Hydronic systems – design principles

Heating, Ventilation and Air Conditioning Systems
A.A. 2022/23

Jacopo Vivian
4/5/2023
Bernoulli’s principle

For incompressible fluids with steady flow:

\[ p + \rho gh + \frac{\rho u^2}{2} = \text{cost}. \]

The pressure drops of a closed circuit should be equal to the head of the pump:

\[ \Delta p = \sum_j \rho \left( f_j \frac{L_j}{D_j} + \beta_j \right) \frac{u^2}{2} = \sum_j \frac{\rho}{2S_j^2} \left( f_j \frac{L_j}{D_j} + \beta_j \right) Q_{v,j}^2 \]
Pressure losses

The pressure losses in a hydronic circuit are of two types:

- **Distributed (or continuous) losses**: proportional to the pipe length
  \[ \Delta p = f \frac{L \rho u^2}{D} \frac{1}{2} \]

- **Localized losses**: dependent on the element (fitting, valve, heat exchanger etc) encountered by the flow
  \[ \Delta p = \beta \frac{\rho u^2}{2} \]
Continuous pressure losses

Friction factor
In general, the friction factor $f$ depends on the Reynolds number and on relative pipe roughness

$$f = f \left( \frac{\varepsilon}{D}, Re \right)$$
Continuous pressure losses

Reynolds number

The Reynolds number is the ratio between between inertial and viscous forces on a fluid in motion

\[ Re = \frac{u D \rho}{\mu} = \frac{u D}{v} \]

\( \mu = \) dynamic viscosity \([\text{Pa} \cdot \text{s}] \) or \([\text{N} \cdot \text{s}/\text{m}^2]\) or \([\text{kg}/(\text{m} \cdot \text{s})]\)

\( v = \frac{\mu}{\rho} \) kinematic viscosity \([\text{m}^2/\text{s}]\)
Continuous pressure losses

Friction factor

- At low Reynolds numbers ($Re < 2000$), the flow is laminar and the friction factor depends only on the Reynolds number

\[ f = \frac{64}{Re} \]

- At high Reynolds numbers ($Re > 3000$), the flow is turbulent and the friction factor depends also on relative pipe roughness according to Colebrook’s correlation:

\[ \frac{1}{f^{0.5}} = -2 \log_{10} \left( \frac{2.51}{Re \cdot f^{0.5}} + \frac{\varepsilon}{3.71} \right) \]
Continuous pressure losses

Friction factor
The graphical representation of Colebrook’s correlation is the Moody Diagram.
Continuous pressure losses

Friction factor

Alternatively, some approximated correlations can be used to calculate the friction factor. Their validity is limited to the case considered:
Continuous pressure losses

Friction factor

Alternatively, some approximated correlations can be used to calculate the friction factor. Their validity is limited to the case considered:

- **Low roughness**

Commercially available copper, inox, multi-layer and plastic pipes can be considered as low roughness pipes (0.001 < ε < 0.007 mm)

\[ f = 0.316 \text{Re}^{-0.25} \]
Continuous pressure losses

**Low roughness pipes**

Commercially available copper, inox, multi-layer and plastic pipes can be considered as low roughness pipes ($0.001 < \varepsilon < 0.007$ mm)

$$r = 14.68 \nu^{0.25} \rho \frac{G^{1.75}}{D^{4.75}}$$

- $r$: mm.w.c./m
- $\nu$: m$^2$/s
- $\rho$: kg/m$^3$
- $G$: l/h
- $D$: mm
Continuous pressure losses

Friction factor

Alternatively, some approximated correlations can be used to calculate the friction factor. Their validity is limited to the case considered:

- **Average roughness**

  Commercially available iron and galvanized steel pipes can be considered as average roughness pipes (0.020 < $\varepsilon$ < 0.090 mm)

  \[ f = 0.07 \, Re^{-0.13} \, D^{-0.14} \]
Continuous pressure losses

Average roughness pipes
Commercially available iron and galvanized steel pipes can be considered as average roughness pipes ($0.020 < \varepsilon < 0.090 \text{ mm}$)

$$r = 3.3 \nu^{0.13} \rho \frac{G^{1.87}}{D^{5.01}}$$

- $r$: mm w.c./m
- $\nu$: m$^2$/s
- $\rho$: kg/m$^3$
- $G$: l/h
- $D$: mm
Continuous pressure losses

Effects of temperature

The viscosity and density of the water are affected by its temperature.
Continuous pressure losses

Effects of temperature
The viscosity and density of the water are affected by its temperature.

[Source: www.engineersedge.com]
Continuous pressure losses

Friction factor
Alternatively, some approximated correlations can be used to calculate the friction factor. Their validity is limited to the case considered:

- **High roughness**

  Pipes with deposits and corroded pipes can be considered as high roughness pipes (0.200 < $\varepsilon$ < 1.000 mm)

  $$f = \ldots$$
Continuous pressure losses

Fig. 14  Friction Loss for Water in Commercial Steel Pipe (Schedule 40)
Continuous pressure losses

Fig. 15  Friction Loss for Water in Copper Tubing (Types K, L, M)
Continuous pressure losses

Fig. 16 Friction Loss for Water in Plastic Pipe (Schedule 80)
Localized pressure losses

Loss coefficients for valves

<table>
<thead>
<tr>
<th>Localised loss type</th>
<th>Symbol</th>
<th>8-16 mm</th>
<th>16-28 mm</th>
<th>30-54 mm</th>
<th>&gt; 54 mm</th>
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</thead>
<tbody>
<tr>
<td>Shut-off valve</td>
<td></td>
<td>10,0</td>
<td>8,0</td>
<td>7,0</td>
<td>6,0</td>
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<tr>
<td>Total passage ball valve</td>
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<td>4-way valve</td>
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<td>5-way valve</td>
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<td>8,0</td>
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<td>Passage through radiator</td>
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<td>Passage through boiler</td>
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[source: M. Doninelli, Design principles of hydronic heating systems, Caleffi Handbooks]
Localized pressure losses

Loss coefficients for elbows, bends, section changes, T-joints and other elements

<table>
<thead>
<tr>
<th>Localised loss type</th>
<th>Symbol</th>
<th>8-16 mm</th>
<th>18-28 mm</th>
<th>30-54 mm</th>
<th>&gt;54 mm</th>
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<tbody>
<tr>
<td>Narrow bend 90°</td>
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<td>1.0</td>
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<td>Wide bend 90°</td>
<td><img src="image3" alt="Symbol" /></td>
<td>1.0</td>
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<tr>
<td>Narrow bend U</td>
<td><img src="image4" alt="Symbol" /></td>
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<td>2.0</td>
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<td>1.0</td>
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<tr>
<td>Normal bend U</td>
<td><img src="image5" alt="Symbol" /></td>
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<td>Wide bend U</td>
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<td>Angle joint (0°-40°)</td>
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[source: M. Doninelli, Design principles of hydronic heating systems, Caleffi Handbooks]
Localized pressure losses

Loss coefficients for valves, section changes, T-joints and other elements

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<th>Diametro nominale (DN)</th>
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[source: Miniguida AICARR]
Pipe design

Rules of thumb for pipe sizing

- The general range for pipe sizing is between 100 and 400 Pa/m, with the mean value of 250 Pa/m being a commonly used target for pipe design.
- Upper limits to avoid noise are 1.2 m/s for piping with D<50 mm and 400 Pa/m for bigger pipes, where higher velocities are allowed.

Note: Noise is not directly caused by high velocity, but rather by free air, pressure drops, turbulence or a combination of these that cause cavitation of flashing of water into steam.
Pipe design

Rules of thumb for pipe sizing

Note: Noise is not directly caused by high velocity, but rather by free air, pressure drops, turbulence or a combination of these that cause cavitation of flashing of water into steam.

▪ **Air in hydronic systems is undesirable** because (i) it causes flow noise, (ii) allows oxygen to react with piping material, (iii) might prevent flow in parts of a system.

▪ The solubility of air in water increases with pressure and decreases with temperature: therefore, **air separation** is best achieved in the point of **lowest pressure and/or highest temperature**.
Pipe design

Rules of thumb for pipe sizing

Note: Air can be entrained in the water and carried to separation units at flow velocities higher than 0.5-0.6 m/s in pipes with D<50 mm.

▪ For this reason, **a minimum velocity of 0.6 m/s** is recommended for pipes with D<50 mm.

▪ For bigger pipes, velocities that correspond to at least 75 Pa/m are sufficient.

Note: The constraint of minimum velocity is particularly important in the upper floors of high rise buildings, where air tends to come out due to reduced pressures.
Pipe design

Sizing procedure

1. Given the heat load of the building zones, size the terminal units and calculate the corresponding flow rates

2. Sketch the distribution system connecting the heat supply station to the terminal units (see previous lecture)

3. Set a target value to the flow velocity (m/s) or to the linear pressure loss (Pa/m) in all pipes, valves and fittings

4. Calculate the corresponding diameter and find the closest available diameter

5. Recalculate velocities and pressures according to the selected diameters and check if they are within upper and lower limits.
Pump selection

Characteristic curve of the circuit

\[ \Delta p = \sum_j \frac{\rho}{2S_j^2} \left( f_j \frac{L_j}{D_j} + \beta_j \right) Q_{v,j}^2 = k Q_v^2 \]
Pump selection

Characteristic curves of the pump

\[ P_i = \rho Q_v g \Delta z \]

\[ P = \frac{\rho Q_v g \Delta z}{\eta} = \frac{Q_v \Delta p}{\eta} \]
Pump selection

Efficiency of the pump

Hydraulic efficiency
\[ \eta = \frac{P_i}{P} \]

Electric efficiency
\[ \eta_e = \frac{P}{P_e} \]

Overall efficiency
\[ \eta_g = \eta \cdot \eta_e = \frac{P_i}{P_e} \]
Pump selection

Net Positive Suction Head (NPSH)

\[ NPSH_a \geq NPSH_r \]

\[ NPSH_a = p_s + \rho g \Delta z - \Delta p_f - p_v(T) \]

Pump selection

Net Positive Suction Head (NPSH)

\[ NPSH_a \geq NPSH_r \]

\[ NPSH_a = p_a + p_{in} + \frac{\rho u^2}{2} - p_v(T) \]

Fig. 33 Net Positive Suction Pressure Available

Pump selection

Affinity laws

\[
\frac{Q_{v1}}{Q_{v2}} = \frac{n_1}{n_2} \quad \frac{\Delta p_1}{\Delta p_2} = \frac{\Delta z_1}{\Delta z_2} = \left(\frac{n_1}{n_2}\right)^2
\]

\[\eta_1 \cong \eta_2 \quad \frac{\Delta P_1}{\Delta P_2} = \left(\frac{n_1}{n_2}\right)^3\]
Pump selection

Characteristic curves of the pump

\[
\frac{Q_{v1}}{Q_{v2}} = \frac{D_1}{D_2}
\]

\[
\frac{\Delta p_1}{\Delta p_2} = \left(\frac{D_1}{D_2}\right)^2
\]

\(n = 1.450 \text{ giri/min}\)
Pump selection

Pumps in series

\[ P_1 + P_2 \]
Pump selection

Pumps in parallel

![Diagram of pumps in parallel]
Pump selection

Working point

Fig. 5  Pump Curve and System Curve
Pump selection

Working point

Fig. 5  Pump Curve and System Curve

Fig. 35  Pump Selection Regions