Hydronic systems – design principles

J. Vivian

Bernoulli's principle

For incompressible fluids with steady flow:

$$\mathrm{p} + \rho g h + \frac{\rho u^2}{2} = cost.$$

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

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

Pressure losses

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

- **Distributed (or continuous) losses**: proportional to the pipe length

$$\Delta\mathrm{p} = f\,\frac{L}{D}\frac{\rho u^2}{2}$$

- **Localized losses**: dependent on the element (fitting, valve, heat exchanger etc) encountered by the flow

$$\Delta\mathrm{p} = \beta\,\frac{\rho u^2}{2}$$

Pressure losses

$$\Delta p = \sum_j \rho \left(f_j \frac{L_j}{D_j} + \beta_j \right) \frac{u^2}{2} = \sum_j \underbrace{\frac{\rho}{2 S_j{}^2} \left(f_j \frac{L_j}{D_j} + \beta_j \right)}_{R_j \,=\, hydraulic\ resistance} Q_{v,j}{}^2$$

The pressure loss can be calculated with:

$$\Delta p = \sum_j R_j \, Q_{v,j}{}^2$$

Over a single j-th element:

$$\Delta p = R \, Q_v{}^2$$

Electrical analogy:

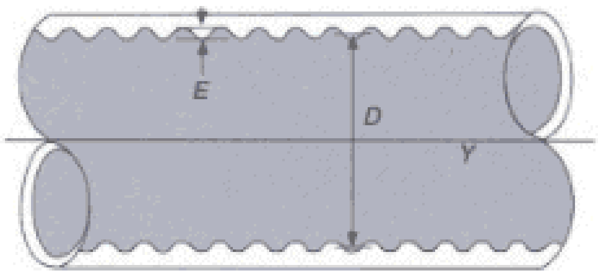
$$\Delta V = R_{el} I$$

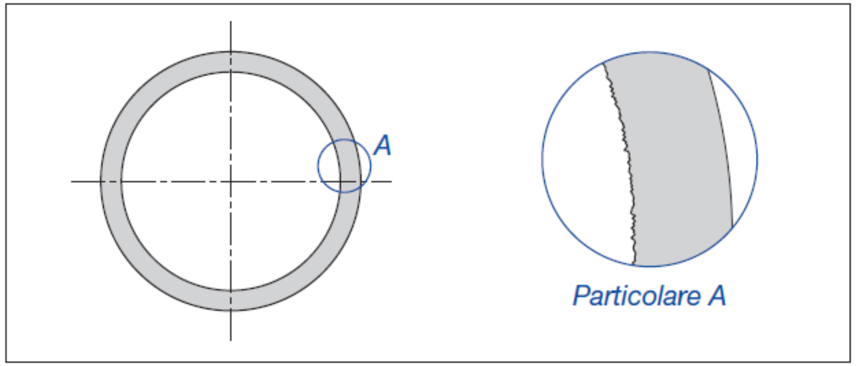
Continuous pressure losses

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

$$Re = \frac{u\,D\,\rho}{\mu} = \frac{u\,D}{\nu}$$

μ = dynamic viscosity [Pa·s] or [N·s/m²] or [kg/(m·s)]

$\nu = \frac{\mu}{\rho}$ kinematic viscosity [m²/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

$$f = \frac{64}{Re}$$

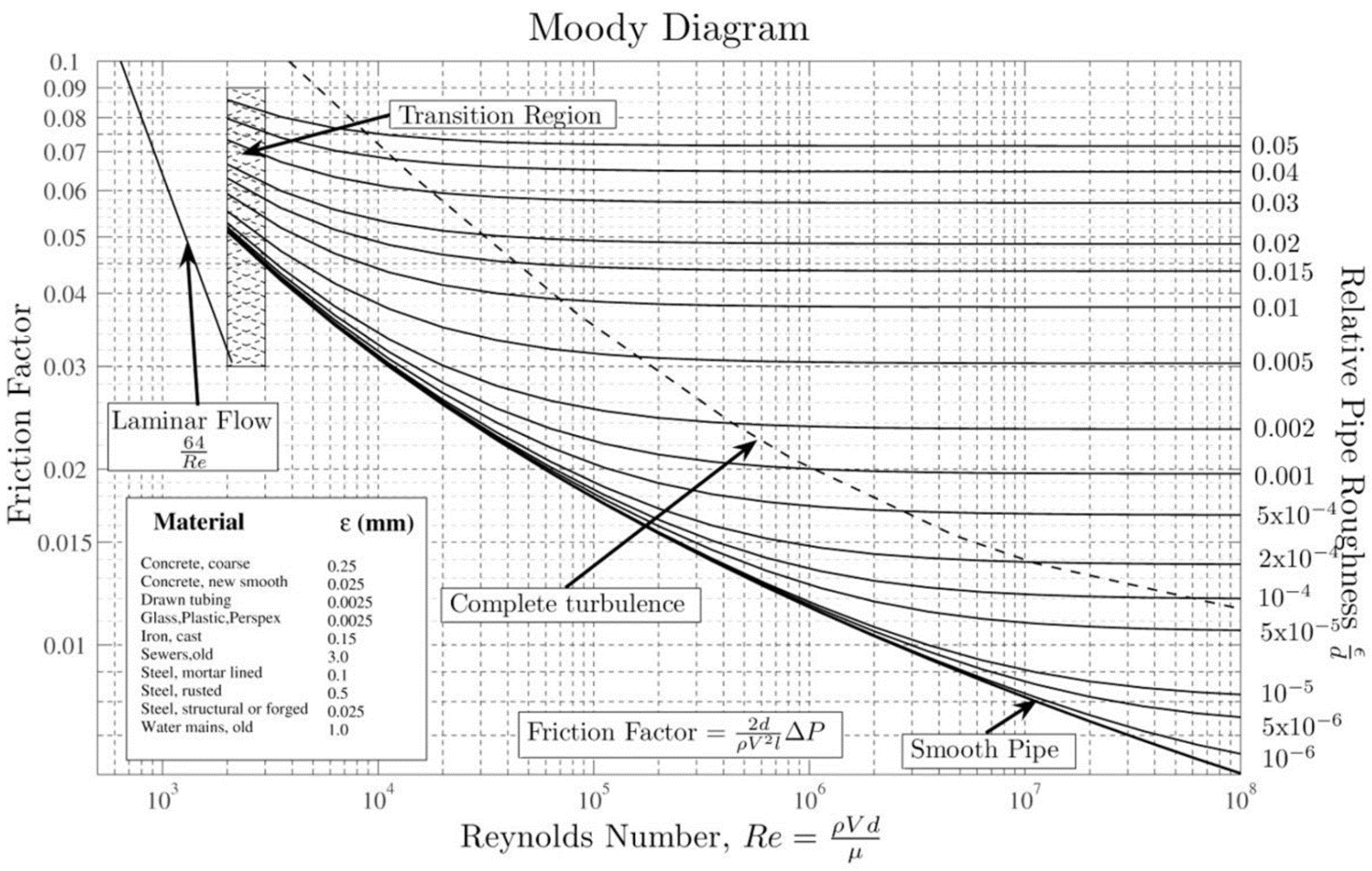
- At high Reynolds numbers ($Re > 3000$), the flow is turbulent and the friction factor depends also on relative pipe roughness according to Colebrook's correlation:

$$\frac{1}{f^{0.5}} = -2\log_{10}\left(\frac{2.51}{Re \cdot f^{0.5}} + \frac{\frac{\varepsilon}{D}}{3.71}\right)$$

Continuous pressure losses

Friction factor

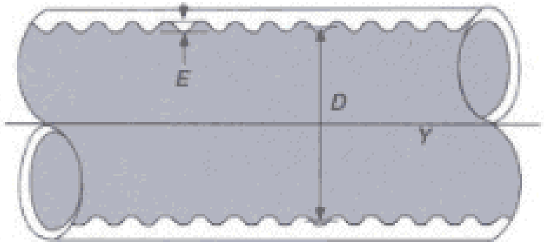
The graphical representation of Colebrook's correlation is the Moody Diagram.



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:



Material	Absolute Roughness (mm)
Copper, Lead, Brass, Aluminum (new)	0.001 - 0.002
PVC and Plastic Pipes	0.0015 - 0.007
Flexible Rubber Tubing - Smooth	0.006-0.07
Stainless Steel	0.0015
Steel Commercial Pipe	0.045 - 0.09
Weld Steel	0.045
Carbon Steel (New)	0.02-0.05
Carbon Steel (Slightly Corroded)	0.05-0.15
Carbon Steel (Moderately Corroded)	0.15-1
Carbon Steel (Badly Corroded)	1-3
Asphalted Cast Iron	0.1-1
New Cast Iron	0.25 - 0.8
Worn Cast Iron	0.8 - 1.5
Rusty Cast Iron	1.5 - 2.5
Galvanized Iron	0.025-0.15
Wood Stave	0.18-0.91
Wood Stave, used	0.25-1
Smoothed Cement	0.3
Ordinary Concrete	0.3 - 1
Concrete – Rough, Form Marks	0.8-3

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 ($0.001 < \varepsilon < 0.007$ mm)

$$f = 0.316\, Re^{-0.25}$$

Continuous pressure losses

Low roughness pipes

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

l/h

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

mm.w.c./m

m^2/s

kg/m^3

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 < ε < 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 < ε < 0.090 mm)

l/h

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

mm.w.c./m

m^2/s

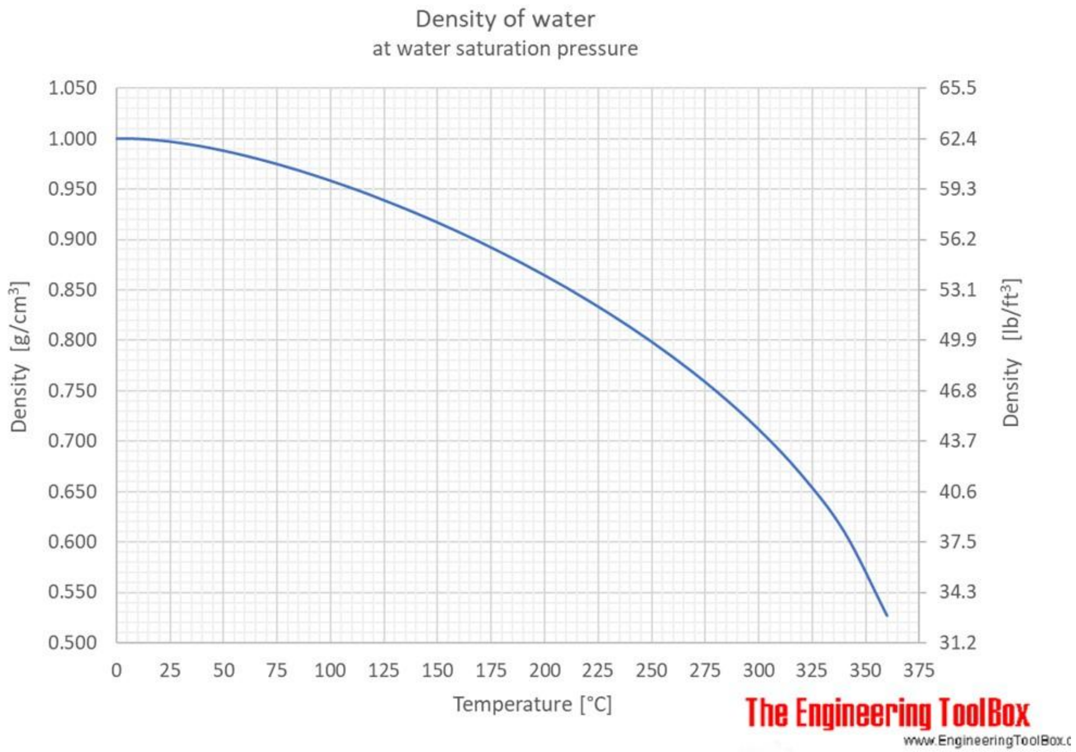
kg/m^3

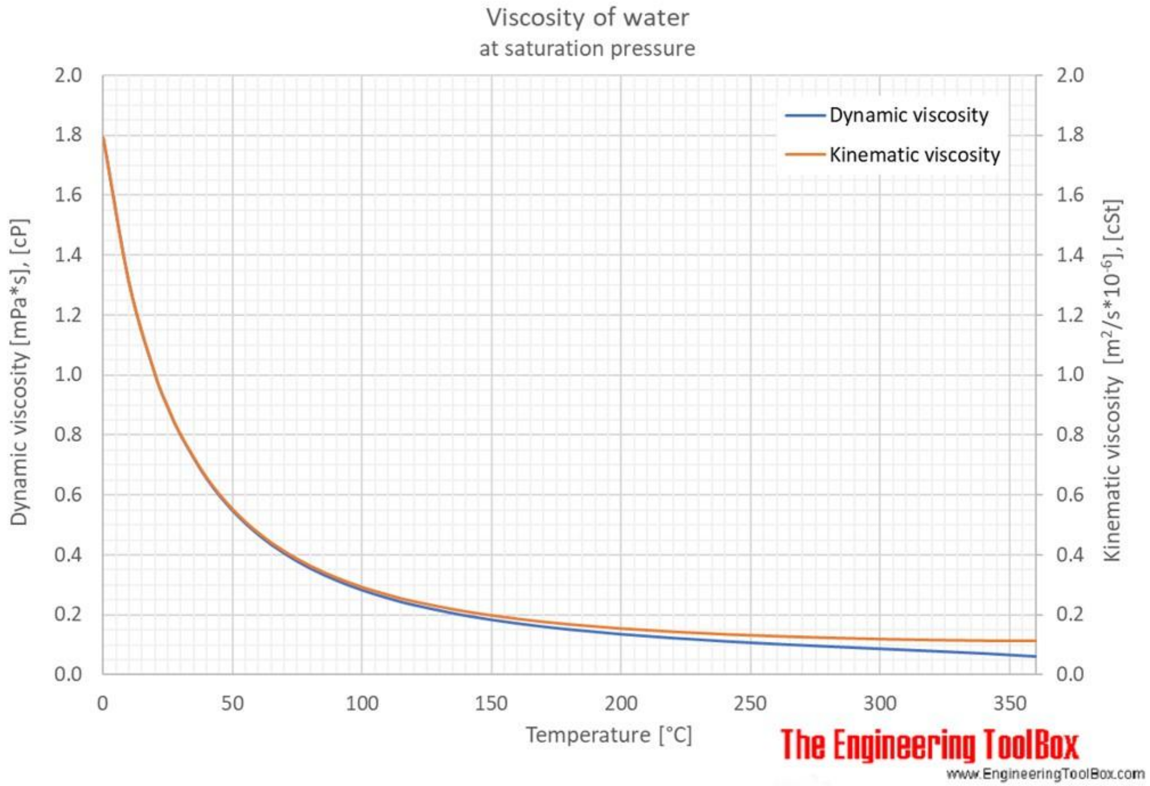
mm

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.

Property of Water	0° C	20° C	40° C	60° C	80° C	100° C	Units
Density	999.84	998.21	992.22	983.20	971.82	958.40	kg m^{-3}
Thermal Expansion	-0.07	0.207	0.385	0.523	0.643	0.752	$*10^{-3}$ K^{-1}
Isothermal Compression (Volume Viscosity)	5.0879	4.5895	4.4241	4.4507	4.6418	4.9015	$*10^{-10}$ Pa^{-1}
Dynamic Viscosity	1.793	1.002	0.6532	0.4665	0.3544	0.2818	$*10^{-3}$ kg m^{-1} s^{-1} (Pa s)
Kinematic Viscosity	1.787	1.004	0.658	0.475	0.365	0.294	$*10^{-6}$ m^2 s^{-1}
Thermal Conductivity	561.0	598.4	630.5	654.3	670.0	679.1	$*10^{-3}$ W m^{-1} K^{-1}
Specific Heat at constant pressure C_p	4.2176	4.1818	4.1785	4.1843	4.1963	4.2159	$*10^3$ J kg^{-1} K^{-1}
Specific Heat at constant volume C_v							$*10^3$ J kg^{-1} K^{-1}
Specific Entropy e	0	0.296	0.581	0.832	1.076	1.307	$*10^3$ J kg^{-1} K^{-1}
Specific Enthalpy	0	83.8	167.6	251.5	335.3	419.1	$*10^3$ J kg^{-1}
Saturation Vapor Pressure	611.3	2,338.8	7,381.4	19,932	47,373	101,325	Pa
Surface Tension	75.64	72.75	69.60	66.24	62.47	58.91	$*10^{-3}$ N m^{-1}
Speed of Sound	1,403	1,481	1,526	1,552	1,555	1,543	m s^{-1}

[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)

$$f = \cdots$$

Continuous pressure losses

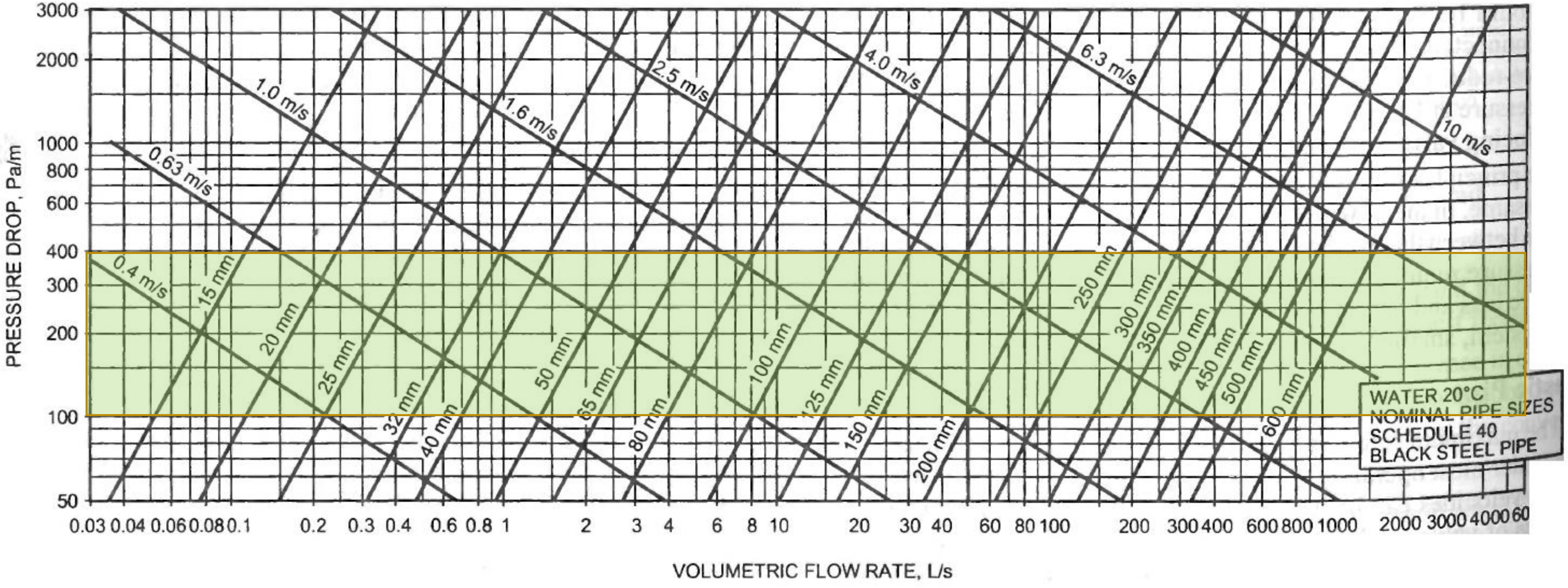


Fig. 14 Friction Loss for Water in Commercial Steel Pipe (Schedule 40)

[Source: ASHRAE Handbooks]

Continuous pressure losses

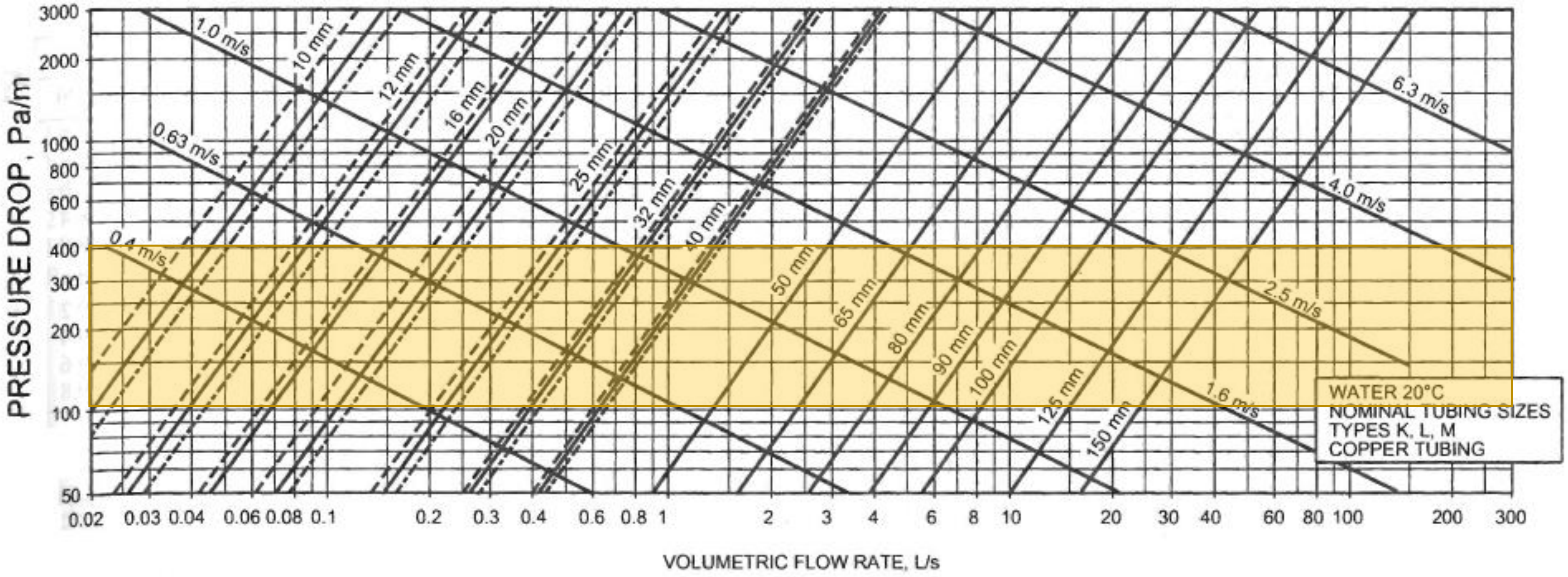


Fig. 15 Friction Loss for Water in Copper Tubing (Types K, L, M)

[Source: ASHRAE Handbooks]

Continuous pressure losses

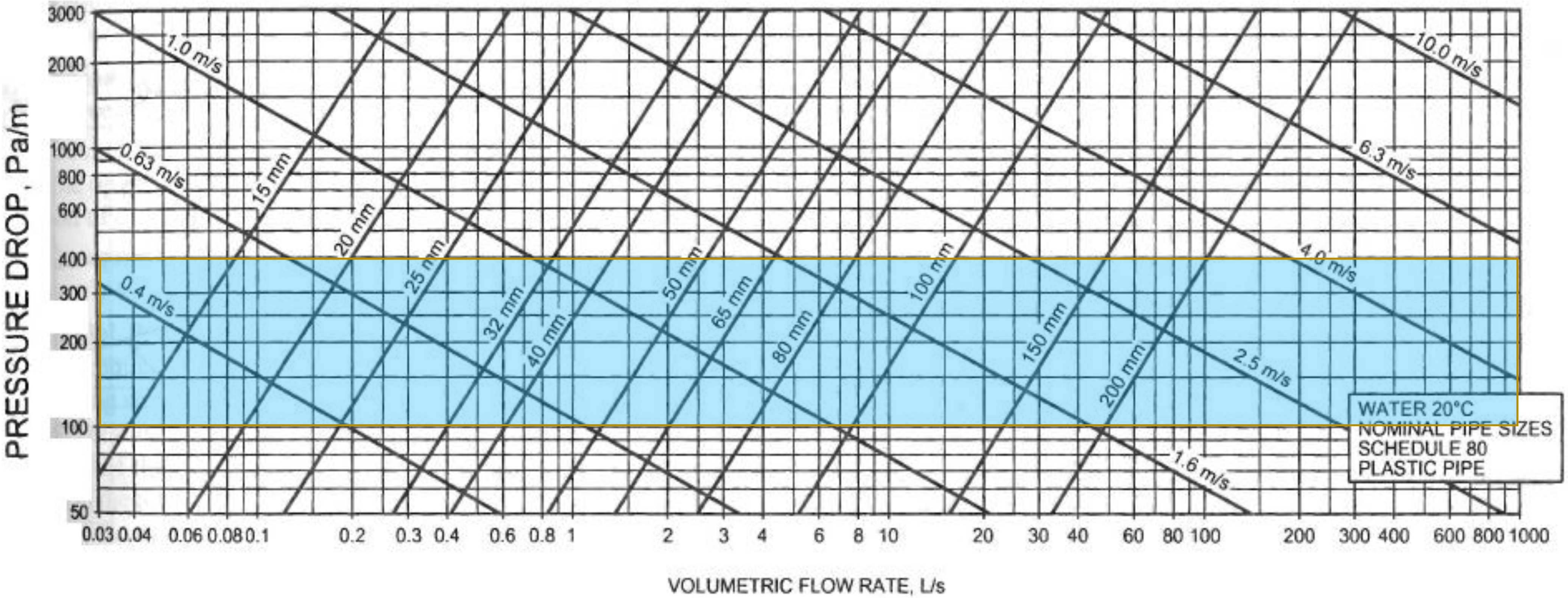


Fig. 16 Friction Loss for Water in Plastic Pipe (Schedule 80)

[Source: ASHRAE Handbooks]

Localized pressure losses

Loss coefficients for valves

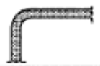
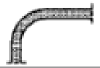
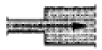
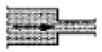
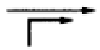
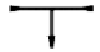
[source: *M. Doninelli, Design principles of hydronic heating systems, Caleffi Handbooks*]

Internal diameter copper tube, PEad, PEX		8÷16 mm	18÷28 mm	30÷54 mm	>54 mm
External diameter steel tube		3/8"÷1/2"	3/4"÷1"	1 1/4"÷2"	>2"
Localised loss type	Symbol				
Shut-off valve		10,0	8,0	7,0	6,0
Shut-off valve		5,0	4,0	3,0	3,0
Reduced passage gate valve		1,2	1,0	0,8	0,6
Total passage gate valve		0,2	0,2	0,1	0,1
Reduced passage ball valve		1,6	1,0	0,8	0,6
Total passage ball valve		0,2	0,2	0,1	0,1
Butterfly valve		3,5	2,0	1,5	1,0
Check valve		3,0	2,0	1,0	1,0
Radiator valve		8,5	7,0	6,0	—
Radiator valve		4,0	4,0	3,0	—
Lockshield		1,5	1,5	1,0	—
Lockshield		1,0	1,0	0,5	—
4-way valve		6,0		4,0	
3-way valve		10,0		8,0	
Passage through radiator		3,0			
Passage through boiler		3,0			

Localized pressure losses

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

[source: *M. Doninelli, Design principles of hydronic heating systems, Caleffi Handbooks*]

		Internal diameter copper tube, PEad, PEX	8÷16 mm	18÷28 mm	30÷54 mm	>54 mm
		External diameter steel tube	3/8"÷1/2"	3/4"÷1"	1 1/4"÷2"	>2"
Localised loss type		Symbol				
Narrow bend 90°	r/d = 1,5		2,0	1,5	1,0	0,8
Normal bend 90°	r/d = 2,5		1,5	1,0	0,5	0,4
Wide bend 90°	r/d > 3,5		1,0	0,5	0,3	0,3
Narrow bend U	r/d = 1,5		2,5	2,0	1,5	1,0
Normal bend U	r/d = 2,5		2,0	1,5	0,8	0,5
Wide bend U	r/d > 3,5		1,5	0,8	0,4	0,4
Section change			1,0			
Section change			0,5			
T joint			1,0			
T joint			1,0			
T joint			3,0			
T joint			3,0			
Angle joint (45°- 60°)			0,5			
Angle joint (45°- 60°)			0,5			
Bend joint			2,0			
Bend joint			2,0			

Localized pressure losses

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

<table>
<tr><th rowspan="2"></th><th colspan="13">Diametro nominale (DN)</th></tr>
<tr><th>10</th><th>15</th><th>20</th><th>25</th><th>32</th><th>40</th><th>50</th><th>65</th><th>80</th><th>100</th><th>125</th><th>150</th><th>200</th></tr>
<tr><td>Curva arrotondata (raggio curv./diam. = 1,5)</td><td colspan="13">← 0,5 →</td></tr>
<tr><td>Gomito</td><td>2</td><td>2</td><td>1,5</td><td>1,5</td><td colspan="9">← 1 →</td></tr>
<tr><td>Brusco allargamento (sbocco)</td><td colspan="13">← 1÷1,2 →</td></tr>
<tr><td>Brusca restrizione (imbocco)</td><td colspan="13">← 0,5÷0,7 →</td></tr>
<tr><td>Radiatore</td><td>7</td><td>4,5</td><td>3,5</td><td>2,5</td><td colspan="9"></td></tr>
<tr><td>Valvola per radiatore (o detentore)</td><td>17</td><td>10</td><td>9</td><td>8</td><td colspan="9"></td></tr>
<tr><td>Caldaia</td><td colspan="13">← 3 →</td></tr>
<tr><td>Valvola a sfera (passaggio totale) o saracinesca</td><td colspan="13">← 0,5 →</td></tr>
<tr><td>Valvola a sfera (passaggio ridotto)</td><td colspan="13">← 1 →</td></tr>
<tr><td>T diretto</td><td>0,8</td><td>0,7</td><td>0,6</td><td>0,5</td><td>0,4</td><td>0,3</td><td>0,3</td><td>0,3</td><td>0,2</td><td>0,2</td><td>0,2</td><td>0,2</td><td>0,1</td></tr>
<tr><td>T deviato</td><td>2,5</td><td>2</td><td>1,5</td><td>1,5</td><td>1,5</td><td>1,5</td><td>1,4</td><td>1,4</td><td>1,4</td><td>1,3</td><td>1,3</td><td>1,2</td><td>1,2</td></tr>
<tr><td>T confluente dir</td><td>0,8</td><td>0,7</td><td>0,6</td><td>0,5</td><td>0,4</td><td>0,3</td><td>0,3</td><td>0,3</td><td>0,2</td><td>0,2</td><td>0,2</td><td>0,1</td><td>0,1</td></tr>
<tr><td>T confluente lat.</td><td>1,7</td><td>1,3</td><td>1</td><td>1</td><td>1</td><td>1</td><td>1</td><td>1</td><td>0,9</td><td>0,9</td><td>0,9</td><td>0,8</td><td>0,8</td></tr>
<tr><td>Valvola di ritegno a clapet</td><td colspan="13">← 4 →</td></tr>
<tr><td>Filtro a “Y” (pulito)</td><td colspan="13">← 4÷5 →</td></tr>
<tr><td>Valvola di ritegno a disco a molla morbida</td><td colspan="13">← 10÷12 →</td></tr>
</table>

[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 leak into the circuit due to reduced pressures.

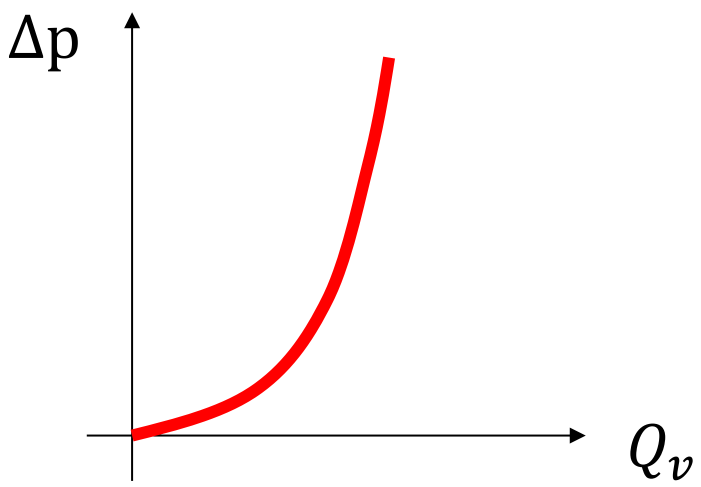
Pipe design

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

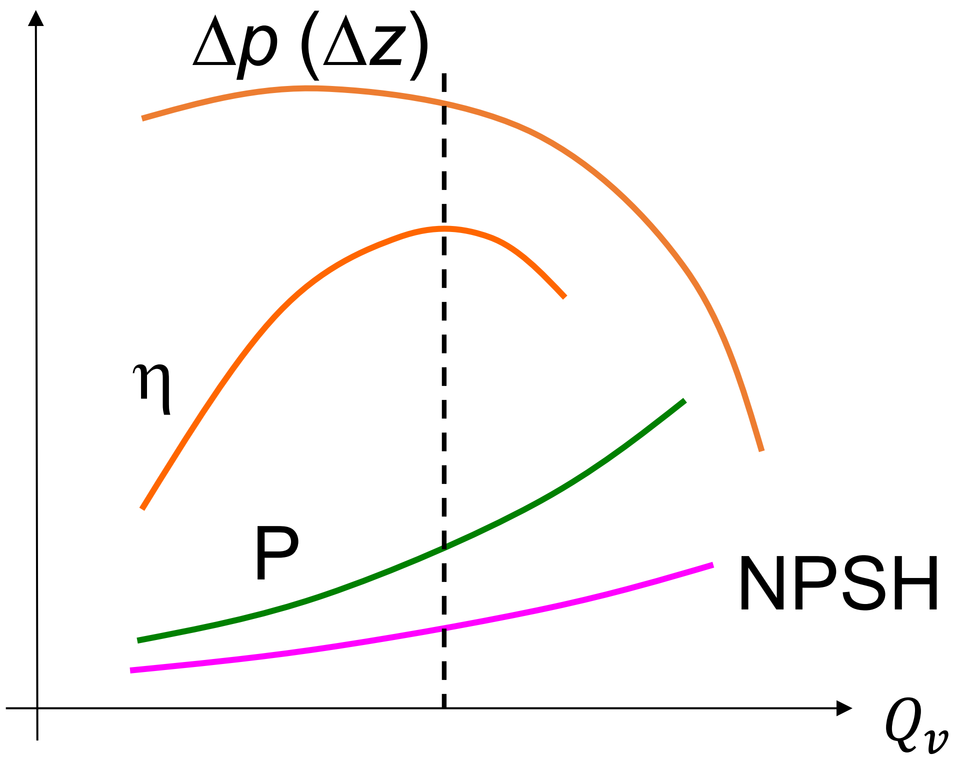
Characteristic curve of the circuit



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

Pump selection

Characteristic (performance) 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

Characteristic (performance) curves of the pump

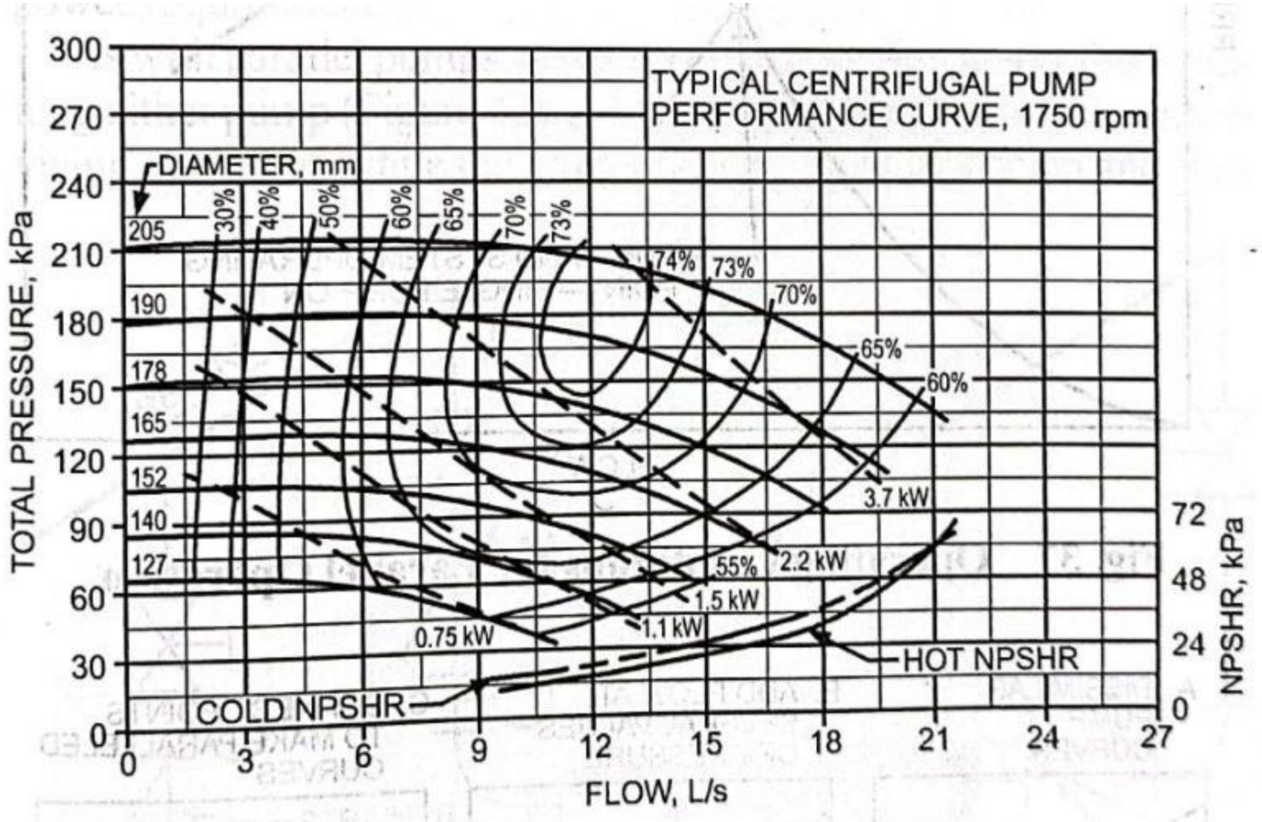
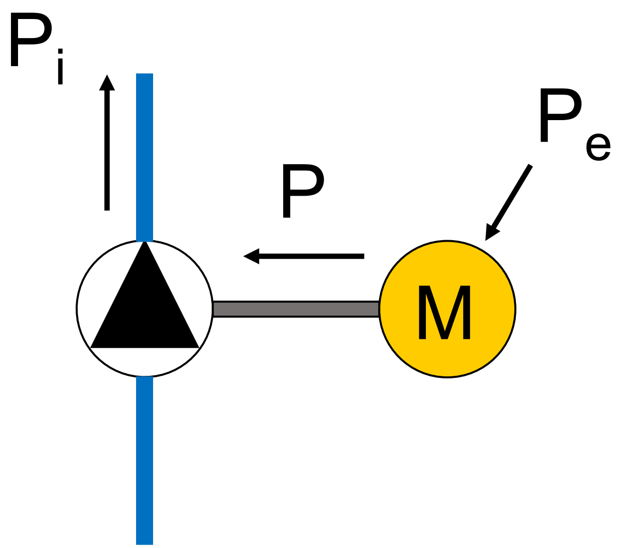


Fig. 34 Pump Performance and NPSR Curves

Pump selection

Efficiency of the pump



Hydraulic efficiency $\eta = \dfrac{P_i}{P}$

Electric efficiency $\eta_e = \dfrac{P}{P_e}$

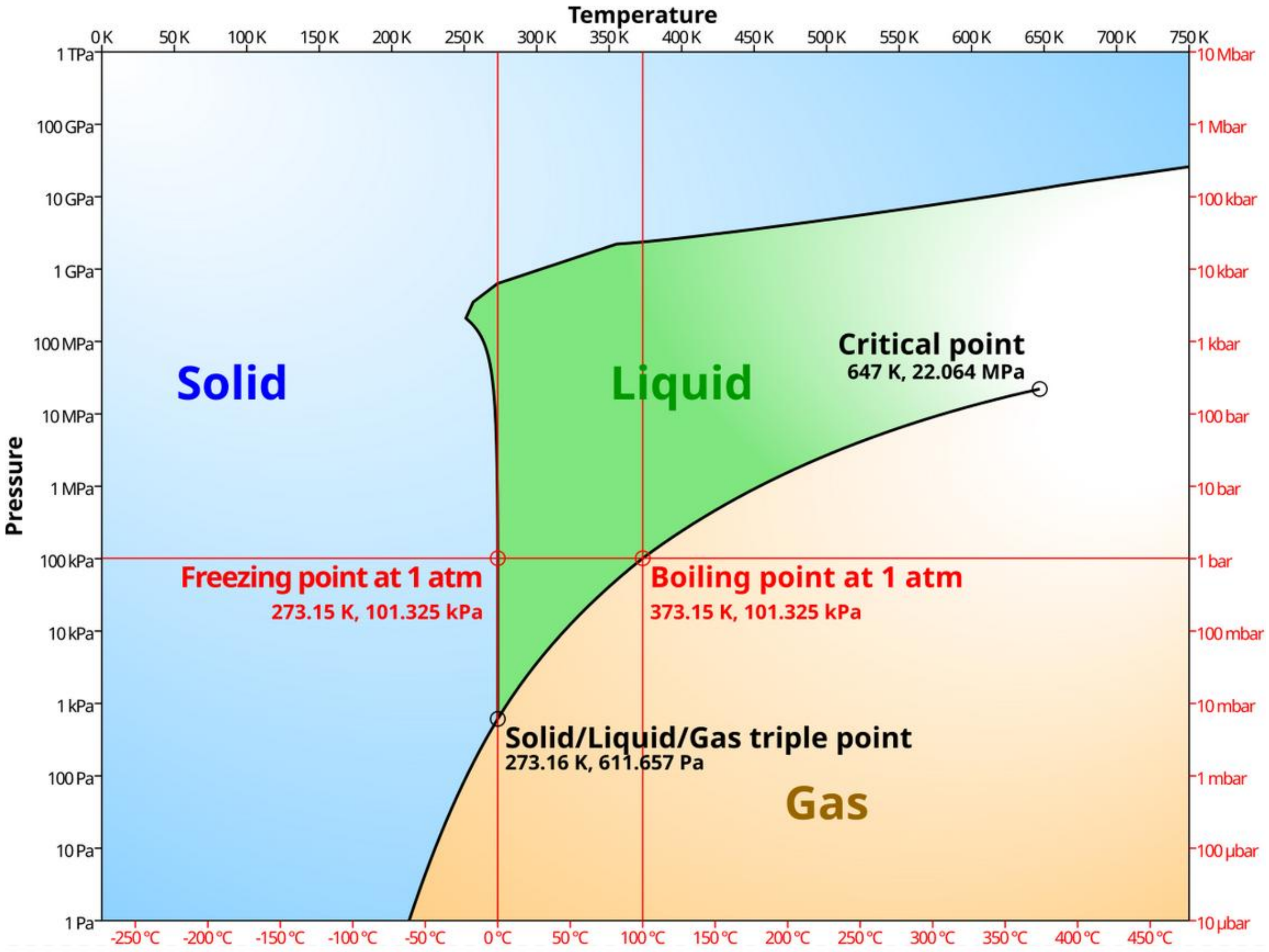
Overall efficiency $\eta_g = \eta \cdot \eta_e = \dfrac{P_i}{P_e}$

Pump selection

Cavitation

Cavitation is the phenomenon in which the static pressure of a liquid reduces to below the liquid's vapor pressure, leading to the formation of small vapor-filled cavities in the liquid.

When subjected to higher pressure, these cavities (bubbles), collapse and can generate shock waves that may damage machinery.



Pump selection

Net Positive Suction Head (NPSH)

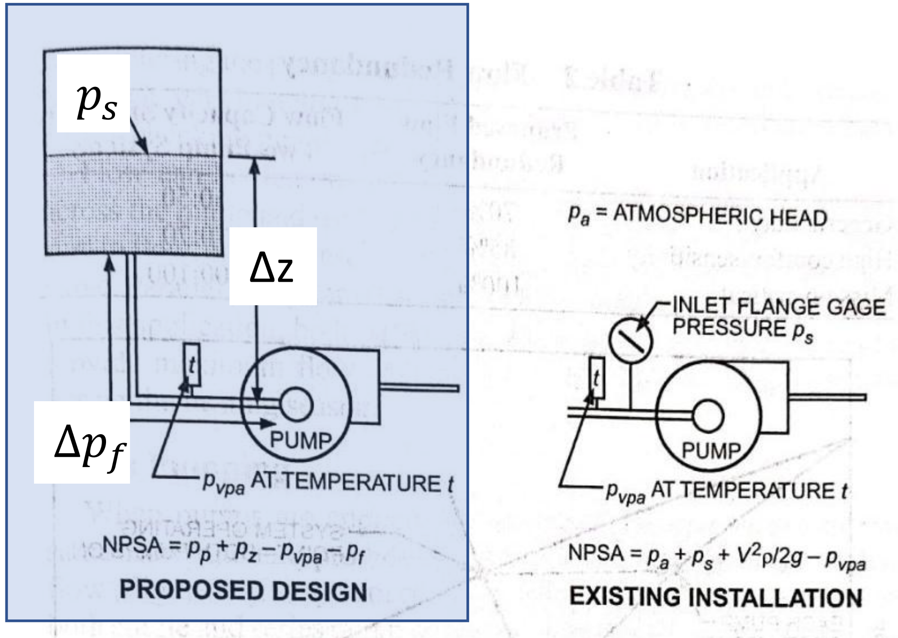


Fig. 33 Net Positive Suction Pressure Available

[source: 2020 *ASHRAE Handbook – HVAC systems and Equipment*]

$$\boldsymbol{NPSH_a \geq NPSH_r}$$

$$NPSH_a = p_s + \rho\, g\, \Delta z - \Delta p_f - p_v(T)$$

Pump selection

Net Positive Suction Head (NPSH)

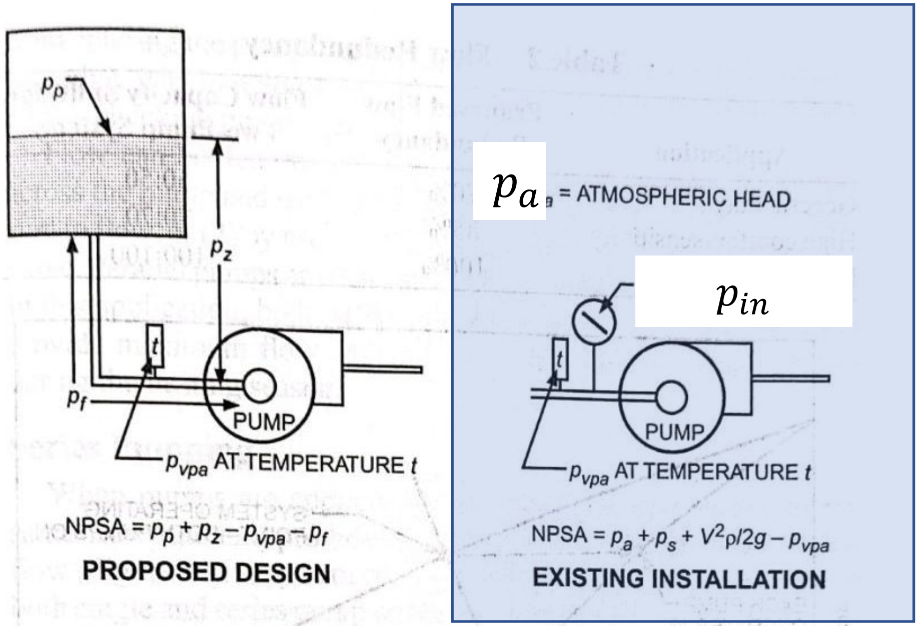


Fig. 33 Net Positive Suction Pressure Available

[source: 2020 *ASHRAE Handbook – HVAC systems and Equipment*]

$$\boldsymbol{NPSH_a \geq NPSH_r}$$

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

Pump selection

Affinity laws

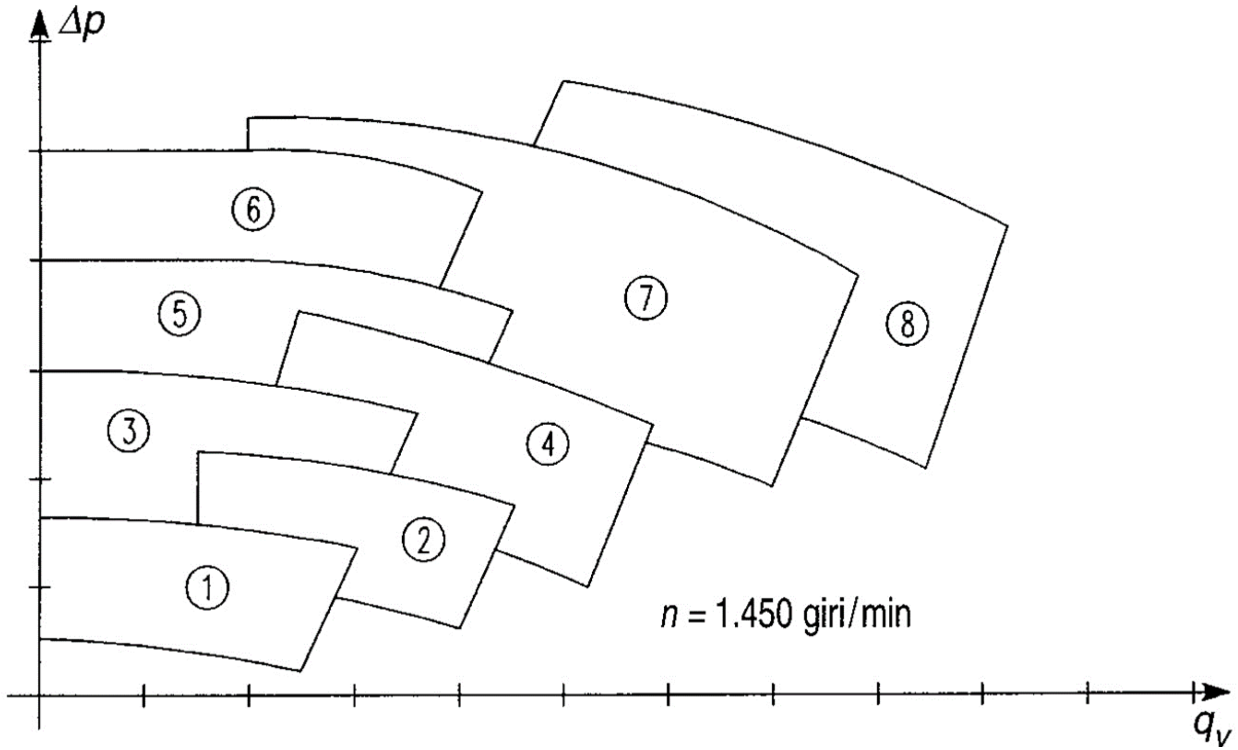
	Speed change (same diameter)	Impeller diameter change (same speed)
Flow	$Q_{v2} = Q_{v1}\left(\frac{n_2}{n_1}\right)$	$Q_{v2} = Q_{v1}\left(\frac{D_2}{D_1}\right)$
Pressure head	$\Delta p_2 = \Delta p_1\left(\frac{n_2}{n_1}\right)^2$	$\Delta p_2 = \Delta p_1\left(\frac{D_2}{D_1}\right)^2$
Power	$P_2 = P_1\left(\frac{n_2}{n_1}\right)^3$	$P_2 = P_1\left(\frac{D_2}{D_1}\right)^3$
Efficiency	$\eta_1 = \eta_2$	$\eta_1 \cong \eta_2$*

Pump selection

Affinity laws

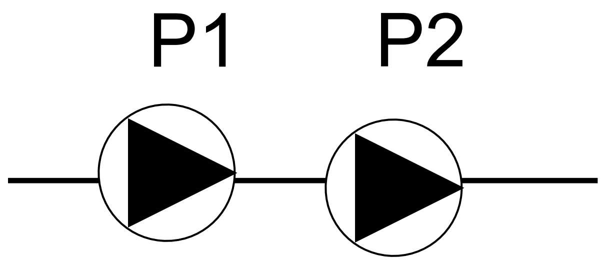
$$\frac{Q_{v1}}{Q_{v2}} = \frac{D_1}{D_2}$$

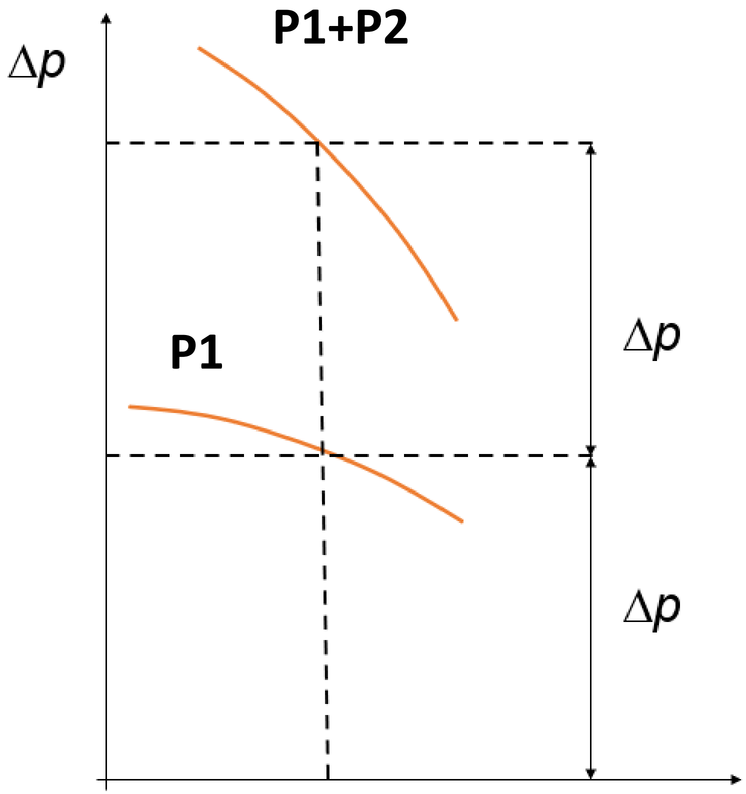
$$\frac{\Delta p_1}{\Delta p_2} = \left(\frac{D_1}{D_2}\right)^2$$



Pump selection

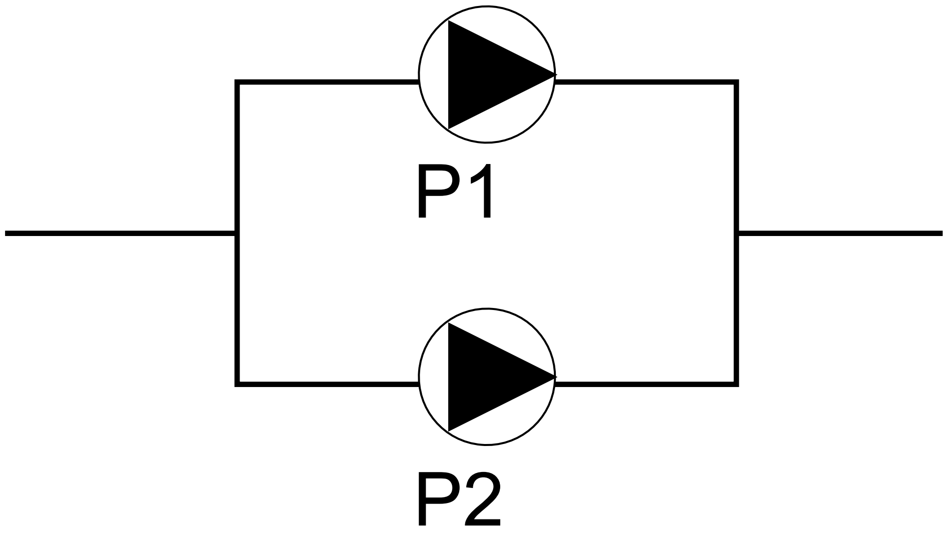
Pumps in series

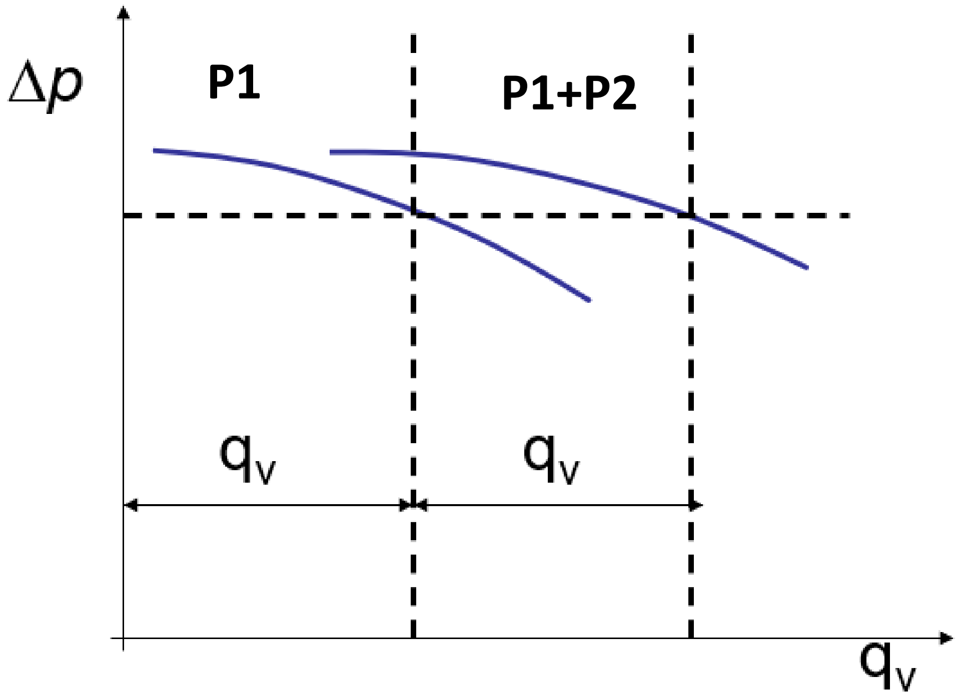




Pump selection

Pumps in parallel





Pump selection

Working point

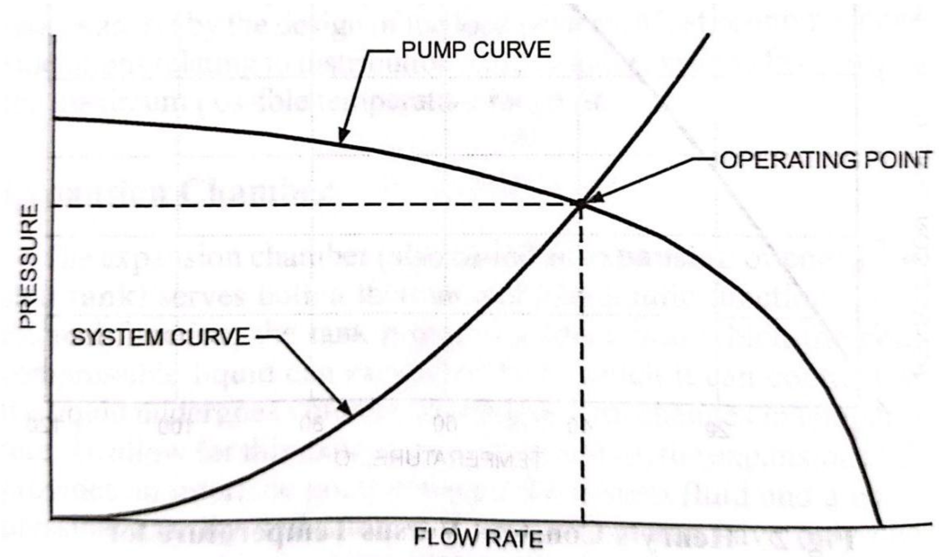
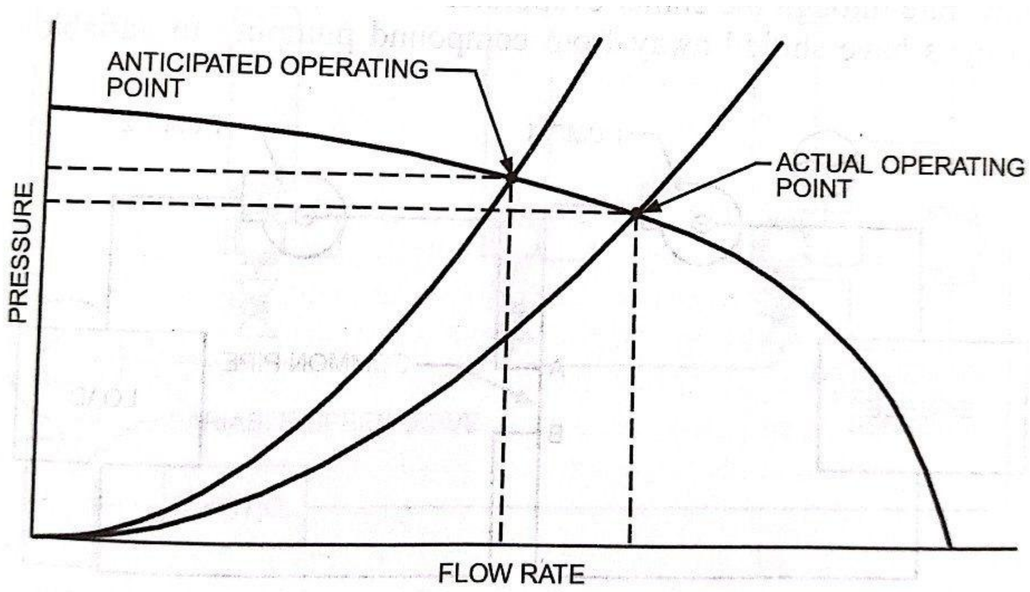


Fig. 5 Pump Curve and System Curve



Pump selection

Working point

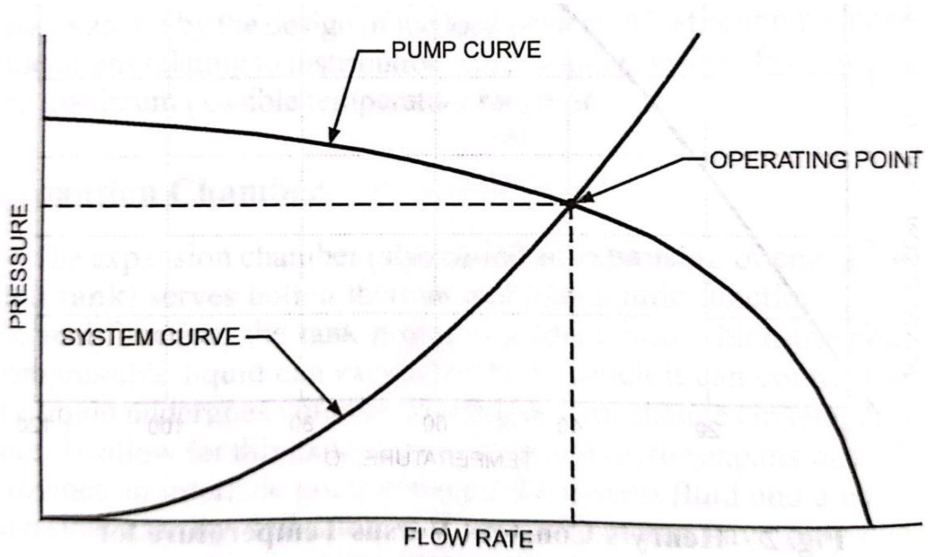


Fig. 5 Pump Curve and System Curve

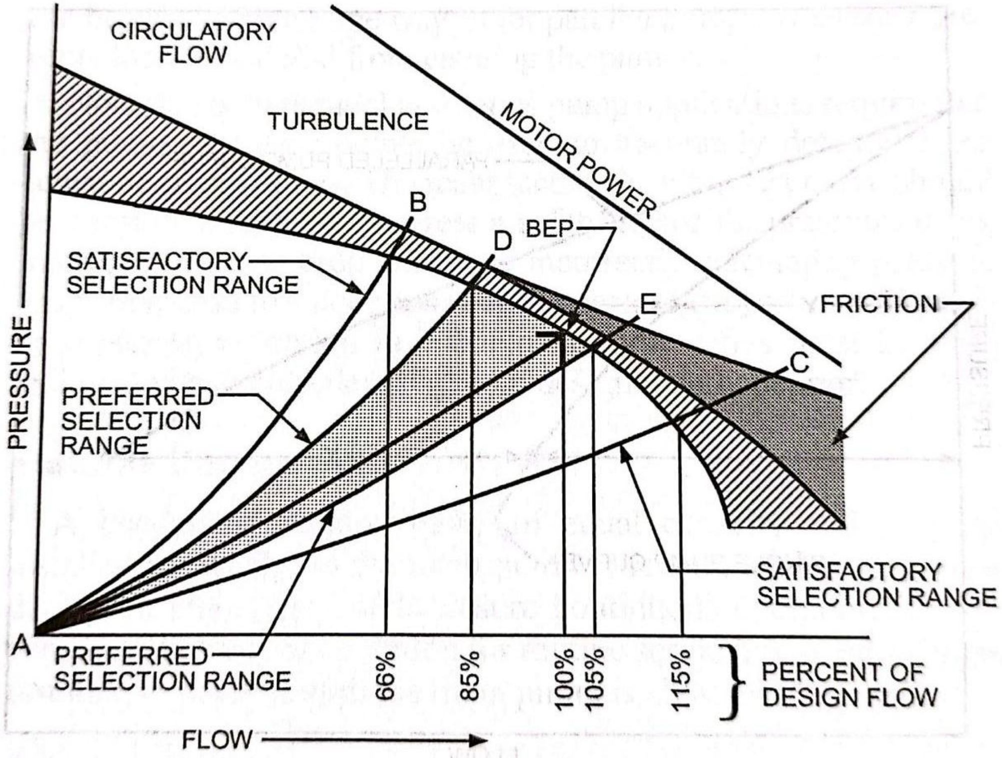


Fig. 35 Pump Selection Regions