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Kolmetz Handbook of Process Equipment Design

Piping Hydraulics Fluid Flow Line Sizing and Material Selection

(ENGINEERING DESIGN GUIDELINE)

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APPLICATION

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KLM Technology Group is providing the introduction to this guideline for free on the internet. Please go to our website to order the complete document.

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INTRODUCTION

Scope

The understanding of how gasses and fluids flow in equipment is the foundation of equipment design. All of the other Engineering Design Guidelines are based on these fundamentals; therefore it is critical that the principles of fluid flow are understood before designing equipment. The principles are not complex, but neither are they simple due to the interdependence of pressure drop and friction.

This design guideline covers the basic elements in the field of Piping Fluid Flow Material Selection and Line Sizing in sufficient detail to design a pipeline and / or other piping classes. This design guideline includes single phase liquid flow, single phase gas flow for hydrocarbon, water, steam and natural gases. Two phase flow is covered in a separate guideline.

Proper pipe sizing is determined by the length of the pipe and the allowable pressure drop in the line. The allowable pressure drop may be influenced by factors, including process requirements, economics, safety, and noise or vibration limitations.

This guideline also covers other piping related equipment, such as valve, fittings and orifices. Pressure drop calculations in these fitting are discussed in detail to help the design of piping systems.

Fluid phases can be considered as pure liquid or pure gas phases. In this guideline, these differences phases were discussed in detail for the engineering design for the laminar and turbulence flow and for various substances of fluids, for example, water, steam and hydrocarbon. A second guideline discusses mixed phase fundamentals.

The theory section covers the selection method of the piping material based on their application and engineering calculations for the sizing of the piping. In the application section of this guideline, four case studies are shown and discussed in detail, highlighting the way to apply the theory for the calculation.

Fundamental theories, such as Bernoulli's theory, is used as the basic of calculations because it is applicable for various conditions. The case studies will assist the engineer develop typical selection and sizing for the piping based on their own plant system.

Example Calculation Spreadsheets are included in this guideline. The Example Calculation Spreadsheets are based on case studies in the application section to make them easier to understand.

INTRODUCTION

General Design Consideration

In designing the piping fluid flow there are many factors have to be considered for the suitability of the material selection for the application codes and standards, environmental requirements, safety, performance of the requirements, and the economics of the design and other parameters which may constrain the work.

They should be included engineering calculations for the piping system design. Combined with the piping design criteria, calculations define the process flow rates, system pressure and temperature, pipe wall thickness, and stress and pipe support requirements.

The service conditions should be the consideration as well because the piping system is designed to accommodate all combinations of loading situations such as pressure changes, temperature changes, thermal expansion and contraction and other forces or moments that may occur simultaneously and they are used to set the stress limit of the design.

Design code and the standards are reviewed for the project of the design for the safety purposes and the verification of the applicability. In this design guideline generally follows the codes and the standards of the American Society of Mechanical Engineers (ASME) Code for Pressure Piping, B31. ASME B31 includes the minimum design requirements for various pressure piping applications.[4]

Normal environmental factors that have the potential for damage due to corrosion must be addressed in the design of process piping. Physical damage may also occur due to credible operational and natural phenomena, such as fires, earthquakes, winds, snow or ice loading, and subsidence. Two instances of temperature changes must be considered as a minimum. First, there are daily and seasonal changes. Second, thermal expansion where elevated liquid temperatures are used must be accommodated. Compensation for the resulting expansions contractions are made in both the piping system and support systems. Internal wear and erosion also pose unseen hazards that can result in system failures.

Most failures of fluid process systems occur at or within interconnect points the piping, flanges, valves, fittings, etc. It is, therefore, vital to select interconnecting equipment and materials that are compatible with each other and the expected environment. Materials selection is an optimization process, and the material selected for an application must be chosen for the sum of its properties. That is, the selected material may not rank first in each evaluation category; it should, however, be the best overall choice. Considerations include cost and availability. Key evaluation factors are strength, ductility, toughness, and corrosion resistance.

Piping material is selected by optimizing the basis of design. The remaining materials are evaluated for advantages and disadvantages such as capital, fabrication and installation costs; support system complexity; compatibility to handle thermal cycling; and catholic protection requirements. The highest ranked material of construction is then selected.

The design proceeds with pipe sizing, pressure integrity calculations and stress analyses. If the selected piping material does not meet those requirements, then second ranked material is used to sizing, pressure integrity calculation and stress analyses are repeated.

For the pressure drop calculation: the primary requirement of the design is to find an inside diameter with system design flow rates and pressure drops. The design flow rates are based on system demands that are normally established in the process design phase of a project. This will involves trial and error procedure to find the suitable inside diameter.

Basically service conditions must be reviewed to determine operational requirements such as recommended fluid velocity for the application and liquid characteristics such as viscosity, temperature, suspended solids concentration, solids density and settling velocity, abrasiveness and corrosively. This information is useful to determine the minimum internal diameter of the pipe for the whole system network.

Normal liquid service applications, the acceptable velocity in pipes is 2.1 ± 0.9 m/s (7 ± 3 ft/s) with a maximum velocity limited to 2.1 m/s (7 ft/s) at piping discharge points. This velocity range is considered reasonable for normal applications.(4)

Pressure drops throughout the piping network are designed to provide an optimum balance between the installed cost of the piping system and operating costs of the system pumps. Primary factors that will impact these costs and system operating performance are internal pipe diameter (and the resulting fluid velocity), materials of construction and pipe routing.

Pressure drop, or head loss, is caused by friction between the pipe wall and the fluid, and by minor losses such as flow obstructions, changes in direction, changes in flow area, etc. Fluid head loss is added to elevation changes to determine pump requirements. A common method for calculating pressure drop is the Darcy-Weisbach equation.

Normally for the line sizing the following rules should be follow

1) Calculate the Pressure drop with expressed in the term “psi/100 ft of pipe”.

2) Select the suitable Velocity which expressed in ft/sec; there is standard for general liquid flow the range of the velocity should be in the suitable range for the basic design.

3) Calculate the Reynolds number to determine the fluid flow. Reynolds number is a factor of pipe diameter, flow rate, density, and viscosity of the liquid; allows analysis of flow characteristics (slug, laminar, transition, turbulent); sanitary systems always require full turbulence (Reynolds number > 10,000)

4) Determine the suitable of pipe diameter - the inside pipe or tube diameter is used in the several equations to determine the pressure drops, Reynolds number, velocity and etc.

5) Determine the roughness of pipe, the more rough the pipe, the larger the friction factor; the larger the friction factor, the more pressure drop.

6) Incompressible flow - liquids; actual pressure is not a factor in pressure drop calculation.

7) Compressible flow - gases and vapors; actual pressure is a direct factor in pressure drop calculation.

Liquids (Incompressible Flow): Size longer lines for less pressure drop than shorter lines. Most water-like liquids, long lines should be sized for 0.5 to 1.0 psi/100 ft; short lines should be sized for 1.0 to 2.0 psi/100 ft; but there are no hard and fast rules.

For liquids with viscosities 10 cp or less consider just like water; above 10 cp, check Reynolds number to see what equations to use for pressure drop calculation. Careful with sizing lines in the fractional line size range; It may cost more to install ¾” pipe and smaller than 1” pipe due to support requirements.

Usually do not save on header sizing to allow for future increase in capacity without changing out piping. Pipeline holdup of process liquids may be a factor; smaller pipe may be desired to limit holdup even though pressure drop goes up.

Gases and Vapors (Compressible Flow): Supply pressure is a major factor in line sizing calculations; also, overall pressure drop by means of typical calculation methods should not exceed 10% of the supply pressure, otherwise, alternative calculation methods must be used. Typically, consider all gases and vapors (including saturated steam) to behave gases in order to calculate vapor densities (PV = nRT).

DEFINITIONS

Compressible Fluid - Molecules in a fluid to be compacted and the density is varies. Energy is exchanged not only among the kinetic energy and the potential energies due to gravity and pressure, but also with the internal energy [7].

Darcy Friction Factor, f -This factor is a function of Reynolds Number and relative pipe wall roughness, ε/d. For a given class of pipe material, the roughness is relatively independent of the pipe diameter, so that in a plot of f vs. Re, d often replaces ε/d as a parameter.

In-Compressible Fluid - An incompressible flow is one in which the density of the fluid is constant or nearly constant. Liquid flows are normally treated as incompressible [6]. Molecules in a fluid to be cannot be compacted. Generally the flow energy is converted to friction, kinetic and potential energy if available and not the internal energy.

Laminar Flow - Laminar flow occurs when adjacent layers of fluid move relative to each other in smooth streamlines, without macroscopic mixing. In laminar flow, viscous shear, which is caused by molecular momentum exchange between fluid layers, is the predominant influence in establishing the fluid flow. This flow type occurs in pipes when Re < 2,100.

Newtonian Fluids - A fluid characterized by a linear relationship between shear rate (rate of angular deformation) and shear stress.

Non-Newtonian Liquids - Fluids may be broadly classified by their ability to retain the memory of a past deformation (which is usually reflected in a time dependence of the material properties). Fluids that display memory effects usually exhibit elasticity.[8] Fluids in which viscosity depends on shear rate and/or time. Examples are some slurries, emulsions, and polymer melts and solutions.

Relative Roughness - Ratio of absolute pipe wall roughness ε to inside diameter d, in consistent units.

Reynolds Number, Re - A dimensionless number which expresses the ratio of inertial to viscous forces in fluid flow
Resistance Coefficient, K - Empirical coefficient in the friction loss equation for valves and fittings. It expresses the number of velocity heads lost by friction for the particular valve or fitting. The coefficient is usually a function of the nominal diameter.

Shear Rate - The velocity gradient (change in velocity with position).

Shear Stress - Force per unit area. Force in direction of flow; area in plane normal to velocity gradient.

Sonic Velocity (Choked Flow) - The maximum velocity that a gas or gas-liquid mixture can attain in a conduit at a given upstream pressure (except in certain converging-diverging nozzles), no matter how low the discharge pressure is. For gases this maximum velocity is equal to the speed of sound at the local conditions.

Specific gravity - Is a relative measure of weight density. Normally pressure has not significant effect for the weight density of liquid, temperature is only condition must be considered in designating the basis for specific gravity.

Steam Hammer - Steam hammer is excessive pipe vibrations that occur due to the collapse of large vapor bubbles in a cool liquid stream.

Transition Flow - Flow regime lying between laminar and turbulent flow. In this regime velocity fluctuations may or may not be present and flow may be intermittently laminar and turbulent. This flow type occurs in pipes when 2,100 < Re < 4,000.

Turbulent Flow - Turbulence is characterized by velocity fluctuations that transport momentum across streamlines; there is no simple relationship between shear stress and strain rate in turbulent flow. Instantaneous properties cannot be predicted in a turbulent flow field; only average values can be calculated. For engineering analyses, turbulent flow is handled empirically using curve-fits to velocity profiles and experimentally determinate loss coefficients. This flow type occurs in pipes in industrial situations when Re > 4,000. Under very controlled laboratory situations, laminar flow may persist at Re > 4,000.

Viscosity- Defined as the shear stress per unit area at any point in a confined fluid divided by the velocity gradient in the direction perpendicular to the direction of flow, if the ratio is constant with time at a given temperature and pressure for any species, the fluid is called a Newtonian fluid.

Water Hammer - Water hammer is the dynamic pressure surge that results from the sudden transformation of the kinetic energy in a flowing fluid into pressure when the flow is suddenly stopped. The sudden closing of a valve can cause a water hammer.

NOMENCLATURE

A	Radius-sectional area, ft^2 (m^2)
a	Sum of mechanical allowances plus corrosion allowance plus erosion allowance, in(mm)
C	Flow coefficient for the nozzles and orifices
c	Compressible factor, for perfect gas c =1.0
D	Internal diameter of pipe, ft (m)
d	Internal diameter of pipe, in
d_1	Pipe with smaller diameter in enlargements or contractions in pipes
d_2	Pipe with smaller diameter in enlargements or contractions in pipes
d_e	Equivalent hydraulic diameter, in (mm)
D_o	Outside diameter of pipe, in. (mm)
E	Weld joint efficiency or quality factor from ASME B31.3
f	Dancy's friction factor, dimensionless
f_t	Friction factor for fitting
g	Acceleration of gravity, ft/s^2 (m/s^2) – $32.2 ft/s^2$
ΔH	Surge pressure, ft-liq (m-liq)
h_L	Head loss, ft (m)
k	Ratio of specific heat at constant pressure to specific heat at constant volume = c_p/c_v

K	Resistance coefficient, dimensionless
K_1	Resistance coefficient for enlargement/contraction, dimensionless
L	Length of pipe, ft (m)
L_{eq}	Equivalent length, ft (m)
L_m	Length of pipe, miles
M	Molecular weight
P	Pressure drop in pipe, lbf/in^2(Pa)
P_i	Internal design pressure, psig (kPa gage)
Q	Volumetric flow rate, ft^3/s (m^3/s)
q	Volumetric flow rate, ft^3/hr (m^3/hr)
Q_1	Rate of flow, gal/min
R	Individual gas constant, MR =1544
R_e	Reynolds Number, dimensionless
S	Specific gravity of a liquid, dimensionless (hydrocarbon in API)
S_g	Specific gravity of a gas, dimensionless
S_m	Allowable stress, from ASME B31.3, psi (MPa)
T	Absolute temperature, R (460+°F)
Tv	Valve stroking time (s)
Te	Effective valve stroking time (s)
t	Pressure design minimum thickness, in. (mm)
t_m	Total minimum wall thickness required for pressure integrity, in. (mm)
t_{nom}	Wall thickness, in. (mm)
V	Mean velocity, ft/s (m/s)
$\overline{V}$	Specific volume, ft^3/lbm (m^3/kg)
$\overline{V_1}$	Inlet specific volume, ft^3/lb
V_{max}	The bigger velocity for enlargement / contraction, ft/s (m/s)
ΔV	Change of linear flow velocity, ft/s (m/s)
v_s	Sonic velocity, ft/s (kg/s)
W	Mass flow rate, lbm/hr (kg/hr)
w	Mass flow rate, lbm/s (kg/s)
Y	Expansion factor (dimensionless)
z	Elevation of pipe, ft (m)

Greek letters

ρ	Weight density of fluid, lbm/ft^3 (kg/m^3)
μ_e	Absolute viscosity, lbm.s /ft (kg.s/m)
μ	Absolute (dynamic) viscosity, cp
ε	Absolute roughness, in (mm)
θ	Angle of convergence or divergence in enlargements or contractions in pipes
Δ	Differential between two points

THEORY

A) General Fluid Flow Theory

I) Physical Properties of Fluids

Physical properties of fluid are important for any flow problem and the accuracy of the values will affect the flow of fluid in the pipeline.

Viscosity

A fluid viscosity can be described by its Dynamic viscosity (sometimes called Absolute viscosity), or its Kinematics viscosity. These two expressions of viscosity are not the same, but are linked via the fluid density.

Kinematics viscosity = Dynamic viscosity / fluid density

Density, Specific Volume and Specify Gravity

The weight density or specific weight of a substance is its weight per unit volume.

The specific volume $\overline{V}$ is the reciprocal of the weight density, is expressed in the SI system as the number of cubic meter of space occupied by one kilogram of the substance.

The specific gravity of a liquid is the ratio of its weight density at specified temperature to that of water at standard temperature, 60F

$$S = \frac{\rho\{\text{any liquid at specific temperature}\}}{\rho(\text{water at } 60\,\text{F})} \qquad \text{Eq (1a)}$$

For hydrocarbon like oil, the API unit is used.

$$S\ (60\ F\ /\ 60\ F) = \frac{141.5}{131.5 + \deg.API} \qquad \text{Eq(1b)}$$

Normal water deg. API unit is 10, that mean water S = 1.00

For gas the specific gravity S_g is expressed as

$$S_g = \frac{\mathrm{M(gas)}}{\mathrm{M(air)}} \qquad \text{Eq (1c)}$$

Mean Velocity

Mean velocity is the average velocity in flow across the given cross section as determined by the continuity equation for steady state flow. It normally express as ratio of the volumetric flow rate (Q) to sectional area (A) of the pipe.

$$\text{Mean Velocity, } V = \frac{Q}{A} \qquad \text{Eq (2)}$$

which,

V = mean velocity, ft/s (m/s)
Q = volumetric flow rate, ft^3/s (m^3/s)
A = radius-sectional area, ft^2 (m^2)

Which, Volumetric flow rate in the pipe line is the ratio of the mass flow rate to density of the fluid.

$$\text{Volumetric flow rate, } Q = \frac{w}{\rho} \qquad \text{Eq (3)}$$

which,

w = mass flow rate, lbm/s (kg/s)

ρ = weight density of fluid, lbm/ft^3 (kg/m^3)

and the Sectional Area in pipe formula is expressed as

$$\text{Sectional area, } A = \frac{\pi D^2}{4} \qquad \text{Eq (4)}$$

II) Flow Characteristic in Pipe

There are three different types of flow in pipe and these determine the pipe sizing. There are laminar flow, between laminar and transition zones flow, and turbulent flow. This is very important for the designer to determine the type of flow of the fluid before proceeding with the calculation.

Reynolds Number

The Reynolds Number is used to determine the nature of flow in pipe whether is the laminar flow or turbulent flow. Reynolds Number with symbol R_e, which depend with pipe diameter (D), the density (ρ) and absolute viscosity (μ) of the flowing fluid and it velocity (V) of the flow. This number is a dimensionless group with combination of these four variables which expressed as

$$R_e = \frac{DV\rho}{\mu_e} \qquad \text{Eq (5a)}$$

$$R_e = \frac{50.6\, Q_1 \rho}{d\mu} \qquad \text{Eq (5b)}$$

which,

- D = internal diameter of pipe, ft (m)
- V = mean velocity of flow, ft/s (m/s)
- ρ = weight density of fluid, lbm/ft^3 (kg/m^3)
- μ_e = absolute viscosity, lbm /ft.s (kg /m.s)
- d = internal diameter of pipe, in
- Q_1 = Mass flow, gal/min
- μ = absolute (dynamic) viscosity, cp

Fluid Flow Equations for the Friction Loss/Pressure Drop in Pipe

Bernoulli's equation is useful in the calculation of the fluid flow. It follows the first law of Thermodynamics and it calculates the energy balance in steady state and incompressible flow. The formula for the friction term in pipe line is expressed as

$$\Delta\left(\frac{P}{\rho} + z + \frac{V^2}{2g}\right) = h_L \qquad \text{Eq (6)}$$

which,

- P = pressure drop in pipe, lbf/in^2(Pa – For the SI unit remember to divide the pressure head with the acceleration of gravity.)
- z = elevation of pipe, ft (m)
- g = acceleration of gravity, ft/s^2 (m/s^2) – 32.2ft/s^2
- h_L = Head loss, ft (m)

Dancy's formula of the friction in pipe line is expressed as

$$h_L = f \frac{L}{D} . \frac{V^2}{2g} \quad \text{Eq (7)}$$

which,

f = friction factor, dimensionless
L = length of pipe, ft (m)

Dancy's friction factor, f is determined experimentally. Normally friction factor for the laminar flow conditions (Re <2100) is simple calculated with just function of the Reynolds number only, which can be expressed as

$$f = \frac{64}{R_e} \quad \text{Eq (8)}$$

In the transition zone which with the Reynolds number of approximately 2100 to 4000. In this zone, the flow is either laminar or turbulent depending upon several factors. In this zone the friction factor is indeterminate and has lower limits based on laminar flow and upper limits based on turbulent flow conditions.

For the turbulent flow with the Reynolds number > 4000, the friction factor is not only factor of the function of Reynolds number it is function of the pipe wall as well. The piping roughness will affect the friction loss as well.

Generally for the turbulent flow and transition flows the friction faction plot based on the Colebrook equation

$$\frac{1}{\sqrt{f}} = 1.14 - 2\log\left(\frac{\varepsilon}{D} + \frac{9.35}{R_e\sqrt{f}}\right) \qquad \text{Eq (9a)}$$

which,

ε = absolute roughness, in (mm)

Or the simplified formula can be written as

$$f = \frac{1}{(2\text{Log}[(\frac{7}{\text{Re}})^{0.9} + (\frac{3.24\varepsilon}{\text{d}})])^2} \qquad \text{Eq (9b)}$$

Roughness is a factor denoting the roughness of the pipe or tube; the more rough the pipe, the larger the friction factor; the larger the friction factor, the more pressure drop. This value is taken from standard table by difference piping material. Normally the value of roughness for the ‘commercial steel pipe’ is 0.00018 in.

Relative roughness of the pipe is normally calculated from the Moody Chart, which expressed as

$$\text{Relative roughness} = \frac{\varepsilon}{D} \qquad \text{Eq (10)}$$

Simplified the relative roughness is the ratio of the pipe internal roughness to internal size of diameter of pipe.

Table 1: General Pipe Material Roughness [3]

Pipe Material	Roughness , ε, in (mm)
Steel, welded and seamless	0.0002 (0.061)
Ductile Iron	0.0002 (0.061)
Ductile Iron, asphalt coated	0.0004 (0.12)
Copper and Brass	0.002 (0.61)
Glass	0.000005 (0.0015)
Thermoplastics	0.000005 (0.0015)
Drawn Tubing	0.000005 (0.0015)

Straight Line Pressure Drop

The pressure drop is expressed as below for the horizontal pipe line,

$$\Delta P = f\frac{L}{D}.\frac{V^2}{2g}.\rho \qquad \text{Eq (11)}$$

Common form of the Darcy Weisbach equation in units of pounds per square inch (psi) is:

$$\Delta P = 0.00000336\frac{fLW^2}{\rho d^5} \qquad \text{Eq (12)}$$

To obtain pressure drop in units of psi/100 ft, the value of 100 replaces L in Equation 12.

In the non-horizontal pipe line pressure drop is expressed,

$$(\Delta P)_e = \rho h_L = \rho(\Delta z) \qquad \text{Eq (13)}$$

For the velocity change the in pressure drop the formula is expressed as

$$(\Delta P)_k = \rho h_L = \frac{\rho \Delta V^2}{2g} \qquad \text{Eq (14)}$$

Effect of Valve, Fitting on Pressure Drop

In the fluid systems the effect of valves, elbows, and etc on the pressure drop is needed to be taken into consideration when designing.

General pressure drop in the fitting expressed as formula for the laminar flow and turbulent flow.

$$\frac{\Delta P_{ft}}{\rho} = K\left(\frac{V^2}{2g}\right) \qquad \text{Eq (15)}$$

Which

K = resistance coefficient, dimensionless
ΔP_{ft} = Pressure Drop in specific fitting, lbf/in^2 (Pa)

$$K = f_t \frac{L_{eq}}{D} \qquad \text{Eq (16)}$$

Table 2a: Example of the equivalent lengths for various kinds of fittings [1]

Type of Fitting	Equivalent Length L_{eq}/D (dimensionless)
Globe valve, wide open	340
Angle valve, wide open	150
Gate valve, wide open	8
Check valve (swing type) (minimum pipe velocity for full disc lift $=35\sqrt{\bar{V}}$)	100
90º standard elbow	30
45º standard elbow	16
90º long-radius elbow	20
Flow thru run standard tees	20
Flow thru branch standard tees	60

Table 2b: Friction factor for the commercial steel pipe [1]

Nominal Size	½ “	¾ “	1”	1¼“	1½”	2”	2½” - 3”	4”	5”	6”	8-10”	12-16”	18-24”
Friction Factor (f_t)	.027	.025	.023	.022	.021	.019	.018	.017	.016	.015	.014	.013	.012

Enlargements and Contraction Pipe Line Pressure Drops Calculation

When the fluid flow from a smaller diameter pipe goes into a bigger diameter pipe, it called enlargement, and vice-visa it called sudden contraction. Generally these processes will cause a friction loss and the changes in the kinetic energy. The pressure change can be expressed in the pressure drop formulation as below.

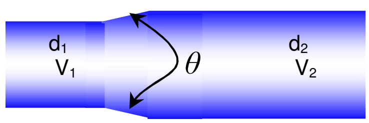


Figure 1a: Sudden/ Gradually Enlargement

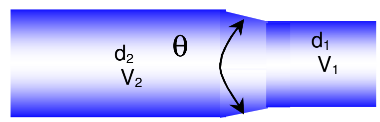


Figure 1b: Sudden/ Gradually Contraction

Friction loss with expressed in pressure drop as below,

$$\Delta P = K_1 \frac{\rho V_{max}^{\ 2}}{2g} \qquad \text{Eq (17)}$$

which,

K_1 = resistance coefficient for enlargement/contraction, dimensionless
V_{max} = the bigger velocity for enlargement / contraction, ft/s (m/s)

For sudden and gradual enlargement

$\theta \leq 45^o$ $$K_1 = \frac{2.6\ sin\frac{\theta}{2}\left(1 - \frac{d_1^{\ 2}}{d_2^{\ 2}}\right)^2}{\left(\frac{d_1}{d_2}\right)^4}$$ Eq (18a)

$45^o \langle \theta \leq 180^O$ $$K_1 = \frac{\left(1 - \frac{d_1^{\ 2}}{d_2^{\ 2}}\right)^2}{\left(\frac{d_1}{d_2}\right)^4}$$ Eq (18b)

For sudden and gradual contraction

$\theta \leq 45^o$ $$K_1 = \frac{0.8\ sin\frac{\theta}{2}\left(1 - \frac{d_1^{\ 2}}{d_2^{\ 2}}\right)^2}{\left(\frac{d_1}{d_2}\right)^4}$$ Eq (19a)

45 $^o < \theta \leq 180^O$ $$K_1 = \frac{0.5\sqrt{sin\frac{\theta}{2}}\left(1 - \frac{d_1^{\ 2}}{d_2^{\ 2}}\right)^2}{\left(\frac{d_1}{d_2}\right)^4}$$ Eq (19b)

Nozzles and Orifices

Nozzles and orifices normally used in piping systems as metering devices and are installed with flange taps or pipe taps accordance with ASME standards [1]. For the flow through the nozzles or orifices the velocity of the flows for the incompressible fluid and compressible fluid are expressed respectively in Eq (20a) and Eq (20b). Both formulas incorporated with the flow coefficient, C.

$$q = CA\sqrt{\frac{2g(144)\Delta P}{\rho}} \qquad \text{Eq (20a)}$$

$$q = YCA\sqrt{\frac{2g(144)\Delta P}{\rho}} \qquad \text{Eq (20b)}$$

which,

C = flow coefficient for the nozzles and orifices
Y = net expansion factor for compressible flow

Water Hammer

Water Hammer or dynamic pressure surge can caused pipe to jump off its supports, damaged anchors and restraints, and resulted in leaks and shutdowns in process plants and terminal facilities.

To minimize hydraulic surge in piping systems, first, the conflicting requirements for surge force minimization and allowance for pipe differential thermal expansion must be balanced; to minimize the impact of unbalanced surge forces, designs would tend to have rigidly supported piping and a minimum number of bends; designing for differential thermal expansion would lead to minimizing supports and providing bends for flexibility. Second, the total energy in the system should be minimized. Pumps, which are the energy source, should not be over designed.

General design recommendations for minimizing surge forces are:

- Minimize the number of bends used in the system.
- Use the largest enclosed angle and radius possible for bends.
- Bend to bend distance should be as far as possible.
- Provide supports near all large components.
- Provide bypasses at pump stations.
- Install control valves on pump discharges.
- Limit flow velocity to a maximum of 10 ft/sec (3 m/s) in pipes, and 35 ft/sec (10 m/s) in loading arms.

General rule, closure times of valves in pipes up to 24 in. (600 mm) in diameter should exceed 15 seconds. For pipe diameters of 24 in. (600 mm) or larger, the closure time should be at least 30 seconds. Valve operators of the air pneumatic piston type should be avoided, because they may cause pressure surges due to sudden closing of valves. If surge forces are unavoidable, protection devices should be used, including safety valves or rupture disks for tube split protection on high pressure exchangers, pulsation bottle for positive displacement pumps, LPG line surge drums, etc.

B) Piping Fluid Flow Material Selection

Many factors have to be considered when selecting engineering materials for the piping. Normally the material selections of piping it depend on the application, refer Table 3.

The most economical material that satisfies both process and mechanical requirements should be selected; this will be the material that gives the lowest cost over the working life of the plant, allowing for maintenance and replacement for the piping [3].

Table 3: Guideline for the Piping Fitting and Pipe Material Selection.

Type of Material & Features	General Applications
Stainless Steel — A corrosion-resistant material that provides high strength at high temperatures, helps prevents contamination of product being transported, maintains cleanliness, and retains a lustrous appearance. It's harder than brass. *Type 304 stainless steel* is a low-carbon chromium-nickel stainless steel. *Type 316 stainless steel* is similar to Type 304, but has a higher nickel content as well as molybdenum for stronger resistance to heat and corrosion.	For use with water, oil, and gas. Good for chemical, pulp, and paper processing as well as for oil refining and pollution-control equipment. It's sanitary and no contaminating which also makes it a good choice for use in pharmaceutical, dairy, brewery, beverage, and food industries.
Brass — This soft, copper-based metal provides tight seals and is easier to install than other metals. It can be used interchangeably with copper where heavier walls are required. It resists corrosion from salt water as well as fresh water polluted with waste from mineral acids and peaty soils.	Primarily used with water for plumbing and heating. Also good for use in pneumatic and marine applications.
Aluminum — Lightweight and strong, this metal is ductile and malleable. It can be anodized for better corrosion resistance.	For low-pressure systems with water-based fluids in agricultural and some food-processing applications.
Iron — *Cast iron* is a harder, more brittle iron while *malleable iron* is a softer, more ductile iron.	*Cast Iron:* For use with water (heating and cooling) and steam. Good for fire protection applications. *Malleable Iron:* For use with gas, oil, and water. Good for industrial plumbing.
Steel — This carbon- and iron-based metal is hard and strong. Commonly used steel pipe ratings are Schedule 40 (standard) and Schedule 80 (extra strong) [5].	For high-pressure systems in petrochemical, oil refinery, hydraulic, and pneumatic applications.

Once the material of construction is selected, the selected piping it should be selected based on piping schedule, pipe diameter (inside or outside diameter) and pipe wall thickness which cover under the standards codes. Below are the standards codes covers:

1. American Standard ASME B36.10
2. American Standard ASME B36.19
3. New United States Legal Standard for Steel Plate Gauges

From the above standard codes, wall thickness of pipe can be calculated as using the formula below with subject to internal pressure:

$$t_{nom} = \frac{t_m}{0.875} \qquad \text{Eq (26a \& 26b)}$$

$$t_m = t + a$$

$$t = \frac{PD_o}{2S_m E} \qquad \text{Eq (26c)}$$

which,

t_{nom} =Wall thickness, in. (mm)

t_m =Total minimum wall thickness required for pressure integrity, in. (mm). Most piping specifications allow the manufacturer a 12.5% dimensional tolerance on the wall thickness; the minimum wall thickness can be as low as 87.5% of the nominal value. Therefore, in selecting the pipe schedule, tm should be divided by 0.875.

t = Pressure design minimum thickness, in. (mm)

a = Sum of mechanical allowances plus corrosion allowance plus erosion allowance, in. (mm)

P = Internal design pressure, psig (kPa gage)

D_o = Outside diameter of pipe, in. (mm)

S_m = Allowable stress, from ASME B31.3, psi (MPa)

E = Weld joint efficiency or quality factor from ASME B31.3. Example seamless pipe, E = 1.0

C) Line Sizing

Typical pipe velocities and allowable pressure drops, which normally use to select the pipe sizes used, are given as Table 4 below;

Table 4: Pipe velocities and allowable pressure drops for various fluids [3]

Type of Fluid	Velocity, ft/sec	ΔP in psi/100ft
Liquid, pumped (not viscous)	3.3 - 10	2.2
Liquid, gravity flow	-	0.22
Gases and vapor	50-100	0.02% of line pressure
High-Pressure steam >8 bar	100-200	-

Table 5: Optimum velocity for various fluid densities [3]

Fluid density lb/ft^3	Velocity, ft/sec
100	7.9
50	9.8
10	16.0
1	30.8
0.01	59.0
0.001	111.5

For gases and vapors the velocity cannot exceed the critical velocity (sonic velocity) and normally will be limited to 30% of the critical velocity. Then exit velocity can be calculated by

$$V_{ex} = \left[\frac{2 \times \left(g \times \Delta p_{tot} + 0.5 \times \rho \times V^2 \right)}{\rho} \right]^{0.5}$$

I) Design Procedure for In-Compressible Flow

Step 1: Determine the internal diameter of pipe. For the non-circular cross sections, equivalent hydraulic diameter has to determine with the following formula:

$$\mathrm{d_e} = 4 \left[\frac{\text{cross sectional area}}{\text{wetted perimeter}} \right] \qquad \text{Eq (27)}$$

The d_e will be replaced with D for calculation of Reynolds Number, frictional pressure drop but cannot use for velocity calculation.

Step 2: Calculate the Reynolds number with Eq (5a) to determine the flow characteristic either is laminar flow, transition flow or turbulent flow for the next step calculation of the friction factor, f (Eq (8); Eq (9a) or Eq (9b) depend on type of flow and Re number).

Step 3: For the fitting and valve use the Table 2a & 2b to find the K values; and enlargement or contraction find the all the K values of them and use the Eq (18a /18b/19a/19b). As well for the pipe line use formula K=fL/D. Sum all the K in the system.

Step 4: Calculated the pressure drop for horizontal line with Eq (11) and non-horizontal line Eq (13) if the system involves the velocity change calculated the kinetic energy pressure drop with Eq (14). Sum of the pressure drop will be total pressure drop of the pipe.

Step 5: By referring to Table 4 and Table 6, compare the calculated and determine whether the pressure drop in line is suitable or not. At the same time make sure the velocity design is in the range. If the pressure drop in the system is too big, trial again with the bigger number of internal diameter of pipe.

Table 6: Reasonable Velocities for flow of water/fluid with almost same density through pipe [1]

Service Condition	Reasonable Velocity
Boiler Feed	8 to 15 feet per second
Pump Suction and Drain Lines	4 to 7 feet per second
General Service	4 to 10 feet per second

II) Design Procedure for Compressible Flow

In compressible flow an accurate determination of pressure drop through a pipe is depending relationship between pressure and specific volume. The have adiabatic flow ($p'V_a^k$=constant) and isothermal flow ($p'V_a$=constant). In adiabatic flow system, usually assumed the flow is short and the insulated pipe is insulated perfectly, means that no heat is transferred to or from the pipe except for the minute amount of heat generated by friction which is added to the flow.

Isothermal flow normally assumed as with constant temperature. Normally gas flow in insulated pipe is closely approximated by isothermal flow for reasonably high pressures.

When the pressure drop in pipe is very great the density and velocity will change appreciably, that mean the calculation should take in consideration of the matter when design piping for compressible flow.

Density of gas can be determine with the formula as below

$$\rho = \frac{MP}{cRT} \qquad \text{Eq (28)}$$

which,

M = molecular weight
P = pressure gas
c = compressible factor, for perfect gas c =1.0
R = individual gas constant
T = temperature of the gas

When using the Darcy formula for the compressible fluids calculation; restrictions should be followed:

1. If pressure drop less than 10% of the inlet pressure, reasonable accuracy will be obtained if the specific volume used is based either the upstream or downstream conditions.

2. If the pressure drop is greater than 10% and less than 40% of the inlet pressure, the Darcy equation may be used with reasonable accuracy with specific volume based upon the average of upstream and downstream conditions.

3. If the pressure drop is greater than 40% than the formula below should be use for calculation instead of Dancy's formula.

i) Isothermal gas flow with the formula

$$w^2 = \left[\frac{144gA^2}{\overline{V}_1\left(\frac{fL}{D} + 2\,ln\frac{P_1}{P_2}\right)}\right]\left[\frac{(P_1)^2 - (P_2)^2}{P_1}\right] \qquad \text{Eq (29)}$$

which,

w = mass flow rate, lb/s
A = cross sectional area in pipe, ft^2
g = acceleration of gravity, 32.2 ft /s^2

Assumptions make during developed on this formula

a) No mechanical work is done on / by the system
b) Steady flow
c) Gas obey the perfect gas law
d) Velocity may be represented by the average velocity at a cross section
e) The friction factor is constant along the pipe
f) The pipe line is straight and horizontal between end points.
g) Acceleration can be neglected because the pipe line is long.

ii) Simplified Compressible flow formula

$$w^2 = \left[\frac{144gDA^2}{\overline{V}_1 fL}\right]\left[\frac{(P_1)^2 - (P_2)^2}{P_1}\right] \qquad \text{Eq (30)}$$

or can be expressed in the volumetric flow rate format as

$$q = 114.2\sqrt{\left[\frac{(P_1)^2 - (P_2)^2}{fL_mTS_g}\right]d^5} \qquad \text{Eq (31)}$$

which,

L_m = Length of pipe, miles
T = Absolute temperature, R
S_g = Specific gravity (ratio of molecular weight of the gas to air)

The sonic velocity for a compressible fluid in a pipe is equivalent to the speed of sound in the fluid; which can be expressed in ft/s as

$$v_s = \sqrt{kgRT} \qquad \text{Eq (32)}$$

which,

k = ratio of specific heat at constant pressure to specific heat at constant volume = c_p/c_v
g = acceleration of gravity, ft/s^2
R = individual gas constant: 1544/Molecular weight
T = absolute temperature, R

The Darcy formula for mass flow rate calculation with Y expansion factor for the compressible fluid is expressed as

$$w = 0.525Yd^2\sqrt{\frac{\Delta P}{K\overline{V_1}}} \qquad \text{Eq (33)}$$

which,

Y = expansion factor (dimensionless)
$\overline{V_1}$ = inlet specific volume, ft^3/lb

Table 7: Reasonable Velocities for flow of steam through pipe [1]

Condition of Steam	Pressure (P) psig	Service	Reasonable velocity V, ft/sec
Saturated	0 to 25	Heating (short lines)	67 to 100
	25 and up	Power house equipment, process piping, etc.	100 to 167
Superheated	200 and up	Boiler and turbine leads, etc.	117 to 3333

D) Pump Suction Piping

In some pumping systems the pump is located at the source of the liquid, and the entire piping system is on discharge side of the pump. In most cases, some portion of the piping system is on the suction side if the pump. The pump can only perform properly if it is supplied with a steady flow of liquid arriving at the pump suction flange under sufficient absolute pressure to equal or exceed the Net Positive Suction Head (NPSH) required by the pump and with uniform velocity with no rotational component

The failure of the suction piping to deliver the liquid to the pump in this condition can lead to noisy operation, random axial oscillations of the rotor, premature bearing, and cavitations damage to the impeller and inlet portions of the casing. These events occurring increase with pump size and suction specific speed, in other words decreasing NPSHR value. When troubles do arise, remedial action can be costly, if not impossible, because the only fic in most cases is a major revision of the piping arrangement.

The root cause of many pump problems and failures can be traced to poor upstream, suction-side, pipeline design. Common problems to avoid are:

1. Insufficient fluid pressure leading to cavitation within the pump.
2. Narrow pipes and constrictions producing noise, turbulence and friction losses.
3. Air or vapour entrainment causing noise, friction and loss of performance.
4. Suspended solids resulting in increased erosion.
5. Poor installation of pipework and other components.

Therefore, it is important to avoid the problem in the first place.

1. Adequate priming or venting is required. In modern piping systems, foot valves are only infrequently used and each pump must therefore be primed before start-up. The point of connection between the priming device and the pump should be at the highest point on the casing, which will be above the pump suction flange.

2. Reducers for lower velocity. Reducers are frequently installed just ahead of the pump suction in order to permit lower suction pipe velocities and therefore, lower friction losses than would otherwise occur. When the liquid source is below the pumps, the reducer should be eccentric and should be installed with the flat side up. For end suction pumps, additional cautions about the application of reducers must be observed because of the close proximity of the impeller inlet to the suction flange. The following precautionary guidelines should be considered.

    - Limit reducers at the pump suction to a change in diameter of one pipe size, such as 12' x 10' or 24' x 20'.
    - Where suction lines larger than one size over the suction flange must be used, two or more standard reducers may be installed in series, or a specially fabricated reducer with low convergence could be used (10º maximum total in angle).
    - When the source of the liquid is above the pump, concentric reducers are preferred for end suction pumps.

3. Piping velocities. These velocities should generally not exceed the value which exists at the pump suction flange. In the simplest of systems, where the inlet pipe is a straight line between the liquid source and the pump suction, the velocity in the pipe itself can be the same as at the pump suction, provided the line losses do not preclude the availability of adequate NPSH at the pump suction.

   - For most industrial centrifugal pump designs, suction flange velocities will vary between approximately 8 ft/s and 15 ft/s.
   - The standard reducer between any two consecutive standard pipe sizes (10 in and 30 in) will reduce these values to ranges of approximately at 4.5 to 5.5 ft/s and 8.4 to 10.4 ft/s. below 10 in, a one size reduction in diameter may affect a greater reduction of velocity. Above 30 in the effect will be less.
   - At a suction line velocity of 5 ft/s, a straight run of pipe equal to 5 pipe diameters should be adequate to rectify irregularities in the velocity profile which result from a 90 degree change in flow direction through an elbow or tee.
   - At a suction line velocity of 10 ft/s, the straight section will probably have to be at least 10 diameters in length.
   - It should be noted that pump suction velocities for saturated liquids are recommended to be in a normal range of three to 5 ft/sec with a maximum of 7 ft/sec. At the maximum velocity the total acceleration loss is 0.8 ft of fluid (0.3 psi for water).

The following requirements are recommended for designing suction piping

1. Piping should be as short and direct as possible with a minimum number of elbows and fittings;

2. Piping should be one or two sizes larger than the pump suction connection;

3. Maintain a velocity and pressure drop in accordance with the Guidelines for Sizing Liquid Lines Table;

4. Friction losses based on rated flow;

5. Piping should contain a minimum number of turns and necessary turns should be accomplished with long-radius elbows or laterals;

6. Piping lead should be one size smaller than the header size, if possible; and

7. Piping should be designed to preclude the collection of vapor in the piping (no high points unless vented).

8. Eccentric reducers should be used near the pump, with the flat side up to keep the top of line level. This eliminates the possibility of gas pockets being formed in the suction piping. If potential for accumulation of debris is a concern, means for removal is recommended.
9. For reciprocating pumps provide a suitable pulsation dampener (or make provisions for adding a dampener at a later date) as close to the pump cylinder as possible.
10. In multi-pump installations, size the common feed line so the velocity will be as close as possible to the velocity in the laterals going to the individual pumps. This will avoid velocity changes and thereby minimize acceleration head effects.

Special Considerations

1. In reference to Pump NPSH Determination, use Vessel Elevation and Pump NPSH Calculation Form (Part A) for pump suction pressure and NPSH calculations.

2. For a circuit going through the Battery Limits to offsite, check for the Battery limit pressure required or establish the battery limit pressure by a point to point calculation to the offsite piece of equipment.

3. Where the process requirements consist of alternate cases or multiple destinations, each must be analyzed individually and a controlling case selected that determines the highest differential.

4. When selecting the control valve in the discharge circuit, adequate pressure drop should be allowed to ensure that the control valve will perform in an acceptable range. In this way the desired Cvc/Cv ratio will be achieved for all operating cases. Minimum acceptable control valve pressure drop is 10 psi at rated flow conditions or 33% of system variable loss at normal flow conditions, whichever is greater.

5. Be sure to check the pressure losses due to equipment internals at the pump circuit terminal point. In some instances strainers or filters are present in the inlet to the equipment, this can add a significant pressure drop. This pressure drop should be considered in the pump calculation.

6. Not in all instances can a pump be chosen that will match the flow and the pump differential pressure specified by Fluid Systems. For instances of low flow and high head, extremely high flow, or any unusual flow scheme the Rotating Engineer should be consulted. A quick inquiry with the Rotating Equipment Engineer can be done to determine whether a pump does or does not exist for the flow and differential pressure required by the circuit. Consulting with the Rotating Equipment Engineer can save time reissuing data.

7. For calculations performed on existing pumps the existing pump curve should be checked to ensure that the existing pump can handle the new flow and the discharge head with the existing size, otherwise different size impellers which fit into the specific pump casing could be considered.

NPSH is simply a measure of the amount of head present to prevent this vaporization at the lowest pressure point in the pump. The NPSH required (NPSHr) depends on the pump design. As the liquid passes from the pump suction to the eye of the impeller, the velocity increases and pressure decreases. The NPSHr is the positive head (in feet absolute) required at the pump suction to overcome these pressure drops in the pump and maintain the liquid above its vapor pressure. NPSHr generally increases with increasing flow rate in a given pump, this is because higher velocities occur within the pump leding to lower pressures. NPSHr is usually higher for larger pumps, meaning that cavitation can be more of a problem in larger pump sizes.

Available NPSH (NPSHa) is a characteristic of the system in which the pump operates. It is the excess pressure of the liquid (expressed in feet of head) over its vapor pressure as it arrives at the pump suction. In an existing system, the NPSHa can be calculated with a vacuum gauge reading on the pump suction line by:.

$$NPSHa = PB - VP \pm Hp + HV$$

Where,

PB = Barometric pressure (ft)
VP = Vapor pressure of water at maximum pumping temperature (ft)
Hp = Pressure head at the pump suction (ft). (Negative value for vacuum and positive if flooded suction) corrected to the elevation of the pump centerline
Hv = Velocity head in the suction pipe (feet) = $V^2/2g$
V = velocity of water (ft/s)
g = the acceleration of gravity = 32.2 ft / s^2

The work performed in pumping a fluid will depend on the volume flow rate, the density of the fluid, the additional head to be added to the fluid pressure and the efficiency of the pump. Power required (Hp)

$$Pwr = \frac{GPM \times NPSHa \times \rho_L \times 231}{33000 \times 1728 \times E}$$

Table 8 . Vapor pressure (VP) versus temperature for water.

Temperature F	Vapor Pressure	
	psi	Ft.H2O
40	0.12	0.3
50	0.18	0.4
60	0.26	0.6
70	0.36	0.8
80	0.51	1.2
90	0.70	1.6
100	0.94	2.2
110	1.28	2.9
120	1.70	3.9

Relationship between NPSHr and NPSHa

- If NPSHr < NPSHa, there should be no cavitation
- If NPSHr > NPSHa, cavitation is impending
- As the NPSH drops below the required value, cavitation will become stronger, the pump efficiency wil drop, and the flow rate will decrease
- At some point, the pump would “ break suction” and the flow rate would go to zero (even with the pump still operating)

Cavitation often occurs in pumps, hydroelectric turbines, pipe valves, and ship propellers. Cavitation is a problem because of the energy released when the bubbles collapse, formation and subsequent collapse can take place in only a few thousands of a second, causing local pressures in excess of 150,000 psi and local speeds over 1000 kph. The collapse of the bubbles has also been experimentally shown to emit small flashes of light upon implosion, followed by rapid expansion on shock waves. Potential problems is

1. Noise and vibration,
2. reduced efficiency in pumps,
3. reduced flow rate and head in pumps,
4. physical damage to impellers, volute case, piping, valves.

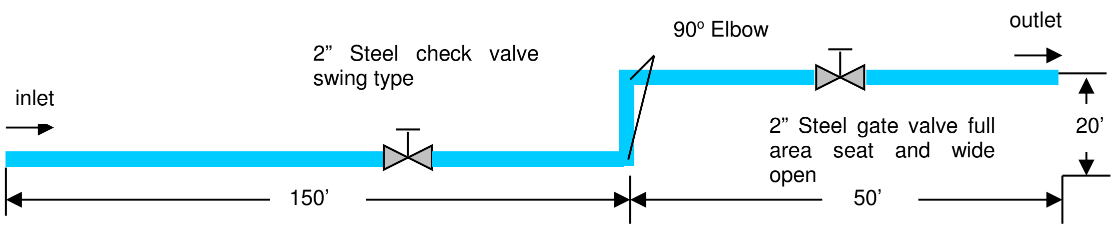
The most common reasons for cavitation are:

1. Suction-side supply blockage: Debris that blocks the pump suction intake will restrict the amount of water needed to operate at peak efficiency. Air leaks that develop in the suction line will also restrict flow of water into the pump.
2. Poor system design: Improper intake design or lack of attention to NPSH requirements may lead to pump cavitation.
3. Drop in water level: Water levels dropping below the design depth may result in insufficient NPSH available in the system.
4. High temperature combined with marginal suction supply: Partial blockage of intake screens and unusual swings in atmospheric conditions can result in cavitation occurring.
5. Reduced system demands: As discharge lines in the system corrode or plug, pump discharge output is restricted and discharge cavitation can occur. Check valves that may not operate properly on either the pump discharge or suction side as they also can cause cavitation.
6. The pump is oversized: Oversizing the pump may occur because of failure to conduct a detailed system analysis to determine the proper head pressure and flows required for the application or the tendency to “fudge” the numbers to be “safe.”

APPLICATION

Example Case 1: In-Compressible flow with Water

Water at 80F with weight density 62.22 lb/ft^3 , viscosity 0.85 cp is flowing through 2" carbon steel Schedule 40 pipe with internal diameter 2.067 in (data from ASME B36.10 and B36.19) at flow rate of 100 gallons per minute in the system as per below;



Find the velocity in ft/s and the pressure drop from the inlet through outlet in lb/in^2 and pressure drop in psig/100ft.

Solution:

Water flows in circular pipe then the d in the all formulas take it as internal diameter.

Part 1:

Velocity can be determined with Eq (2)

$$V = \frac{Q}{A} = \frac{0.408 Q_1}{d^2}$$

$$= \frac{0.408x100}{2.067^2}$$

V **= 9.55 ft/s**

Part 2:

Determine the Reynolds number with Eq (5b)

$$Re = \frac{50.6\, Q_1 \rho}{d\mu}$$

$$= \frac{50.6 x 100 x 62.22}{2.067 x 0.85}$$

$= 1.8x10^5 \quad > 2100$; turbulence flow

Because of the turbulence flow and the imperial friction factor Eq (9b) is used for calculation

$$f = \frac{1}{\left(2 \text{Log}\left[\left(\frac{7}{Re}\right)^{0.9} + \left(\frac{3.24\varepsilon}{d}\right)\right]\right)^2}$$

take , ε as 0.00018 in

f = 0.021

Valve and fitting resistance coefficient K determination with Eq (17)

$$K = f_t \frac{L_{eq}}{D}$$

By referring to the Table 2a for the

Gate valve the $L_{eq}/D = 8$; $K = 8f_t$

Check valve the $L_{eq}/D = 100$; $K = 100\ f_t$

2X 90° Elbow the $L_{eq}/D = 30$; $K = 30\ f_t$

f_t for the 2" pipe from Table 2b is $f_t = 0.019$

K = 8(0.019) = 0.152 ………gate valve
K = 100(0.019) = 1.900 ………check valve
K = 30 (0.019) = 0.570 ……… 90° elbow

And for the pipe

K = f L/D

$$= \frac{(0.021)(150+50+20)\,(12)}{2.067}$$

= 26.82

Total of the resistance coefficient of the system, K = 0.152+1.9+ (2x0.570) +26.82
= 30.01

Pressure of the system = Pressure drop in horizontal line + Pressure drop in elevation

$$= \frac{30.01(9.55)^2}{2x144x32.2}x62.22 \quad + \frac{20x62.22}{144}$$

= (18.36 + 8.64)lb/in^2

= **27.00 lb/in^2**

$$\text{Pressure drop in psi/100ft} = \frac{(0.021)(100)(9.55)^2(12)(62.22)}{2(144)(32.2)(2.067)}$$

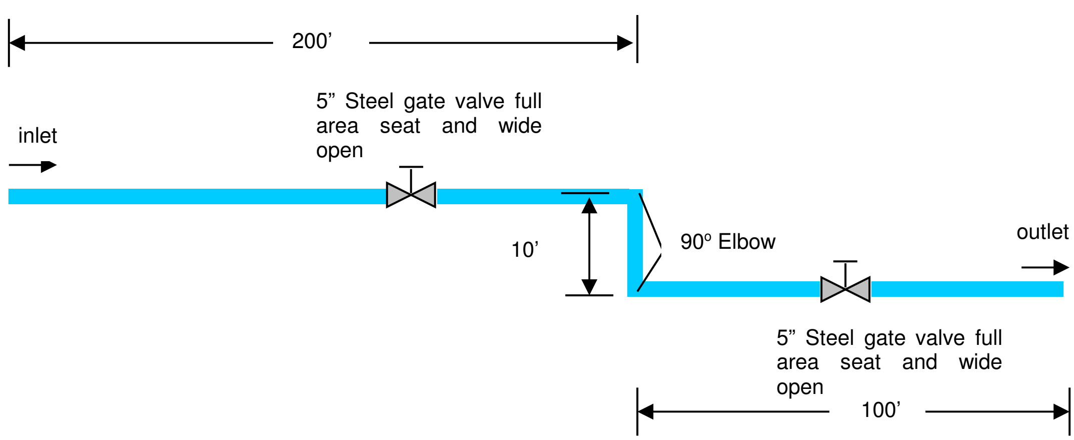
= 7.46 psi/100ft

Exit velocity

$$V_{ex} = \left[\frac{2\times\left(g\times\Delta p_{tot} + 0.5\times\rho\times V^2\right)}{\rho}\right]^{0.5} = \left[\frac{2\times\left(32.174\times 27 + 0.5\times 62.22\times 9.55^2\right)}{62.22}\right]^{0.5} = 10.93\text{ ft/s}$$

Example Case 2: In-Compressible flow with HC

Fuel 5 (Min.) with flow rate 500 gal/min, flowing through 5" Schedule 40 pipe with internal diameter 5.047 for the system as below. The temperature of the fuel oil is 100 F which weight density 59.25 lb/ft^3, viscosity 7cp.



Find velocity in ft/s of the flow and the pressure drop in system from inlet to outlet and in format psi/100ft.

Solution:

Fuel 5 Min. flows in circular pipe then the d in the all formulas takes it as internal diameter.

Part 1:

Velocity can be determined with Eq (2)

$$V = \frac{Q}{A} = \frac{0.408Q_1}{d^2}$$

$$= \frac{0.408x500}{5.047^2}$$

V **= 8.00 ft/s**

Part 2:

Determine the Reynolds number with Eq (5b)

$$R_e = \frac{50.6\, Q_1 \rho}{d\mu}$$

$$= \frac{50.6x500x59.25}{5.047x7.00}$$

$= 4.2x10^4 \;\; > 2100$; turbulence flow

Because of the turbulence flow and the imperial friction factor Eq (9b) is used for calculation

$$f = \frac{1}{(2Log[(\frac{7}{Re})^{0.9} + (\frac{3.24\varepsilon}{d})])^2}$$

take , ε as 0.00018 in

$$f = 0.023$$

Valve and fitting resistance coefficient K determination with Eq (17)

$$K = f_t \frac{L_{eq}}{D}$$

By referring to the Table 2a for the

2XGate valve the $L_{eq}/D = 8$; $K = 8f_t$
2X 90° Elbow the $L_{eq}/D = 30$; $K = 30\ f_t$

f_t for the 5" pipe from Table 2b is $f_t = 0.016$

K = 8(0.016) = 0.128 ………gate valve
K = 30 (0.016) = 0.480 ……… 90° elbow

And for the pipe

$$K = f\ L/D$$

$$= \frac{(0.023)(200+10+100)\,(12)}{5.047}$$

$$= 16.96$$

Total of the resistance coefficient of the system, K = (2x0.128)+ (2x0.480) +16.96
=18.18

Pressure drop of the system = Pressure drop in horizontal line - Pressure gain in elevation

$$= \frac{18.18(8.00)^2}{2x144x32.2} x59.25 \; - \; \frac{10x59.25}{144}$$

= (7.43 – 4.11) lb/in^2

= **3.32 lb/in^2**

Pressure drop in psi/100ft $= \dfrac{(0.023)(100)(8.00)^2(12)(59.25)}{2(144)(32.2)(5.047)}$

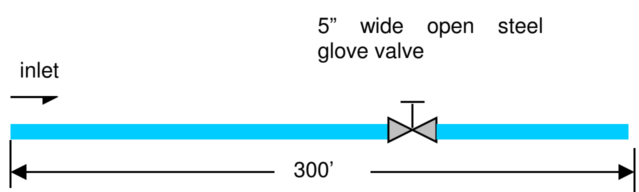
= 2.23 psi/100ft

Exit velocity

$$V_{ex} = \left[\frac{2\times\left(g\times\Delta p_{tot} + 0.5\times\rho\times V^2\right)}{\rho}\right]^{0.5} = \left[\frac{2\times\left(32.174\times3.32 + 0.5\times59.25\times8^2\right)}{59.25}\right]^{0.5} = 8.24 \text{ ft/s}$$

Example Case 3: Compressible flow with Steam

Steam at 500 psia , 600 F flows through a 300 ft horizontal pipe with a 5” Schedule 40 pipe at a rate of 90,000 lb/hr with viscosity 0.021 and the specific volume, 1.1584 ft^3/lb (density = 0.82 lb/ft3). The system consists of a wide open 5” steel glove valve.



Find the pressure drop through system in lb/in², pressure drop psi/100ft and the velocity in ft/s.

Solution:

The pressure drop in system can be determine with Eq (12)

$$\Delta P = 0.00000336 \frac{fLW^2}{\rho d^5}$$

K = fL/D (which D is diameter in ft; d is diameter in inches)

So the formula can be expressed as

$$\Delta P = 0.00000336 \frac{KW^2}{12\rho d^4}$$

$$= \frac{28x10^{-8} KW^2}{\rho d^4}$$

Resistance coefficient for the wide open glove valve from the Table 2(a) is

L_{eq}/D =340 ; K = 340f_t (f_t = 0.016 for 5" valve)

= 340(0.016)

= 5.44 …………….. Glove valve

Reynolds number can be determine with Eq (5b)

$$R_e = \frac{50.6\, Q_1 \rho}{d\mu}$$ Q_1 is in 1.0 gal/min = 8.02 ft³/hr

$$R_e = \frac{50.6\,W}{8.02 d\mu} \quad or \quad Re = \frac{6.31\,W}{d\mu}$$

$$= \frac{6.31x90000}{5.047x0.021}$$

$$= 5.35x10^6$$

Because of the turbulence flow and the imperial friction factor calculated with Eq (9b) is

f =0.0157

for the pipe

K = f L/D

$$= \frac{(0.0157)(300)\,(12)}{5.047}$$

= 11.20

Total of resistance coefficient in the system K = 5.44+11.20 = 16.64

Pressure drop in the system, $P = \dfrac{28x10^{-8}(16.64)(90000)^2(1.1584)}{(5.047)^4}$

= 67.4 lb/in^2

Pressure drop in psi/100ft = $\dfrac{33.6x10^{-7}(0.0157)(100)(90000)^2(1.1584)}{(5.047)^5}$

= 15.12 psi/100ft

Velocity of the flow, V = $\dfrac{4W\overline{V}}{3.142D^2} = \dfrac{90000(1.1584)(144)x4}{3.142(5.047)^2(3600)}$

= **<u>208 ft/s</u>**

Exit velocity

$$V_{ex} = \left[\frac{2\times\left(g\times\Delta p_{tot} + 0.5\times\rho\times V^2\right)}{\rho}\right]^{0.5} = \left[\frac{2\times\left(32.174\times 67.4 + 0.5\times 0.863\times 208^2\right)}{0.863}\right]^{0.5} = 220.11\,\text{ft/s}$$

Example Case 4: Compressible flow with Natural Gas

A natural gas flowing in a horizontal pipe line made of 10-in (internal diameter =10.25 in) Schedule 20 pipe with 120 miles long. The inlet pressure is 1200 psia and the outlet pressure is 250 psia, in a isothermal system with temperature 60 F (viscosity =0.011cp). The gas consist of 70% methane (CH_4), 25% ethane (C_2H_6), and 5% propane (C3H8)

Find the volumetric flow of in million of standard cubic feet per hr.

<u>Solution:</u> The flow rate of the nature gas in the system above can be determined with the simplifier Eq (31)

$$q = 114.2\sqrt{\left[\frac{(P_1)^2 - (P_2)^2}{fL_m TS_g}\right] d^5}$$

d= 10.25 in

Because the in the formulation there have two unknown (q and f), trial and error method is use to get the correct number of the flow rate.

At first, f is assumed for turbulent flow with 0.013

T in the Rankin = 460 + F

= 460 + 60 = 520 R

Molecular weights for

- Methane = 16
- Ethane = 30
- Propane = 44

Natural gas molecular weight = (0.70x16) + (0.25x30) + (0.05x44)
= 20.9

From Eq (1c) $S_g = \frac{M(gas)}{M(air)} = \frac{20.9}{29} = 0.720$

so, $q = 114.2 \sqrt{\left[\frac{(1200)^2 - (250)^2}{0.013x120x520x0.720}\right](10.25)^5}$

$= 1.87 \times 10^6$ ft^3/hr

Reynolds number with Eq (5b)

$$R_e = \frac{50.6\, Q_1 \rho}{d\mu}$$

Can be expressed in q and S_g,

$$Re = \frac{0.482 q S_g}{d\mu}$$

$$= \frac{0.482\,(1.87x10^6)(0.720)}{10.25x\,0.011}$$

$$= 5.76 \times 10^6$$

f calculated with Eq (9b)

$$f = \frac{1}{(2\text{Log}[(\frac{7}{\text{Re}})^{0.9} + (\frac{3.24\varepsilon}{d})])^2}$$

take, ε as 0.00018 in

f = 0.014

Trial and error continue, and the result as per table below;

f	q ft^3/hr	Re
0.013	1.87 x 10^6	5.76 x 10^6
0.014	1.79 x 10^6	5.53 x 10^6
0.014	**1.79 x 10^6**	

Than mean the natural gas flow rate is **1.79x10^6 ft^3/hr.**

Initial density

$$\rho = \frac{P_1 \times MW}{10.72(460+T)} = \frac{1200\times 20.9}{10.72(460+60)} = 4.5 \text{ lb/ft}^3$$

Inlet velocity

$$V = \frac{144q}{0.25\pi d^2 \times 3600} = \frac{144\times 1.79\times 10^6}{0.25\times 3.14\times 10.25^2} = 868.15 \text{ ft/s}$$

Exit velocity

$$V_{ex} = \left[\frac{2\times\left(g\times\Delta p_{tot} + 0.5\times\rho\times V^2\right)}{\rho}\right]^{0.5}$$

$$= \left[\frac{2\times\left(32.174\times(1200-250) + 0.5\times 4.5\times 868.15^2\right)}{4.5}\right]^{0.5} = 875.8 \quad ft/s$$

Example Case 5: Pump suction

Calculate the NPSHa and power required for a system pumping water with the following conditions:

Discharge rate of 1500 gpm
12-inch Schedule 40 PVC suction line (I.D. = 11.8 inches)
Barometric pressure is 29.8 in. Hg
Water temperature of 80° F
Vapor pressure of 1.2 ft.H_2O
Vacuum reading of 12.5 ft H2O on suction line.

Solution

Barometric pressure

PB = in.Hg x 0.882 = 29.8 in. Hg x 0.882 = 26.3 ft H2O

Flow rate =

Q = $\frac{GPM}{448.8} = \frac{1500}{448.8} = 3.34$ ft^3/sec

Pipe area

Di =11.8 in = 0.98 ft

$$A = \frac{\pi Di^2}{4} = \frac{3.14 \times 0.98^2}{4} = 0.76 \text{ ft}^2$$

flow velocity

$$V = \frac{Q}{A} = \frac{3.34}{0.76} = 4.4 \text{ ft/s}$$

Head in suction pipe

$$Hv = \frac{V^2}{2g} = \frac{4.4^2}{2 \times 32.2} = 0.3 \text{ ft}$$

Net Positive Suction Head

NPSHa = PB – VP ± HP + Hv = 26.3 – 1.2 – 12.5 + 0.3 = 12.88 ft.H2O

Pump velocity

$$vp = \frac{3540}{60(\rho_L)^{0.25}} = \frac{3540}{60(62.3)^{0.25}} = 21 \text{ ft/s}$$

Power required

$$HP = \frac{GPM \times NPSHa \times \rho_L \times 231}{33000 \times 1728 \times E} = \frac{1500 \times 12.88 \times 62.3 \times 231}{33000 \times 1728 \times 0.7} = 6.97 \text{ Hp}$$

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KLM Technology Group Practical Engineering Guidelines for Processing Plant Solutions	PROJECT: REV: 01 PAGE OF	EQUIPMENT: PIPELINES LOCATION: CLIENT:

ITEM NO	-						
TAG NUMBER	-						
LENGTH	-						
SERVICE OR LOCATION	-						
DESIGN PRESS	lbf/in^2						
DESIGN TEMP	F						
MATERIAL OF BODY	-						
FLOW CHARISTERISTIC	-						
PHASE (LIQ, GAS, STM)	-						
FLUID	-						
FLOW RATE	lbm/hr						
INLET PRESS.	lbf/in^2(g)						
EQUIPMENT LOSSES	lbf/in^2						
TEMPERATURE	F						
DENSITY	lbm/ft^3						
LOSSES / 100m	lbf/in^2						
VISCOSITY	cP						
DESTINATION P	lbf/in^2(g)						
FLUID VELOCITY	ft/s						
LINE SIZE, INLET	in						
LINE SIZE, OUTLET	in						
LINE Sch. OUTLET	-						
LINE NO. INLET	-						
INSPECTION	-						
LINE CLASS	-						
P&ID NO.	-						

REMARKS

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