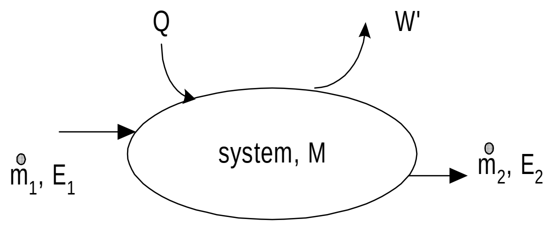
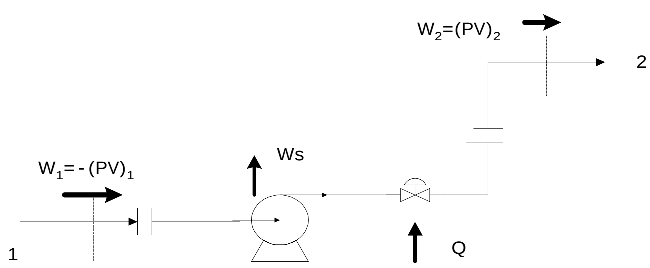
Chapter 1

MATERIAL TRANSPORT

1.1 Fundamental Relationships

1.1.1 Energy Balances





Let the *system* of concern for balance equations be a lumped parameter system surrounded by a fixed boundary. The energy balance around the system is represented by

$$\dot{m}_1\bar{E}_1 - \dot{m}_2\bar{E}_2 + Q - W' = \frac{d}{dt}\left(\bar{E}M\right)$$

where $\dot{m}$ is the mass flow rate; $\bar{E}$ represents the energy content per unit mass of the fluid: Q and W' represent heat flow from the surrounding and work delivered on the surrounding by the system per unit time.

If only the internal energy, kinetic energy, and potential energy (by gravity) are concerned for the energy of the fluid,

$$\bar{E} = \bar{U} + \frac{v^2}{2\alpha g_c} + \frac{g}{g_c} z$$

$$\text{where} \quad v = q/S = \text{ mixing cup average velocity}, \quad \alpha = \begin{cases} 1 & \text{for turbulent flow} \\ 0.5 & \text{for laminar flow} \end{cases}$$

Rate of work done on the surrounding W' are composed of two terms, shaft work (W_s) plus PV work of the system on the surrounding at the inlet and outlet of the flow.

$$W' = W_s + \dot{m}_2 (P\bar{V})_2 - \dot{m}_1 (P\bar{V})_1 = W_s + \dot{m}_2 \left(\frac{P}{\rho}\right)_2 - \dot{m}_1 \left(\frac{P}{\rho}\right)_1$$

where $\bar{V}$ denotes the specific volume of the fluid and P_1 and P_2 are opposing forces against the system at the respective points.

Q: Physical meaning of the PV work ?

Summarizing the above with the use of the definition $\bar{H} = \bar{U} + P\bar{V}$ gives

$$\underline{\frac{d}{dt}(\bar{U}M) = \dot{m}_1 \left(\bar{H} + \frac{g}{g_c} z + \frac{v^2}{2\alpha g_c}\right)_1 - \dot{m}_2 \left(\bar{H} + \frac{g}{g_c} z + \frac{v^2}{2\alpha g_c}\right)_2 + Q - W_s}$$

Rate of change of the KE and PE of the system are neglected by assumption (fixed and no motion).

- For **closed systems**, $\dot{m} = 0$ and $M=$ constant. Hence

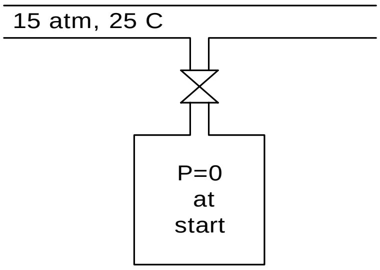
$$\underline{Q} - \underline{W}_s = \frac{d}{dt}\bar{U} \quad \text{where } \underline{Q} \triangleq Q/M \text{ and } \underline{W}_s = W_s/M.$$

- For **open systems** at steady state ($\dot{m} = \dot{m}_1 = \dot{m}_2$),

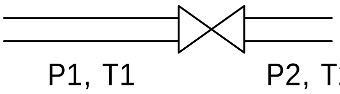
$$\bar{H}_1 + \frac{g}{g_c} z_1 + \frac{v_1^2}{2\alpha_1 g_c} + \bar{Q} - \bar{W}_s = \bar{H}_2 + \frac{g}{g_c} z_2 + \frac{v_2^2}{2\alpha_2 g_c}$$

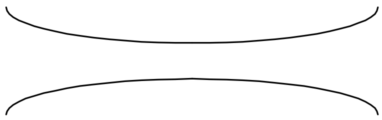
where $\bar{Q} \triangleq Q/\dot{m}$ and $\bar{W}_s = W_s/\dot{m}$.

Q: A well-known example from thermodynamics



Q: Energy balance around a throttling device and a converging-diverging nozzle



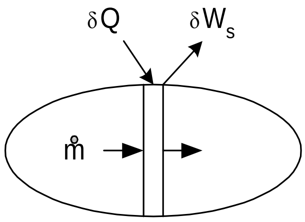


The required (ideal) pumping energy (per unit mass flow of the fluid) is

$$-\bar{W}_s = \frac{g}{g_c}(z_2 - z_1) + \frac{1}{2g_c}\left(\frac{v_2^2}{\alpha_2} - \frac{v_2^1}{\alpha_1}\right) + (\bar{H}_2 - \bar{H}_1) - \bar{Q}$$

- useful for design and analysis of gas transportation systems.

• Conversion of the open-system steady state balance to the **mechanical energy form**



Entropy balance around the differential volume

$$\dot{m}d\bar{S} = \frac{\delta LW(\text{lost work})}{T} + \frac{\delta Q}{T} \;\Rightarrow\; \delta Q = \dot{m}Td\bar{S} - \delta LW$$

Energy balance around the differential volume

$$\dot{m}d\left(\bar{H} + \frac{g}{g_c}z + \frac{v^2}{2\alpha g_c}\right) = \delta Q - \delta W_s \;\Rightarrow\; \dot{m}d\left(\bar{H} + \frac{g}{g_c}z + \frac{v^2}{2\alpha g_c}\right) = \dot{m}Td\bar{S} - \delta LW - \delta W_s$$

$\bar{S}$ is cancelled using $d\bar{H} = Td\bar{S} + \bar{V}dP$, we have

$$\bar{V}dP + \frac{g}{g_c}dz + d\frac{v^2}{2\alpha g_c} = -\delta L\bar{W} - \delta\bar{W}_s \quad \text{where} \quad \delta L\bar{W} = \delta LW/\dot{m} \quad \delta\bar{W}_s = \delta W_s/\dot{m}$$

Integrating over the volume yields

$$\frac{g}{g_c}z_1 + \frac{v_1^2}{2\alpha g_c} - \bar{W}_s = \frac{g}{g_c}z_2 + \frac{v_2^2}{2\alpha g_c} + \underbrace{\sum F_i}_{\int \delta LW} + \int_1^2 \frac{1}{\rho(P)}dP \quad \text{[energy/mass]}$$

where F_i is the mechanical energy loss due to friction.

For incompressible fluids,

$$\frac{g}{g_c}z_1 + \frac{v_1^2}{2\alpha g_c} + \frac{p_1}{\rho} - \bar{W}_s = \frac{g}{g_c}z_2 + \frac{v_2^2}{2\alpha g_c} + \sum F_i + \frac{p_2}{\rho}$$

The required pumping energy is

$$-\bar{W}_s = \frac{g}{g_c}(z_2 - z_1) + \frac{v_2^2 - v_1^2}{2\alpha g_c} + \frac{p_2 - p_1}{\rho} + \sum F_i$$

- used for piping design for liquid transportation

1.1.2 Friction Factor

- Fanning equation:

$$F = f\frac{2v^2L}{g_cD}$$

- Friction factor f ?
  - Pipes and tubes : Refer to Fig. 14-1 in pp. 482.
    - Drawn tube - tubes drawn through die casting, usually has small diameter (< 0.5").
    - Commercial steel, wrought iron - ordinary process pipes
    - Asphalted cast iron - external is coated with asphalt. used for piping in sea water or other corrosive environment.

    * galvanized iron - zinc coating, used for utility and underground piping

  In practice, 10% to 15% of allowance is given to the required pump horse power to account for roughness that will be developed by corrosion.

- Valves and fittings: Normally use special formulas or the concept of *equivalent length*.

$$F = f\frac{2v^2 L_e}{g_c D}$$

  Refer to Table 1 in pp 484-485.

1.2 Piping Practice

1.2.1 Piping Standards

◇ **Pipe**

- Used for process and utility piping.
- Size(OD) is referred by the nominal dimension.
  For example, 1" pipe has 1.32" OD.
  Refer to Table 13 in pp. 888.
- Pipe strength ($\propto$ thickness/diameter) is referred by the schedule number.
  - Schedule number = $1000P_s/S_s = 2000t_m/D_m$ where

$$\begin{aligned} P_s &= \text{safe working pressure (psi)}, \quad S_s = \text{tensile strength (psi)} \\ t_m &= \text{mean thickness}, \quad D_m = \text{mean diameter} \end{aligned}$$

  Schedule 40 and 80 are usually referred as "standard" and "extra-strong".

◇ **Tube**

- Used for s, heat exchangers, instrument air, $\cdots$
- Nominal size is the true OD. Refer to Table 12 in pp. 886.
- Thickness is defined by BWG (Birmingham Wire Gauge) number.

◇ Fittings

- Fittings - union, cross, tee, elbow, 45^o elbow
- Thereaded fittings - used for small pipes
- Flanges - used for 3” or larger pipes
- Screwed fittings - used for smaller pipes or tubes
- Bell-and-spigot joint - for underground piping

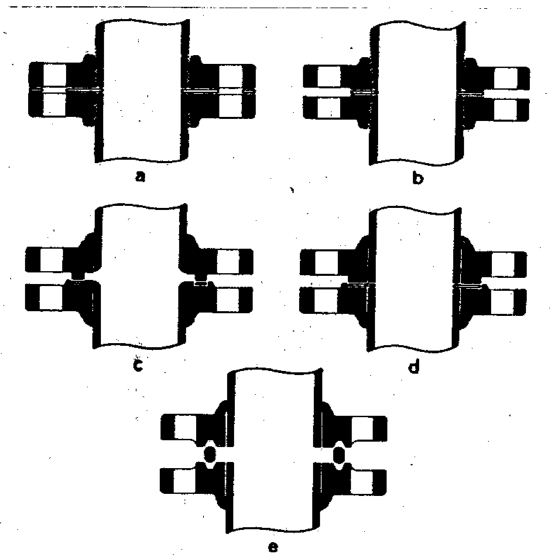


Fig. 3-3. Methods of facing flanges: (*a*) plain face; (*b*) raised face; (*c*) tongue-and-groove face; (*d*) male and female face; (*e*) ring joint.

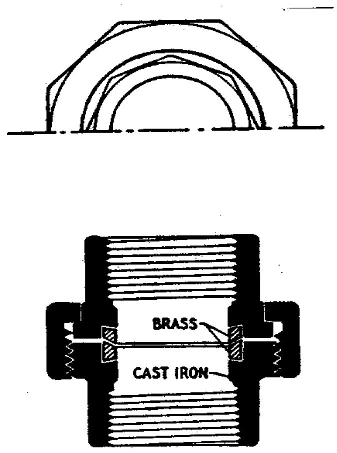


Fig. 3-2. Union.

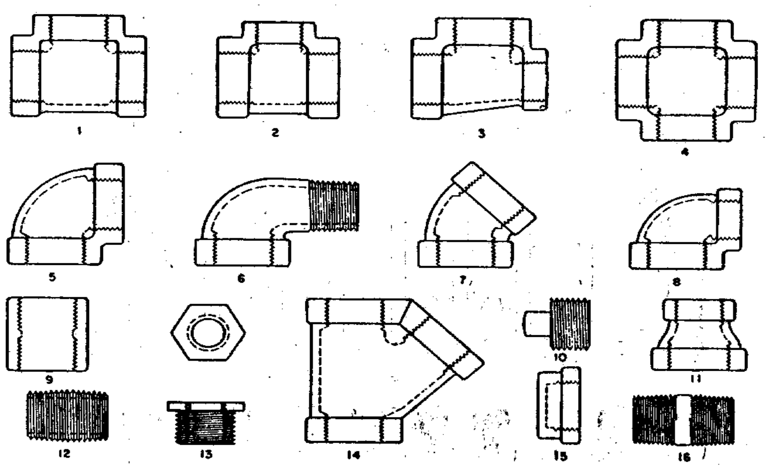


Fig. 3-7. Screwed pipe fittings: (1) tee; (2) tee reducing on outlet; (3) tee reducing on run; (4) cross; (5) elbow; (6) street elbow; (7) 45° elbow; (8) reducing elbow; (9) coupling; (10) plug; (11) reducer; (12) close nipple; (13) bushing; (14) Y branch; (15) cap; (16) short nipple.

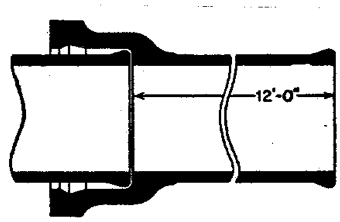


Fig. 3-1. Bell-and-spigot joint for cast-iron pipe.

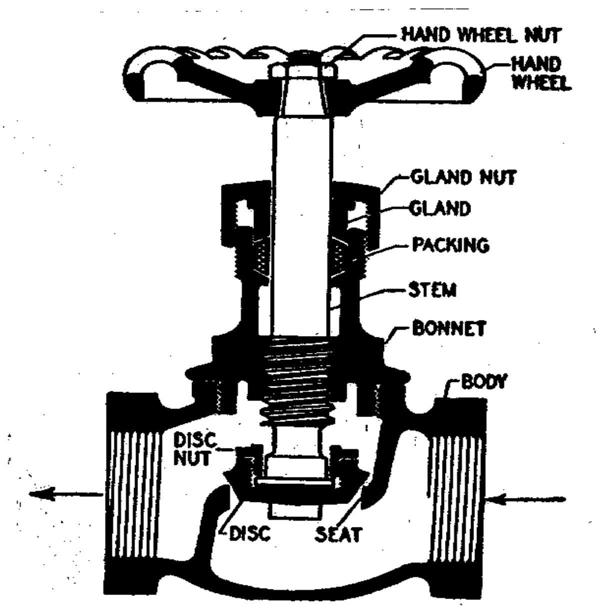


Fig. 3-10. Globe valve.

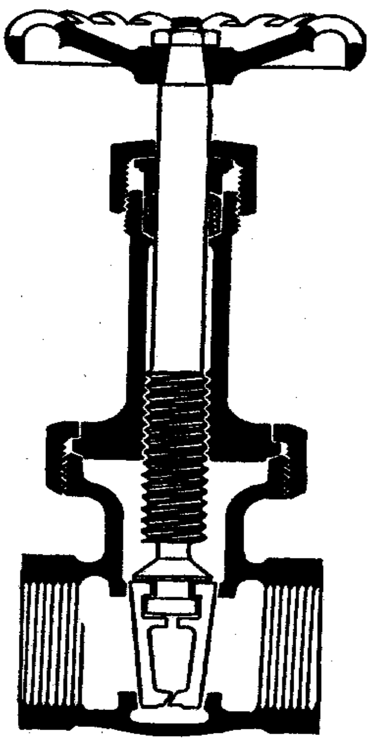


Fig. 3-12. Rising-stem gate valve.

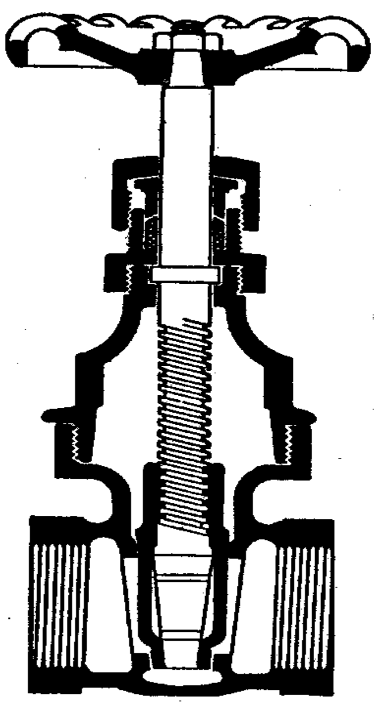
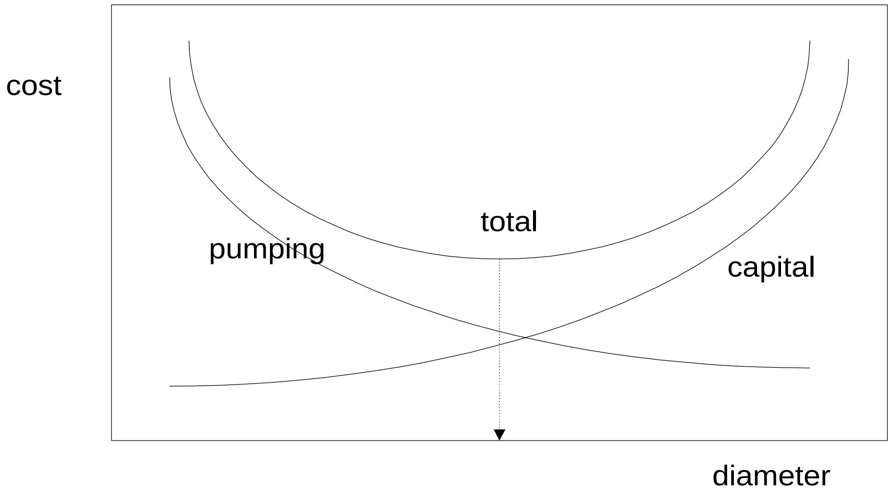


Fig. 3-13. Nonrising-stem gate valve.

1.2.2 Pipe Sizing

- need to be optimized between the pumping cost by friction loss and the capital cost.



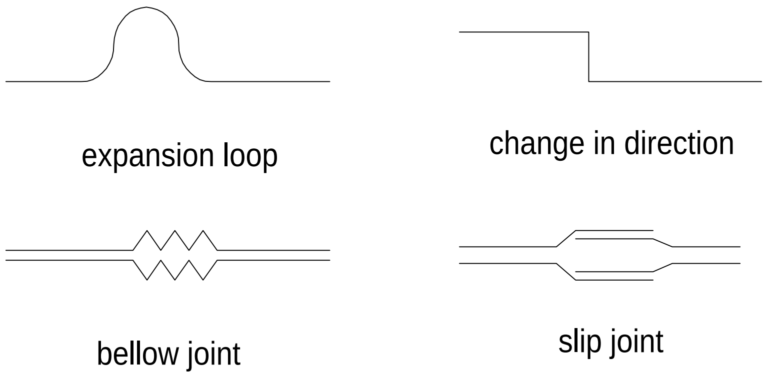
- Usual Guide

Type of fluid	Reasonable velocity (ft/s)
Normal fluid	3-10
Lo-pressure steam(25 psig)	50-100
Hi-pressure steam(100 psig or up)	100-200
Air at ordinary pressure(25-50 psig)	50-100

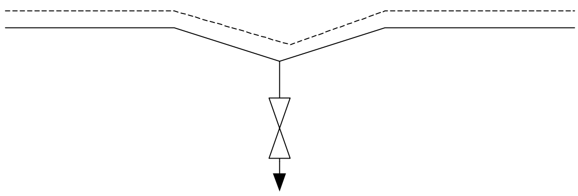
- For estimation of optimum pipe diameter, see Fig. 14-2.

1.2.3 Considerations in Piping Design

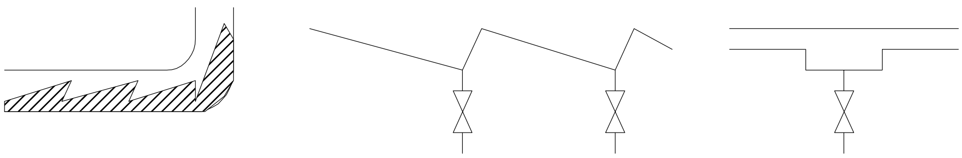
- Effects of temperature level and temperature change
  - Insulation
  - thermal expansion
    $\Delta L = 5''$ in $100ft$ steel pipe for $\Delta T = 300C$
    $\Delta L = 7''$ in $100ft$ steel brass for $\Delta T = 300C$



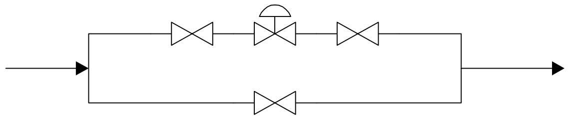
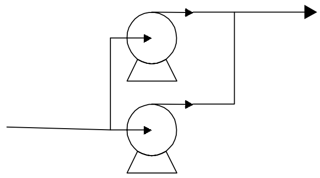
– Freezing, solidification – insulation, steam tracing, sloping to drain valve



- Flexibility for physical and thermal shock
  Drain condensate for prevention of water hammering



- adequate support and anchor
- Alterations in the system and the service
- Maintenance and inspection
- Ease of installation
- Auxiliary or stand-by pumps and lines

- Safety - design allowance, relief valves, rupture disks, flare systems $\cdots$

1.2.4 Costs for Piping and Piping-System Auxiliaries

Refer to Fig. 14-3 through Fig. 14-34.

1.3 Pumps, Liquid Transportation

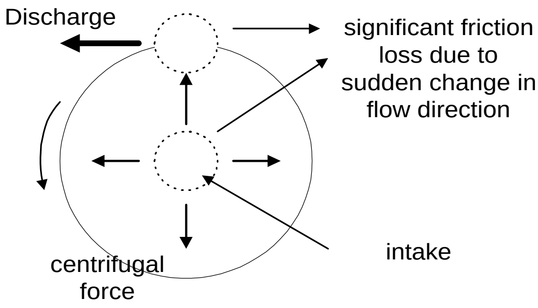
♢ Factors that Influence Pump Selection

- Flow rate
- Pressure (head)
- Fluid properties : density, viscosity, slurry, corrosion, solvent, $\cdots$
- Type of flow distribution : pulsating, nonpulsating
- Type of power supply : electric, steam, air, hydraulic
- Cost and mechanical efficiency

1.3.1 Rotary Centrifugal Pumps

- Fluid transportation by centrifugal force
- Cannot be used gases or vapor
- Most widely used for liquid transportation

◇ Principle

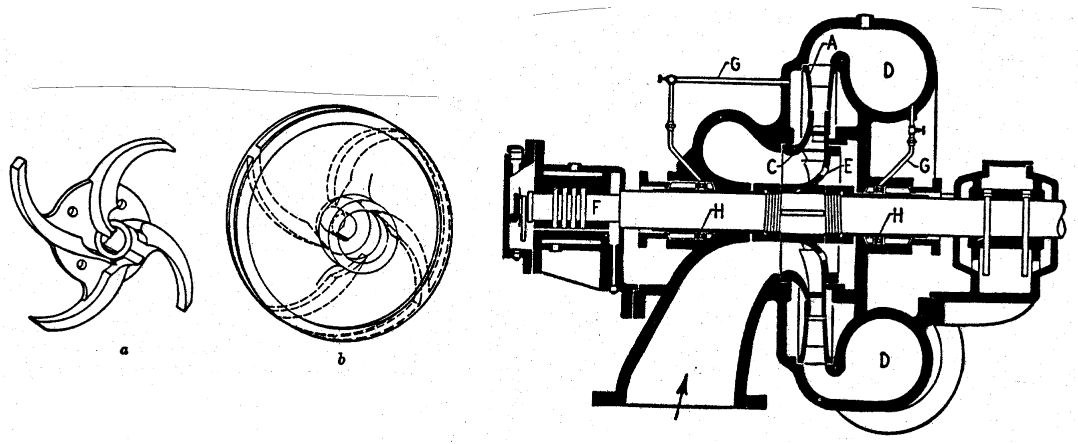


◇ Impeller type

(a) open-impeller volute pump

(b) closed-impeller volute pump

(c) turbine pump



◇ Performance Relationships for Ideal Centrifugal Pumps

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}, \quad \frac{\text{Head}_1}{\text{Head}_2} = \frac{N_1^2}{N_2^2}, \quad \frac{\text{Power}_1}{\text{Power}_2} = \frac{Q_1\text{Head}_1}{Q_2\text{Head}_2} = \frac{N_1^3}{N_2^3}$$

◇ Pump Efficiency

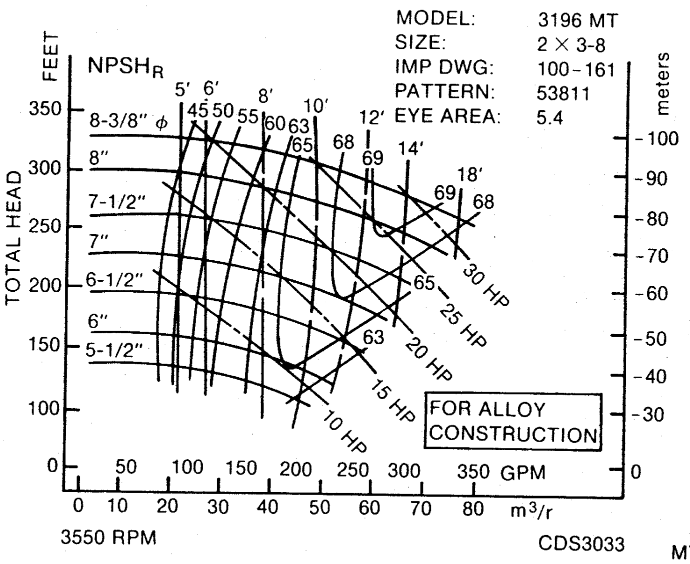
$$\eta_P = \frac{\text{Hydraulic energy delivered to the fluid}(Q\text{Head})}{\text{Break horse power}}$$

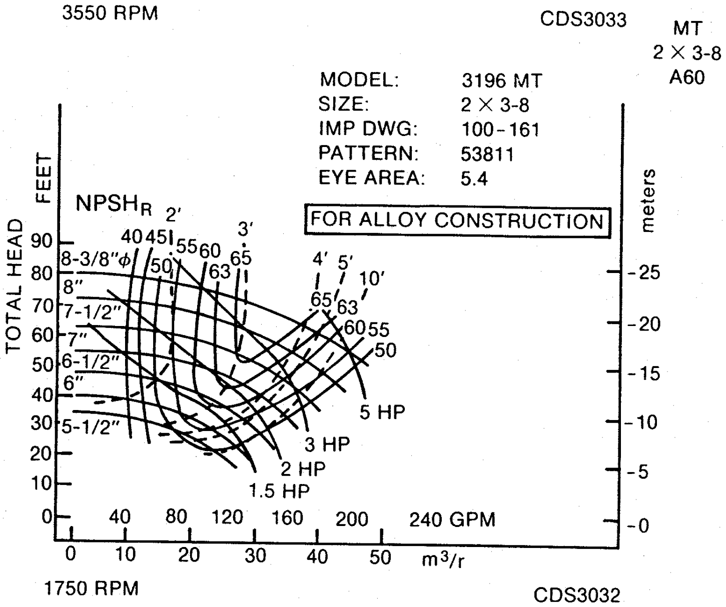
where Break horse power = hp delivered to the pump shaft
Hence,

$$\text{Required Motor hp} = \frac{W \text{ from the energy balance}}{\eta_P \eta_M}$$

where η_M is the motor efficiency ($\approx 80 - 90\%$).

◊ Performance Curve for Centrifugal Pumps





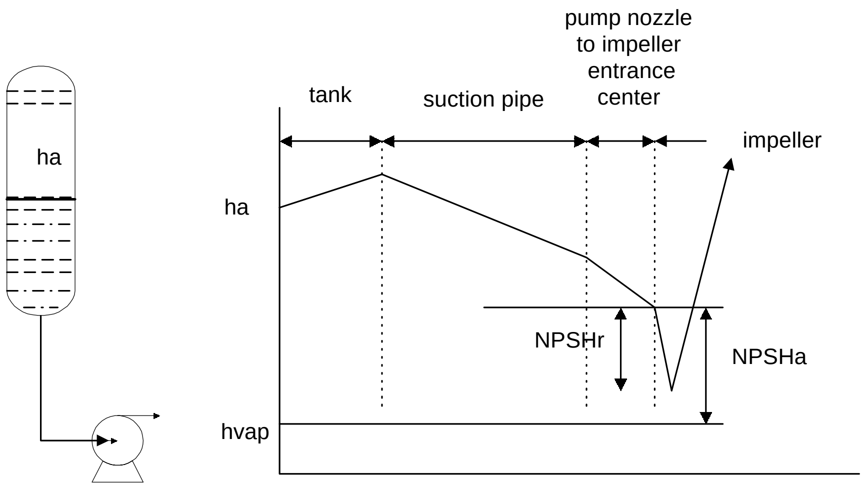
◊ **Cavitation**

Once cavitation (forming of vapor bubbles) occurs, centrifugal force suddenly drops and pump and/or piping damage is caused.

NPSHr (Net Positive Suction Head Required) = the amount of pump suction head, provided by the pump manufacturer

NPSHa (Net Positive Suction Head Available) = estimated pressure at the pump suction entrance point minus vapor pressure of the liquid at the operating conditions.

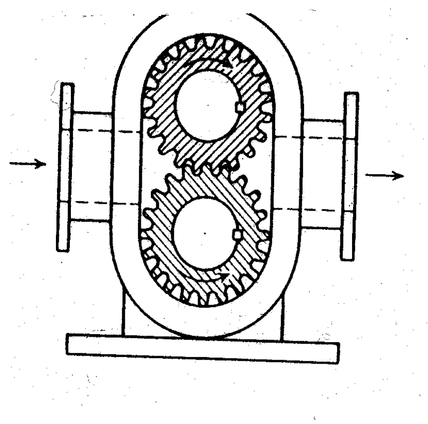
For no cavitation, NPSHa > NPSHr.



1.3.2 Other Types of Pumps

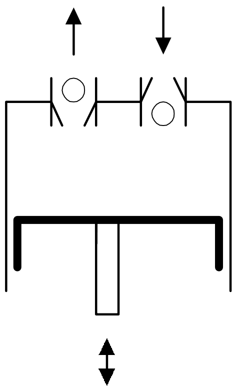
◊ Rotary Positive-Displacement Pumps

- Used for high viscous fluids
- Gear pump, $\cdots$

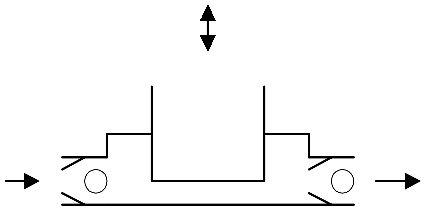


◊ Reciprocating or Positive Displacement Pumps with Valve Action

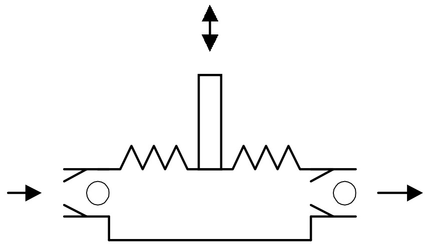
Used when high pumping head is required.



piston

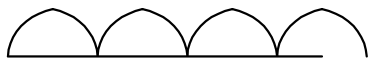


plunger

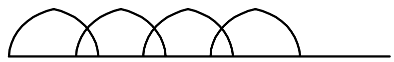


diaphram

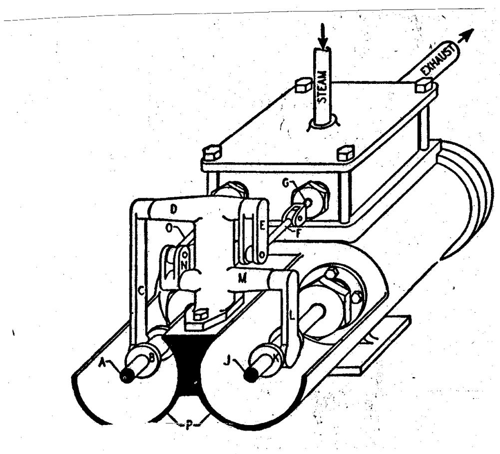
simplex

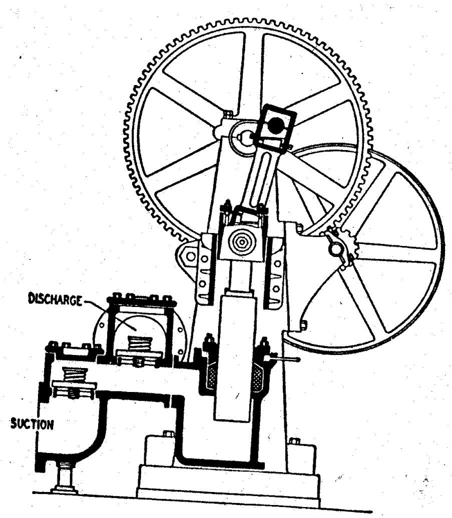


duplex



triplex



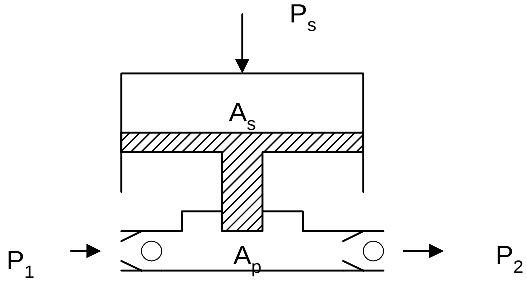


◇ Efficiency

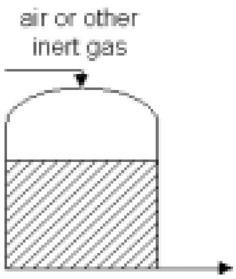
For pumping,

$$A_s P_s \geq A_p P_2 \ \rightarrow \ P_2^{max} = \frac{A_s}{A_p} P_s$$

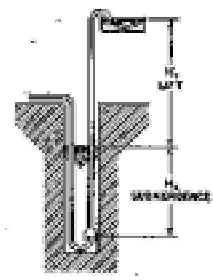
- Volume efficiency = (delivered volume)/(Volume by A_p strock)
- Pressure efficiency = (delivered P_2)/(theoretical P_2^{max})



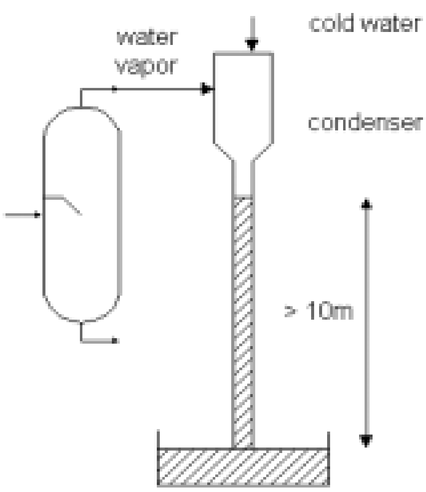
◇ Non-Mechanical Pumping Systems



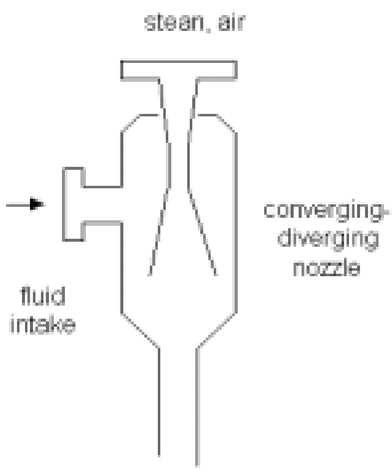
Blow Case or Aicd Egg



Air Lift



Barometric Leg



Ejector or Jet Pump

1.3.3 Valve Sizing and Pump Selection

Flow Characteristics

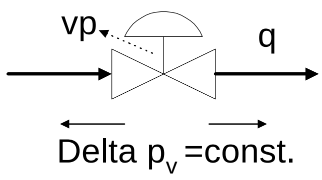
- Ideal valve performance equation

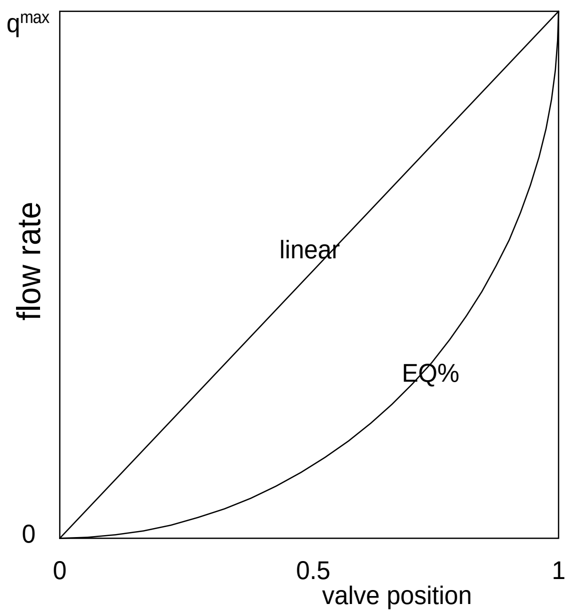
$$q(\text{gal/min}) = C_v \times f(vp) \times \sqrt{\frac{\Delta P_v(\text{psig})}{\text{sp.gr}}}, \quad f(vp) = \begin{cases} vp & \Rightarrow \text{ linear} \\ \sqrt{vp} & \Rightarrow \text{ quick opening} \\ R^{vp-1} & \Rightarrow \text{ equal percentage} \end{cases}$$

  - C_v - valve coefficient, customarily refers to the water flow rate (gallon/min) when $\Delta P_v = 1psig$. For example, $C_v = 0.3$ means $0.3[gallon/min]$ of water can flow when the valve is fully open under $\Delta P_v = 1psig$.
  - vp - valve position, 0-1.
  - R -rangeability, refers to the ratio of the maximum flow rate ($vp = 1$) to the minimum controllable flow rate ($vp = 0$).

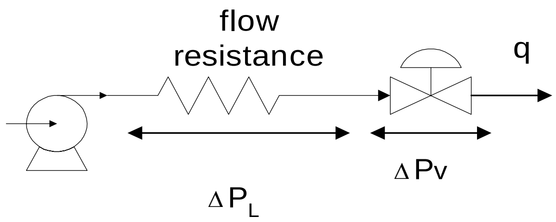
- Intrinsic flow characteristic

When ΔP_v is kept constant,





- Installed flow characteristic



Assume that $\Delta P_L + \Delta P_v = \Delta P_o =$ constant.
Let sp.gr = 1 (water). From

$$\Delta P_L = \frac{sp.gr}{C_L^2} q^2 \qquad \Delta P_v = \frac{sp.gr}{(C_v f(vp))^2} q^2$$

$$\Delta P_O = sp.gr \left[\frac{1}{C_L^2} + \frac{1}{(C_v f(vp))^2} \right] q^2 = \frac{sp.gr}{C_E(vp)^2} q^2$$

Therefore,

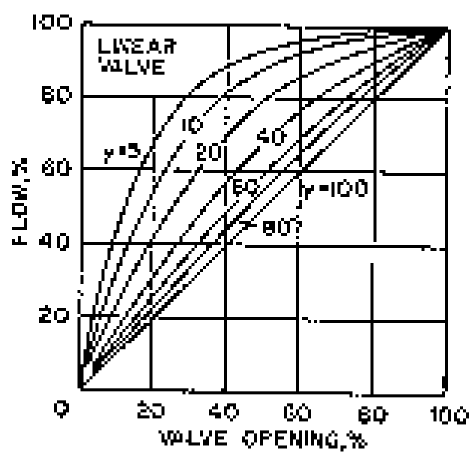
$$q = C_E(vp) \sqrt{\frac{\Delta P_O}{sp.gr}} \quad \text{where} \quad \frac{1}{C_E(vp)} = \sqrt{\frac{1}{C_L^2} + \frac{1}{(C_v f(vp))^2}}$$

Also

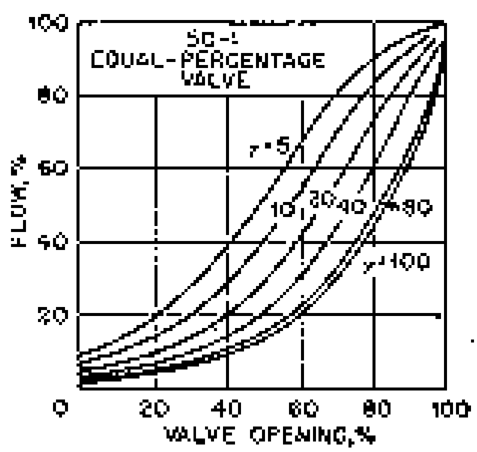
$$\Delta P_v = \frac{C_E^2(vp)}{C_v^2 f(vp)^2} \Delta P_O \quad \Rightarrow \quad \frac{\Delta P_v}{\Delta P_O} = \frac{C_E^2(vp)}{C_v^2 f(vp)^2}$$

Define

$$\gamma = \frac{\Delta P_{v,min}}{\Delta P_{v,max}} = \frac{\Delta P_v(vp = 1)}{\Delta P_O} = \frac{1}{1 + (C_v/C_L)^2}$$



(a)



(b)

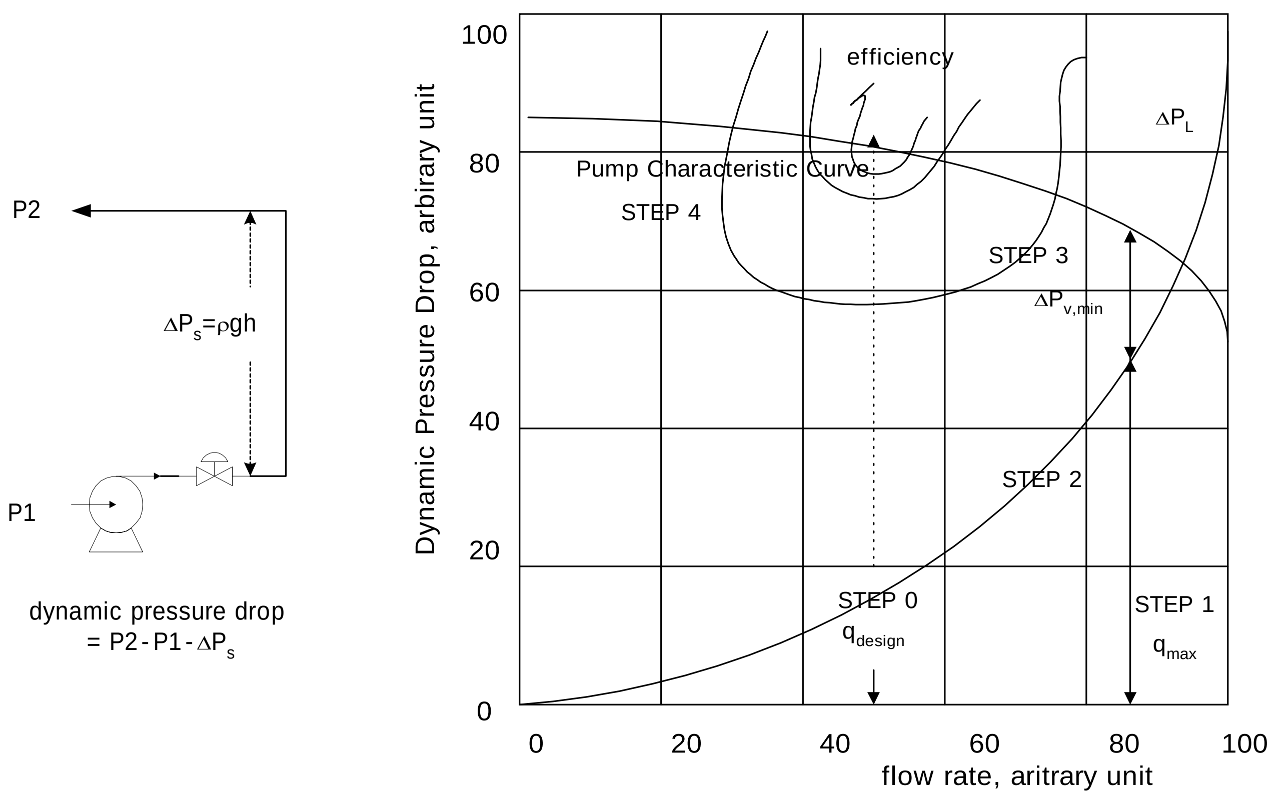
– Since ΔP_v decreases as q increases, the flow characteristic of EQ-valve tends to be linearized while that of the linear valve gets nonlinear.

– The nominal opening of a properly sized valve is around 50-70%. Based on this fact, $\gamma = 5 - 30\%$ is considered an appropriate choice for the sizing of an EQ% valve in view of linearity.

– An adverse effect for choosing a small γ (large C_v) is that the effective rangeabiliy (q_{max}/q_{min}) is decreased. For example, $R_{effective} = 16$ when $R = 50$ and $\gamma = 10\%$. Hence, $\gamma = 10 \sim 30\%$. This can be calculated using

$$\frac{q_{max}}{q_{min}} = \frac{C_E(vp=1)}{C_E(vp=0)} = \frac{\sqrt{1/C_L^2 + R^2/C_v^2}}{\sqrt{1/C_L^2 + 1/C_v^2}} = \frac{\sqrt{C_v^2/C_L^2 + R^2}}{\sqrt{C_v^2/C_L^2 + 1}} = \sqrt{(R^2-1)\gamma + 1}$$

Summary: EQ% Valve Sizing and Pump Selection



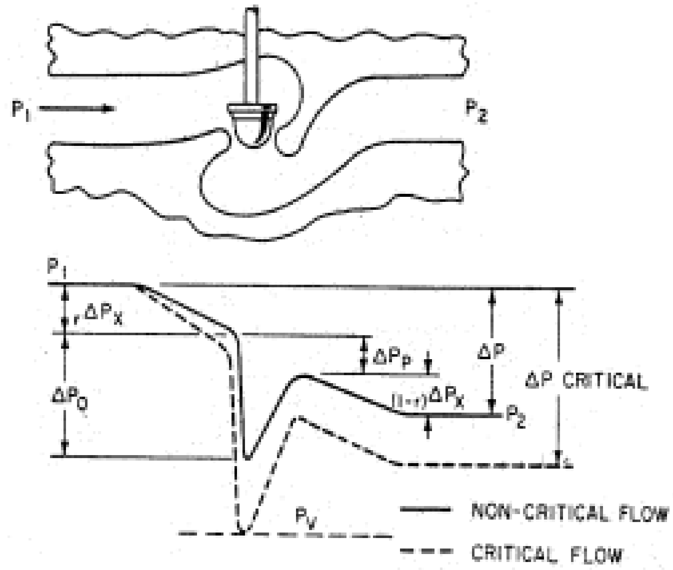
- Step 0 : Given a pipeline, or equivalently C_L, and a design flow rate q_{design}.
- Step 1 : Choose $q_{max} = (1.5 \sim 2.0) q_{design}$

- Step 2 : Draw ΔP_L vs. q using $\Delta P_L = (sp.gr/C_L^2)q^2$.
- Step 3 : Choose $\Delta P_{v,min}$ as $10 \sim 30\%$ of $\Delta P_{L,max}$ at q_{max}.
- Step 4 : Select a pump which has sufficiently high efficiency at q_{design} and, at the same time, can provide $\Delta P_{v,min} + \Delta P_{L,max} + \Delta P_s$ of head at q_{max}.
- Step 5 : Calculate C_v using

$$C_v = q_{max}\sqrt{\frac{sp.gr}{\Delta P_{v,min}}}$$

Further on Valve Performance

◇ **Critical Flow Factor**



- The dynamic pressure of the fluid becomes minimum at the vena contracta.
- When the minimum pressure is higher than the vapor pressure at the fluid temperature, the fluid is said to be in a *non-critical flow* state.
- Otherwise, we call the flow state as *critical flow*. In the critical flow, part of liquid starts to vaporize. As the fluid pressure is recovered as the fluid flows down along the pipeline, the vapor is condensed to liquid. During this transition, implosion may occurs. This causes noisy sound and vibration.
- Critical flow factor C_f represents the ratio of C_v at the critical flow state to C_v at the normal state. C_f value is different for different valve design and but usually lies between 0.6 and 0.9.

$$C_f = \frac{C_v \text{ at the critical flow state}}{C_v \text{ at the normal state}}$$

♢ Sizing Equation for Incompressible Fluids

- For subcritical flow, $\Delta P < C_f^2 \Delta P_s$

$$C_v = q\sqrt{\frac{sp.gr}{\Delta P}}$$

- For critical flow, $\Delta P \geq C_f^2 \Delta P_s$

$$C_v = \frac{q}{C_f}\sqrt{\frac{sp.gr}{\Delta P}}$$

where

$$\begin{aligned} \Delta P_s[psig] &= P_1 - P_{vapor}, \quad \Delta P[psig] = P_1 - P_2 \\ q[gal/min] &= \text{volumetric flow rate} \end{aligned}$$

♢ Sizing Equations for Compressible Fluids

For both critical and subcritical states,

- Gases

$$C_v = \frac{Q\sqrt{GT}}{834 C_f P_1 (y - 0.148 y^3)}$$

- Saturated Steam

$$C_v = \frac{W}{1.83 C_f P_1 (y - 0.148 y^3)}$$

- Superheated Steam

$$C_v = \frac{W(1 + 0.0007 T_{sh}}{1.83 C_f P_1 (y - 0.148 y^3)}$$

where

$$y = \min\left[\frac{1.63}{C_f}\sqrt{\frac{\Delta P}{P_1}}, \quad 1.5\right]$$

$$
\begin{aligned}
G &= \text{specific gravity of the fluid relative to air at } 1atm, 15^oC \\
Q &= \text{Volumetric flow at } 1atm, 15^oC(ft^3/min) \\
T_{sh} &= \text{degree of superheat}[^oF] \\
W &= \text{mass flow rate } [lb/min]
\end{aligned}
$$

1.4 Compressors, Gas Transporation

1.4.1 Gas Transportation Devices

- Fan - $\Delta P < 0.5psi$
- Blower - $\Delta P < 50psi$
- Compressor - ΔP up $4,000$ atm or more
- Vacuum pump

1.4.2 Compressors

Performance Equation

$$
\frac{g}{g_c}z_1 + \frac{v_1^2}{2\alpha g_c} + H_1 + Q + W = \frac{g}{g_c}z_2 + \frac{v_2^2}{2\alpha g_c} + H_2
$$

Usually Δv^2, Δz and Q are much smaller than Δh and W. Hence

$$
H_2(T_2, P_2) - H_1(T_1, P_1) = W
$$

1. The above relationship (adiabatic compression) holds for both reciprocating compressors as well as rotary compressors.
2. The reverse process to the compressor is the expander (usually turbines).
3. Note that all the thermodynamic properties in the above and in the subsequent derivations are on the unit-molar basis.

Two Ideal Compressors: Isentropic and Isothermal Compressors for an Ideal Gas

- **Isentropic Compressor** : $S_2 = S_1$ - reversible process (no lost work)
  - PVT Relationship:

    For any single component gas, $S = S(T, P)$ also $S = S(T, V)$.
    For the first relationship and the ideal gas law,

    $$dS = \left(\frac{\partial S}{\partial T}\right)_P dT + \left(\frac{\partial S}{\partial P}\right)_T dP = \frac{C_p}{T} dT - \left(\frac{\partial V}{\partial T}\right)_P dP = C_p d \ln T - R d \ln P$$

    For the second relationship and the ideal gas law,

    $$dS = \left(\frac{\partial S}{\partial T}\right)_V dT + \left(\frac{\partial S}{\partial V}\right)_T dV = \frac{C_v}{T} dT + \left(\frac{\partial P}{\partial T}\right)_V dV = C_v d \ln T + R d \ln V$$

    $dS = 0$ implies

    $$\ln \frac{T_2}{T_1} = \frac{R}{C_p} \ln \frac{P_2}{P_1} \quad \text{and} \quad \ln \frac{T_2}{T_1} = -\frac{R}{C_v} \ln \frac{V_2}{V_1}$$

    Hence, we obtain

    $$\frac{T_2}{T_1} = \left(\frac{V_2}{V_1}\right)^{-R/C_v}, \quad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{R/C_p} \quad \text{and} \quad \frac{P_2}{P_1} = \left(\frac{V_2}{V_1}\right)^{-C_p/C_v} \tag{1.1}$$

  - Enthalpy Change and Required Work:

    From

    $$dH = TdS + VdP$$

    and dS expression in the above,

    $$dH = C_p dT + \left[V - T\left(\frac{\partial V}{\partial T}\right)_P\right] dP$$

    Inserting the ideal gas law yields

    $$dH = C_p dT \quad \rightarrow \quad W = \Delta H = C_p(T_2 - T_1) \tag{1.2}$$

    Using (1.1) and (1.2), we can obtain

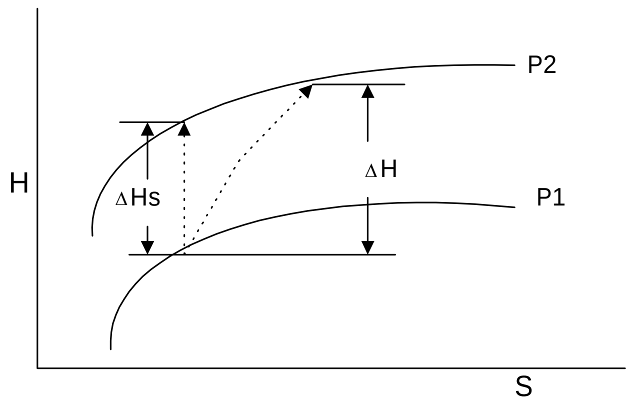
$$W_S = \frac{k}{k-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right] \quad \text{(energy/mole)} \quad \text{where} \quad k = C_p/C_v$$

- **Isothermal Compressor** : $T_2 = T_1$

$$W_T = P_1 V_1 \ln \left(\frac{P_2}{P_1} \right)$$

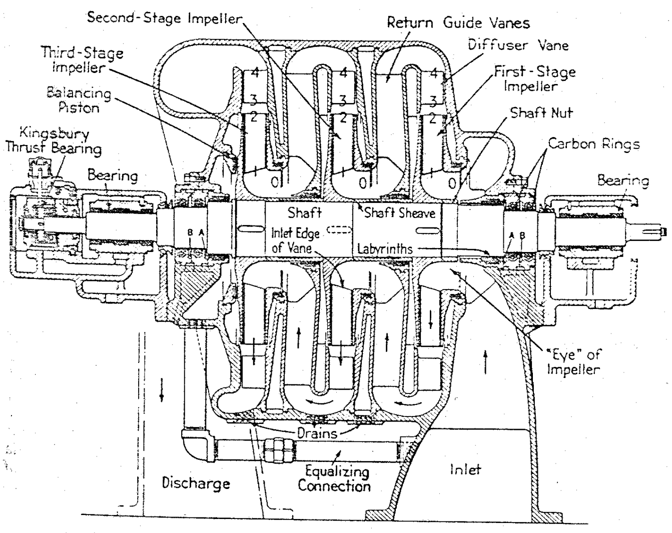
Efficiency

$$\eta = \frac{\text{Theoretical Power Required}}{\text{Actual Power Required}} = \frac{\Delta H_S}{\Delta H_{true}} = \frac{(P_2/P_1)^{(k-1)/k} - 1}{(T_2/T_1) - 1}$$

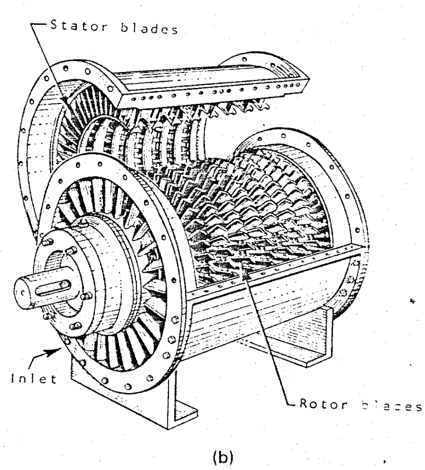


Types and Characteristics of Rotary Compressors

- **Centrifugal compressor vs. axial compressor**



(a)



(b)

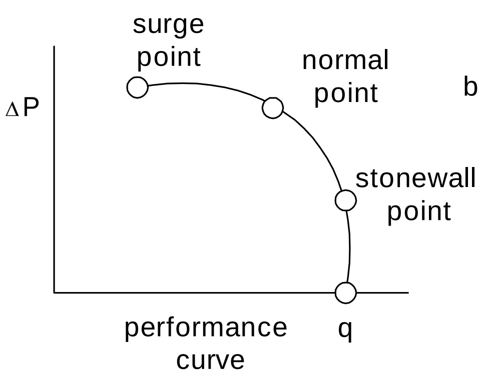
- **Surge Problem**

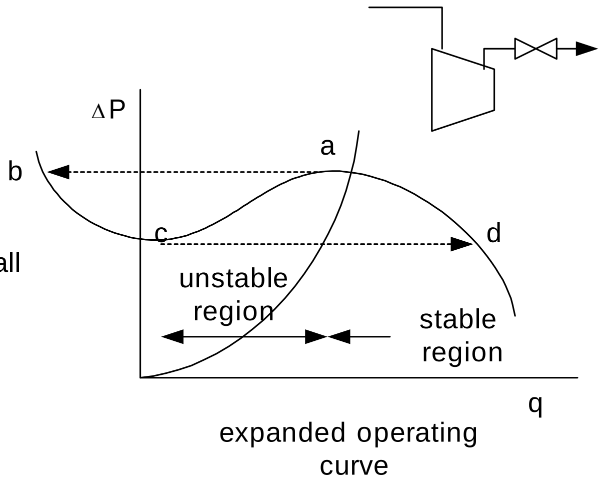
  After the stonewall point, opening the valve wider does not increase the flow rate more as the velocity approaches the sonic velocity somewhere in the compressor.

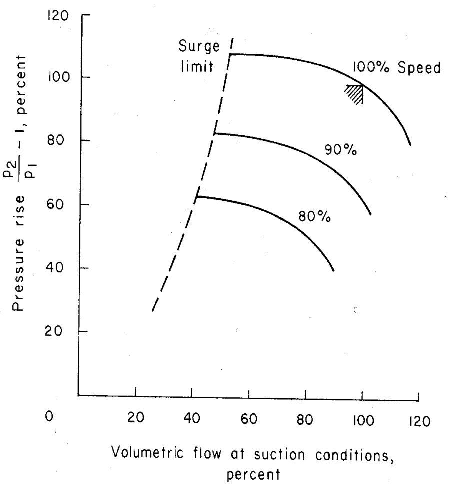
  Closing the valve causes the discharge pressure to rise and the throughput to decrease. As the valve is closed further, an aero-dynamical instability, called "surge", occurs.

  Surge refers to a very fast repetition (with period of less than 1 or 2 seconds) between sudden back-flow and high forward flow of gas. When the load curve meets the compressor performance curve at "a", the system oscillates along a-b-c-d-a. It is due to the negative resistance (unstable) characteristic of rotary compressors.

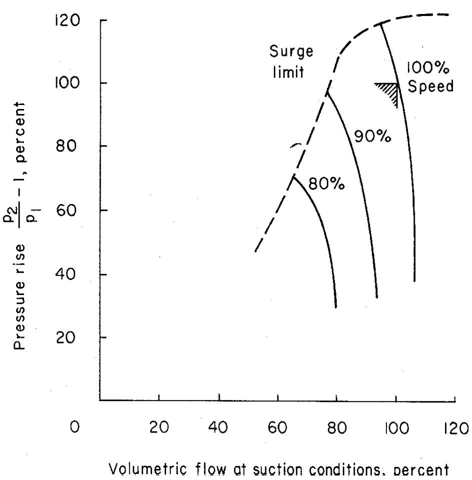
  Surge causes serious damage to the compressor.
  Surge protection control is mandatory in industrial compressors







Centrifugal Compressor



Axial Compressor

1.4.3 Costs for Pumping Machinery

Fig.14-39 through 14-54

1.5 Flow Measuring Equipments

	FLUID PROPERTY LIMITATIONS				APPLICATION FACTORS					INSTALLATION FACTORS					
METER TYPE	Type of Fluid	Pressure Limits	Temperature Limits	Suitability for Slurries, etc.	Range of Maximum Flows	Individual Meter Range	Accuracy	Type of Scale	Construction Materials Available	Line Sizes Available	Normal Connections	Special Piping Considerations	Power Requirements for Basic Measurement	Installation Limits	Pressure Loss
VARIABLE-AREA Float Type (glass tube)	Suitable for most fluids except alkalis and molten metals	Varies with sizes — ½ in. - 300 psi 1 in. - 200 psi 1½ in - 100 psi 2 in and larger - 50 psi	-50°F to +400°F	Not Recommended	½ cc/min to 200 gpm	10:1	±2% of full scale uncalibrated; ±1% of full scale calibrated	Linear	All Metals Plastics Ceramics (floats and fittings)	¼ in. to 4 in.	Screwed or Flanged	None	None	Meter body must be installed in vertical plane	Available from ¼ in. to 30 in. of water — constant throughout meter range
VARIABLE-AREA Float Type (metal tube)	Suitable for all fluids	150 lb ASA to 2500 lb ASA	-300°F to +1600°F	Recommended	0.1 to 5000 gpm	10:1	±2% of full scale uncalibrated; ±1% of full scale calibrated	Linear	All Metals	½ in. to 12 in.	Flanged	None	None	Meter body must be installed in vertical plane	Available from 1 to 200 in. of water — constant thoughout meter range
VARIABLE HEAD (static manometer)	Suitable for most fluids	To 1500 psi	-60°F to +400°F	Requires purges of pressure top connections	100 gpm and up	4:1 useful range	±1% of differential measured — additive to accuracy of primary restriction	Square Root	Steel and Stainless Steel (purges or seal pots used on corrosive services)	2 in. and up	Screwed or Flanged	Straight pipe run required for approx. 10 pipe diameters upstream, 3 downstream	None	Horizontal or Vertical	Available from 1 to 800 in. of water — varies as square of flow change
INTEGRATING OR VOLUME (positive displacement type)	Suitable for all clean or filtered fluids	150 lb ASA to 300 lb ASA standard	0°F to +300°F	Not Recommended	1 to 5000 gpm	Up to 20:1	0.1% - 0.5% of total volume	Linear	Bronze Steel Stainless Steel	½ in. to 12 in.	Screwed or Flanged	None	None	Horizontal Recommended	Up to 7 psi (varies with flow)
INTEGRATING OR VOLUME (turbine type)	Suitable for all clean or filtered liquids	Up to 3000 psi	-300°F to +700°F	Not Recommended	0.25 to 2500 gpm	Up to 15:1	±0.5% of rate	Linear	Most Metals	½ in. to 6 in.	Screwed or Flanged	None	Power Supply Needed (115 v, 60 c, 350 w)	Operates in any position	Up to 7 psi (varies with flow)
OBSTRUCTIONLESS (ultrasonic type)	Suitable for most liquids	Up to 1500 psi	-300°F to +600°F	Recommended	200 gpm and higher	20:1	±2% of full scale	Linear	Bronze Steel Stainless Steel	2 in. and up	Flanged	Straight pipe run required for approx. 10 pipe diam. upstream	Power Supply Needed (115 v, 60 c, 100 w)	Horizontal or Vertical	None
OBSTRUCTIONLESS (magnetic type)	Suitable for most liquids that are electrical conductors	Up to 600 psi	-200°F to +360°F	Recommended	5 to 5000 gpm	20:1	±1% of full scale	Linear	Kel-F or Neoprene lined — Stainless Steel	1 in. to 12 in.	Flanged	None	Power Supply Needed (115 v, 60 c, 200 w)	Horizontal or Vertical	None

Table VII. FACTORS TO BE CONSIDERED IN FLOWMETER SELECTION (2)

◇ Area Flowmeter

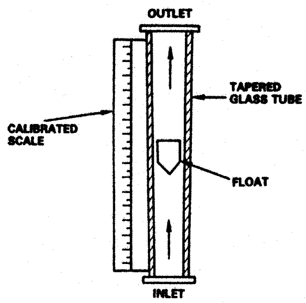


Figure 6-34. AREA FLOWMETER, ROTAMETER

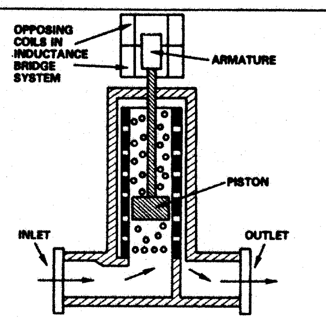


Figure 6-35. AREA FLOWMETER, CYLINDER AND PISTON TYPE

◇ Head Flowmeter

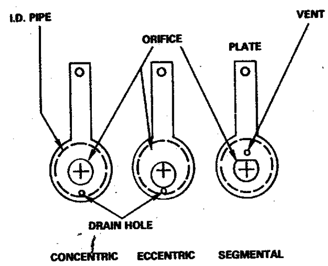


Figure 6-6. ORIFICE PLATES, TYPES.

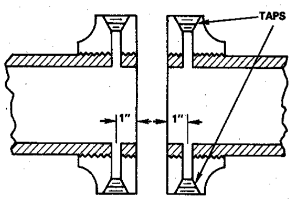


Figure 6-9. FLANGE TAPS, HIGH & LOW

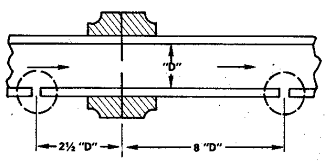


Figure 6-10. PIPE TAPS, HIGH & LOW

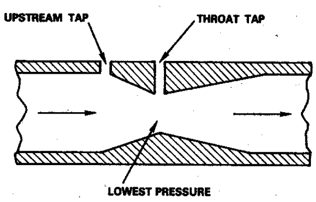


Figure 6-14. VENTURI PRINCIPLE

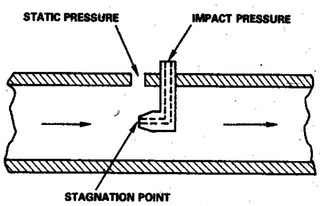


Figure 6-15. PITOT TUBE, INDEPENDENT STATIC TAP

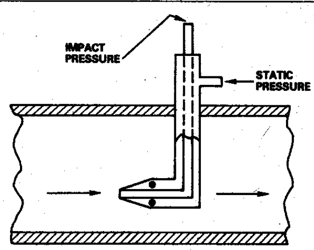


Figure 6-16. PITOT TUBE, COMBINED STATIC & IMPACT TAPS

◇ Positive Displacement Meter

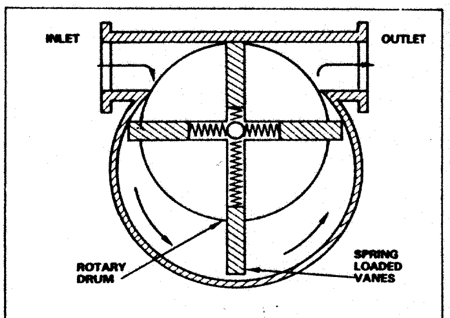


Figure 6-28. ROTARY VANE METER

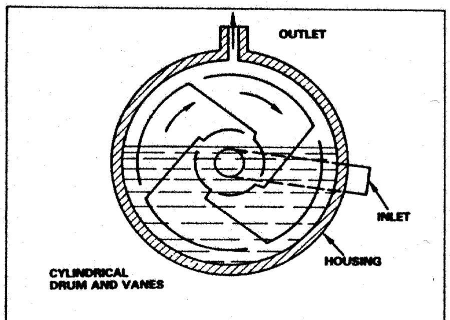


Figure 6-29. SEALED-DRUM METER

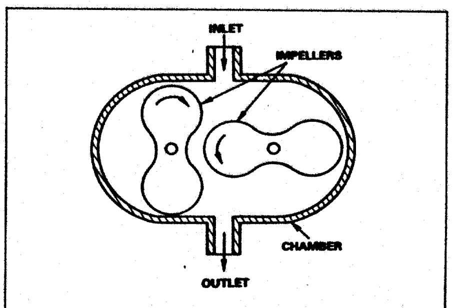


Figure 6-30. LOBED-IMPELLER METER

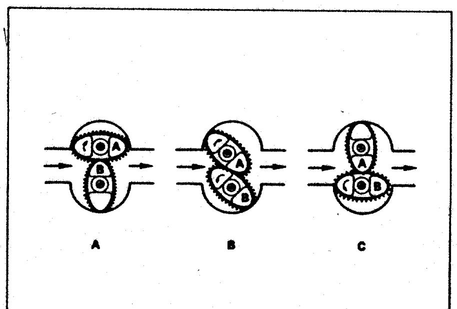


Figure 6-33. PRINCIPLE OF OPERATION DIAGRAM FOR THE OVAL GEAR FLOWMETER (41)

◇ Volumetric Flowmeter

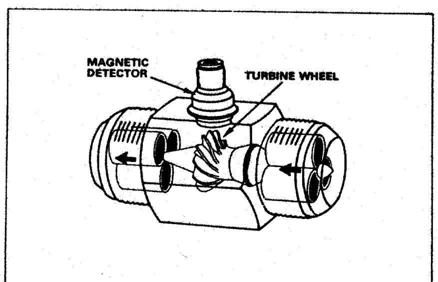


Figure 6-42. TURBINE-TYPE FLOWMETER (4)

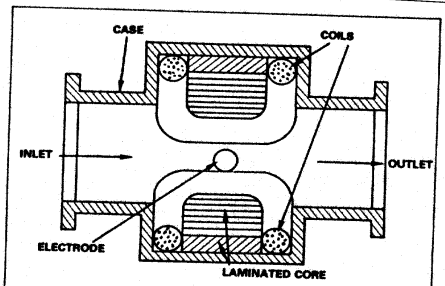


Figure 6-43. MAGNETIC FLOWMETER

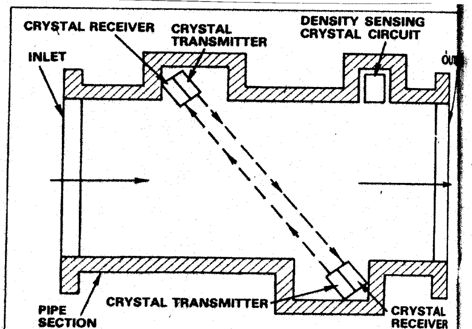


Figure 6-44. ULTRASONIC FLOWMETER

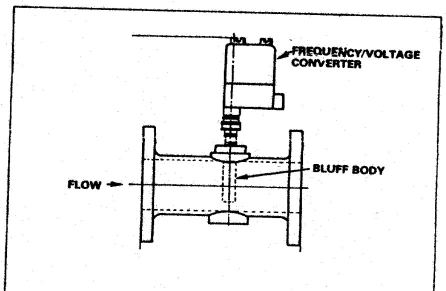


Figure 6-47. VORTEX SHEDDING FLOWMETER

◇ Mass Flowmeter

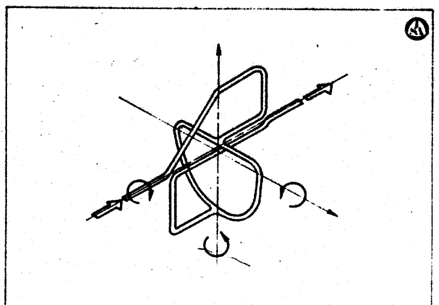


Figure 6-39. GYROSCOPIC SENSING ELEMENT, PIPE CONFIGURATION

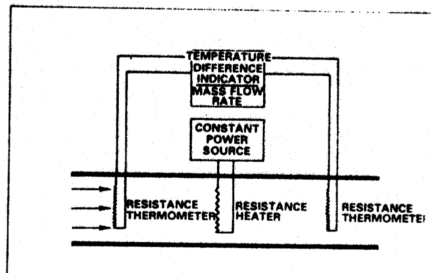
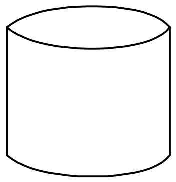


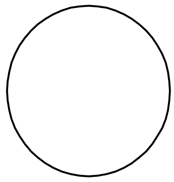
Figure 6-41. THERMAL MASS FLOWMETER (53)

Plus the mass flowmeter using Corilolis force principle.

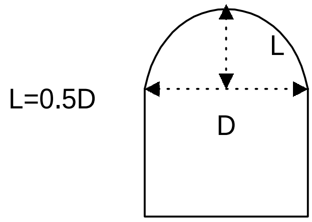
1.6 Tanks and Pressure Vessels



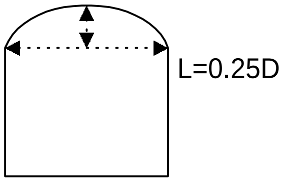
cylindrical shell tank - usual shape



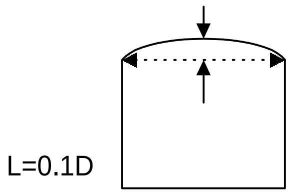
spherical shell tank - for high pressure



hemispherical head tank - high pressure

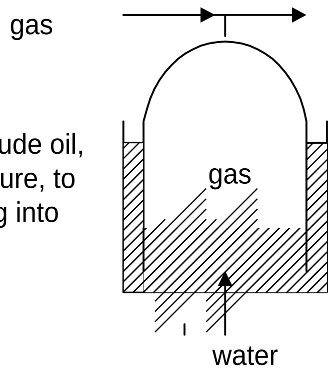


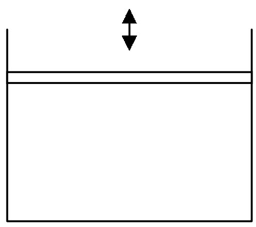
Ellipsoidal head tank - medium pressure



Torispherical(10% dished) head tank

Floating head tank - for crude oil, gas strorage at low pressure, to protect air from intruding into vacant space





◇ Thickness calculation for cylindrical shell

$$t(\text{inch}) = \frac{P(\text{max.press, pis})r_i(\text{inner radius, in})}{S(\text{tensile strength, psi})E_J(\text{joint/welding efficiency, } 0-1) - 0.6P(\text{psi})} + C_c(\text{allowance})$$