PUMPS THAT EXPERTS SELECT.

Fluid Mechanics

Nomenclature	
atm	atmospheric pressure
A	area
BHP	brake horsepower; hp
c	speed of sound
CC	capital charge constant, 1/yr
D	diameter
D_0	diameter of the pipe wall
f	friction factor
F	friction heating per unit mass
g	gravity constant
h	head
h_{fs}	suction friction head
h_{gs}	gauge reading at the suction flange of a pump
h_{ss}	static suction head
h_{vs}	velocity head at point of gauge attachment
H	total dynamic head
K	bulk modulus
m	mass flowrate
M	Mach number
(NPSH)a	available net positive suction head
NPSH)r	required net positive suction head
N	impeller rotational speed, rpm
p	vapor pressure
P	pressure
PC	pumping cost constant, $/(hp-yr)
PP	purchasing price constant, $/in., (dia.) x feet (of length)
Q	volumetric flowrate
Re	Reynolds number
s	specific gravity
V	velocity
z	position in the direction opposite that of gravity
Δ	pipe length
ε	pipe roughness
μ	fluid viscosity
ρ	density

<u>Energy Relationships</u> (incompressible flow)

Bernoulli's Equation:

$$P_2 - P_1 = \rho \frac{V_1^2}{2}\left(1 - \frac{A_1^2}{A_2^2}\right) - \rho \cdot F$$

Diffuser and sudden expansions:

$$P_2 - P_1 = \rho \frac{V_1^2}{2} - \rho \cdot F$$

Friction Heating in a Pipe:

$$F = Q \cdot \Delta \times \frac{\mu \;\; 128}{\rho \;\; \Pi \cdot D_0^4}$$

Torricelli's Equation:
(For flow from the bottom of a vessel)

$$V_2 = \sqrt{2 \cdot g \cdot h}$$

Pressure-Depth Relationship:
(Constant density)

$$P_2 - P_1 = -\rho \cdot g(z_2 - z_1)$$

Reynold's Number:

$$\mathrm{Re} = \frac{D \cdot V \cdot \rho}{\mu}$$

Friction Factor (f):

$$\frac{1}{\sqrt{f}} = -4\log = \left[\frac{0.27\varepsilon}{D} + \left(7/\mathrm{Re}\right)^{0.9}\right]$$

$$\mathrm{Re} \geq 4000$$

Mach number:

$$M = V/c$$

Speed of Sound:

$$c = \sqrt{\frac{K}{\rho}}$$

A) <u>Pump, Compressor, and Pipe Equations</u>

Economic Pipe Diameter:

$$D_{econ} = \left[\frac{10 \cdot PC \cdot m^3 f (4/\Pi)^2 (1/p^2)}{CC \cdot PP} \right]^{1/6}$$

Pump Power Output:

$$kW = H \cdot Q \cdot \rho / 3.670 \times 10^5$$

(H in $N \cdot m / kg$; Q in m^3 / h ; ρ in kg / m^3)

$$kW = H \cdot Q / 3.599 \times 10^6$$

(H in Pa; Q in m^3 / h)

$$hp = H \cdot Q \cdot s / 3.960 \times 10^3$$

(H in $lb_f \cdot ft / lb_m$; Q in gal/min)

$$hp = H \cdot Q / 1.714 \times 10^3$$

(H in lb_f / in^2 ; Q in gal/min)

Net Positive Suction Head*:
(Be sure to convert pressure units to head)

$$(NPSH)a = h_{ss} - h_{fs} - p$$

For an existing installation:

$$(NPSH)a = atm + h_{gs} - p + h_{vs}$$

* To avoid cavitiation, (NPSH)a $\geq$ (NPSH)r

Head-Flow Relationship:
(pumps)

$$\frac{(Q_2)^2}{(Q_1)^2} = \frac{h_2}{h_1}$$

Specific Speed (N_s):**

$$N_s = \frac{N \cdot Q^{0.5}}{H^{0.75}}$$

(Q in gal/min; H in $ft \cdot lb_f / lb_m$)

** For compressors, H is adiabatic head

Suction Specific Speed (S):

$$S = \frac{N \cdot Q^{1/2}}{(NPSH)^{3/4}}$$

Specific Diameter (D_s):
(Compressors)

$$D_s = \frac{D \cdot H^{0.25}}{Q^{0.5}} \quad D_s = \frac{D \cdot H^{0.25}}{Q^{0.5}}$$

Flow Coefficient (ϕ):
(Compressors)

$$\phi = \frac{Q}{N \cdot D^3}$$

Pressure Coefficient (Ψ):

$$\Psi = \frac{H}{N^2 \cdot D^2}$$

The Affinity Laws (pumps)

Capacity: Constant Impeller Dia. | Constant Impeller Speed (D = impeller dia.)

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \qquad \frac{Q_1}{Q_2} = \frac{D_1}{D_2}$$

Head: Constant Impeller Dia. | Constant Impeller Speed (D = impeller dia.)

$$\frac{H_1}{H_2} = \frac{(N_1)^2}{(N_2)^2} \qquad \frac{h_1}{h_2} = \frac{(D_1)^2}{(D_2)^2}$$

Break Horsepower: Constant Impeller Dia. | Constant Impeller Speed (D = impeller dia.)

$$\frac{BHP_1}{BHP_2} = \frac{(N_1)^3}{(N_2)^3} \qquad \frac{BHP_1}{BHP_2} = \frac{(D_1)^3}{(D_2)^3}$$

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