W/m^2K}$$

$$h_o = \underbrace{0.15 \cdot (6400)^{0.6} \cdot Pr^{0.3}}_{Nu_o} \cdot \frac{k}{d_o} = 3530 \ \mathrm{W/m^2K}$$

$$\Rightarrow U = 2789 \ \mathrm{W/m^2K}$$

- Total area $\quad A_i = \pi d_i \cdot L \cdot 13 = 0.0858 \ \mathrm{m^2}$

- I solve Eqtn I – III above by an iteration process, to get

$$\underline{\underline{T_{c,out} = 23.93 \, ^\circ C}} \qquad (T_{h,in} = 60^\circ C, \ T_{c,in} = 20^\circ C)$$

$$\underline{\underline{T_{h,out} = 55.42 \, ^\circ C}}$$

- **Note**: I have used an iteration procedure to get the outlet T converge to the second decimal place.
This is a time-consuming part, if you're doing hand-calculations & it is not realistic to aim for too much accuracy.

- Overall heat transfer

$$Q = \dot{m}_c c\,(T_{c,out} - T_{c,in}) = \underline{\underline{8221.2\,W}}$$

- Think (& estimate) how uncertain is this value in view of the accuracy of your iteration to find $T_{c,out}$ & $T_{h,out}$. It may be important especially if you the calculations by hand.

- Effectiveness $= \dfrac{Q}{(\dot{m}c)_{min}(T_{h,in} - T_{c,in})} = \underline{\underline{10.2\%}}$

$(\dot{m}c)_{min}$ ↑ 0.48×4179 ↑ ~~cold~~ hot fluid

- NOTES:
  - I have one-shell, one-pass, so I need not employ the correction factor F for the ΔT_{em}.
  - If $\dot{m}_c = \dot{m}_h$ (& c_p = constant), the LMTD concept fails. In that case, $\Delta T_{em} = \Delta T_1 = \Delta T_2$ i.e. the temperature difference between the two streams is constant along the exchanges. Eqtns I–III (p.3 of this example) are still valid.
  - With your own Excel spreadsheet or code for these eqtns, ~~or by DEVIZE for more accurate predictions~~, you can now re-think the original design to increase Q.