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PROCESS DESIGN OF PIPING SYSTEMS (PROCESS PIPING AND PIPELINE SIZING)

(PROJECT STANDARDS AND SPECIFICATIONS)

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TRANSMISSION PIPELINES FOR LIQUID AND GAS

TWO PHASE FLOW

SCOPE

This Project Standards and Specifications covers process piping design and pipeline sizing, in addition to presenting most popular pressure drop equations and fluid velocity.

The subject of this Standard is to present mathematical relationships, based on which pipe size is calculated. The relationships presented cover Newtonian fluids which include most useful process piping application.

Unless noted otherwise, the methods suggested here do not contain any built-in safety factors. These should be included, but only to the extent justified by the problem at hand.

REFERENCES

Throughout this Standard the following dated and undated standards/codes are referred to. These referenced documents shall, to the extent specified herein, form a part of this standard. For dated references, the edition cited applies. The applicability of changes in dated references that occur after the cited date shall be mutually agreed upon by the Company and the Vendor. For undated references, the latest edition of the referenced documents (including any supplements and amendments) applies.

1. API (American Petroleum Institute)

   API Publication 2564
   Third Ed., December 2001, "Manual of Petroleum Measurement Standards; Chapter 15 Guidelines for the Use of International System of Units (SI) in the Petroleum and Allied Industries"

2. NACE (National Association of Corrosion Engineers)

   NACE MR 0175-2002, "Standard Material Requirements Sulfide Stress Cracking Resistant Metallic”

3. GPSA (Gas Processors Suppliers Association)

"Engineering Data Book", Vol. II, Section 17, 10th. Ed., 1987

4. Hydraulic Institute Standard

"Centrifugal, Rotary and Reciprocating Pumps", 14th. Ed., January 1982

DEFINITIONS AND TERMINOLOGY

AGA - American Gas Association

BBM - Begg's-Brill-Moody

dB - Decibles (unit of sound pressure level)

DN - Diameter Nominal, in (mm). The Nominal Pipe Size (NPS) will be designated by "DN" although in calculations the diameter generally has the units of millimeters (mm). The following table gives equivalents of Nominal Pipe Size in DN and Nominal Pipe Size (NPS) in inches:

DN (mm)	NPS (inches)	DN (mm)	NPS (inches)
15	½	400	16
20	¾	450	18
25	1	500	20
40	1½	600	24
50	2	650	26
80	3	700	28
100	4	750	30
150	6	800	32
200	8	900	36
250	10	1000	40
300	12		
350	14		

Eq. – Equation

ERW - Electric Resistance Welding

mmH$_2$O - In adopting the SI System of Units in this Standard it has been tried to satisfy the requirements of API Publication 2564. To this end, kilopascal (kPa) is adopted as the unit of pressure in calculations. But in cases where the pressure drop is expected to be small, millimeters of water column (mm H_2O) is also used [9.80665 Pa = 1 mm H_2O (Conventional)].

MSC - The Metric Standard Conditions. For measuring gases and liquids as referred to in the Standard is defined as 101.325 kPa and 15°C.

NGL - Natural Gas Liquids

NPS - Nominal Pipe Size, in (inch)

NPSHA - Net Positive Suction Head Available

NPSHR - Net Positive Suction Head Required

Re - Reynolds number

r/min - Rotations (revolutions) per minute (RPM)

s - second.

SYMBOLS AND ABBREVIATIONS

SYMBOL/ABBREVIATION	DESCRIPTION
A	Area of cross-sectional of pipe, in (m^2)
Am	Minimum pipe cross-sectional flow area required, in (mm per m^3/h liquid flow)
Amm	Cross-section of pipe, in (mm^2)
B_d	Rate of flow in barrels (42 U.S gallons)per day
B_h	Rate of flow in barrels (42 U.S gallons) per hour
B_X & B_Y	Baker parameters
C	Hazen-Williams constant
D	Inside diameter of pipe, in (m)

D_p	Particle diameter, in (mm)
d_i	Inside diameter of pipe, in (mm)
E	Efficiency factor
f	Friction factor of pipe, (dimenssionless)
f_D	Darcy's friction factor = f_m, (dimensionless)
f_m	Moody friction factor, (dimenssionless)
f_f or f_F	Fanning friction factor $f_D = f_m = 4f_F$, (dimenssionless)
g	Gravitational acceleration (usually is equal to 9.81 m/s^2)
G	Relative density of gas at the prevailing temperature and pressure relative to air, G =M(gas)/M(air), at 20°C and 760 mm of mercury.
h_f	Head loss due to friction, in (mm)
h_c	Head loss due to friction, in (mm)
h_R	Enthalpy of condensate at supply pressure, in (J/kg)
H	Enthalpy of condensate at return line pressure, in (J/kg)
h_1	Static head, in (m)
h_2	Initial elevation of pipeline, in (m)
K	Final elevation of pipeline, in (m)
K_e	Ratio of specific heat at constant pressure to the specific heat at constant volume cp/cv, (dimensionless)
L	Coefficient of resistance in pipe, fitting, valves and etc., in (m)
L_{km}	Length of pipe, in (m)
L_e	Equivalent length of pipe, in (m)
L_R	Latent heat of steam at return line pressure, in (J/kg)
M	Molecular mass, in (kg/mol)
P	Operating pressure, in [kPa (absolute)]
P_{ave}	Average gas pressure = $\frac{2}{3}\left(P_1 + P_2 - \frac{P_1 P_2}{P_1 + P_2}\right)$

P_f	Operating pressure in fittings, in [kPa (absolute)]
P_v	Vapor pressure of liquid in suction temperature of pump, in [kPa (absolute)]
P_o	Base pressure, in [101.325 kPa (absolute)]
P_1	Initial or inlet pressure, in [kPa (absolute)]
P_2	Final or outlet pressure, in [kPa (absolute)]
ΔP_{100}	Operating pressure, along 100 m of pipe, in kPa/100 (absolute) or [P_1-P_2/100 (absolute)]
ΔP_{100}	Pressure loss, in (kPa/100 m)
ΔP_{TP100}	Two-phase pressure, loss, in (kPa/100 m)
Q_L	Liquid flow rate, in (m^3/h)
Q_{sc}	Gas flow rate at P_o, T_o, in (m^3/h)
Q_v	Vapor flow rate, in (m^3/h)
q	Rate of flow at flowing conditions, in (m^3/s)
q_l	Liquid flow rate, in litre/minute (L/min)
q_s	Liquid flow rate, in (m^3/s)
R	Universal gas constant, in 8314.3/M (J/kg. mol. K)
R_e	Reynolds number
R_{em}	Modified Reynolds number
R_{gl}	Gas/liquid ratio in m^3 (gas)/m^3 (liquid) at MSC
S	Relative liquid density (water = 1)
T	Flowing temperature, in kelvin (K)
T_o	Base temperature = (273 + 15) = 288 K
V	Fluid velocity, in (m/s)
V_{ave}	Average fluid velocity, in (m/s)
V	Specific volume, in (m^3/kg)
V_c	Critical velocity with respect to sound velocity, in (m/s)
V_e	Fluid erosional velocity, in (m/s)
V_R	Specific volume of steam at return line pressure, in (m^3/kg)
W	Mass flow rate, in (kg/h)
W_T	Total fluid mass flow rate, in (kg/h), (liquid+vapor)

W_c	Condensate load, in (kg/h)
W_L	Liquid mass flow rate, in (kg/h)
W_g	Gas mass flow rate, in (kg/h)
x	mass (weight) fraction of vapor, (dimenssionless)
X	L & M modulus for two-phase
Z	Gas compressibility factor

Greek Letters:

Δ (delta)	Differential between two points
ε (epsilon)	Absolute pipe roughness in (mm)
γ(nu)	Kinematic viscosity, in (m^2/s) = $\frac{\text{absolute viscosity}}{\text{relative density}}$
μ (mu)	Absolute viscosity at flowing temperature and pressure, in (cP)
μ_g (mu)	Gas viscosity at flowing temperature and pressure, in (Pa.s)
ρ(rho)	Density, in (kg/m^3)
ρ_L (rho)	Liquid density, in (kg/m^3)
ρ_g (rho)	Gas density, in (kg/m^3)
ρ_V (rho)	Vapor density, in (kg/m^3)
ρ_m (rho)	Mixture density, in (kg/m^3)
ρ_{TP} (rho)	Two-phase flow density, in (kg/m^3)
λ(lambda)	Liquid volumetric fraction
φ(phi)	A fraction of L & M modules
δ(sigma)	Surface tension of liquid, in (dyne/cm = mN/m)

Subscripts:

1-	Refer to initial, or upstream conditions
2-	Refer to second, downstream or outlet
g-	Refers to gas
L-	Refers to liquid

UNITS

This Standard is based on International System of Units (SI) except where otherwise specified.

PROCESS PIPE SIZING FOR PLANTS LOCATED ONSHORE-SINGLE PHASE

GENERAL SIZING CRITERIA

The optimum pipe size should be based on minimizing the sum of energy cost and piping cost. However, velocity limitations causing erosion or aggravating corrosion must be taken into consideration. Sometimes, the line size must satisfy process requirements such as pump suction line. Although pipe sizing is mainly concerned with pressure drop, sometimes for preliminary design purposes when pressure loss is not a concern, process piping is sized on the basis of allowable velocity. When there is an abrupt change in the direction of flow (as in elbow or tees), the local pressure on the surface perpendicular to the direction of flow increases dramatically. This increase is a function of fluid velocity, density and initial pressure. Since velocity is inversely proportional to the square of diameter, high velocity fluids require special attention with respect to the size selection.

FLUID FLOW

In vapor systems, the use of rule of thumb or approximate sizing methods can lead to critical flow and subsequent vibration and whistling. With two-phase systems, improper sizing can lead to slug flow with its well known vibration and pressure pulsations.

With both vapor and two-phase systems, approximate calculations often neglect the importance of momentum on total pressure drop; the result being that, pressure drop available for controllability, is reduced; and rigorous calculations to determine pressure drop involving trial and error should be performed by computers. The problem is further complicated when a diameter is to be found which will produce a specified pressure drop or outlet velocity for a given flow. In this situation additional trial and error is required to determine the proper diameter. The design problem as described above is correctly defined as line sizing. The opposite problem, that of calculating velocity and pressure loss for a given diameter is very frequently encountered during hydraulic or "spool" checks. In general an evaluation of the total system equivalent length must be made based on fittings, valves, and straight line in the system. In addition, fitting and valve losses are not constant, but are functions of diameter. A preliminary line sizes must often be selected before an accurate knowledge of the system equivalent length is available, spool check calculations are required before final specifications for prime movers can be written on final diameter, chosen.

REYNOLDS NUMBER

The relationship between pipe diameter, fluid density, fluid viscosity and velocity of flow according to Reynolds number is as follows:

$$Re = \frac{d.V.\rho}{\mu} \qquad \text{Eq. (1)}$$

FRICTION FACTOR

The basis of the Moody friction factor chart (see Appendix A, and B) is the Colebrook equation.

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left[\frac{\varepsilon}{3.7D} + \frac{2.51}{R_e\sqrt{f}}\right] \qquad \text{Eq. (2)}$$

For reference chart and method of solution see Appendix A, and B.

FLUID FLOW CALCULATIONS

For calculation pressure loss for a single phase (liquid-gas-vapor) fluid at isothermal condition when flow rate and system characteristics are given; presented in this Standard through the application of Darcy-Weisbach (often referred to as simply Darcy) and Fanning principles

For compressible (gas and vapor lines, where the pressure losses are small relative to line pressure) reasonable accuracy can often be predicted providing the following conditions are met.

The average gas density of flow in uses i.e., $\rho = \frac{(\rho_1 + \rho_2)}{2}$

The pressure drop is less or equal 40% of up stream pressure
i.e., $(P_1 - P_2) \leq 0.4\ P_1$

This is because energy losses due to acceleration and density variations can be neglected up to this limit. In cases where the pressure loss is less than 10% of the upstream pressure, an average value of ñ is not required and either the downstream or upstream density can be used.

SINGLE PHASE LIQUID FLOW

For the calculation of pressure loss in liquid lines, the Darcy-Weisbach or Fanning methods shall be used. The calculation is simplified for liquid flows since the density can reasonably be assumed to be constant.
As a result the Darcy-Weisbach calculation can be applied to a long run of pipe rather than segmentally as directed by the variable density in gas flow. Elevation pressure drops must be calculated separately, using Equation (3):

$$\Delta P_e = \frac{h_L.\rho}{10200} \text{ kg/m}^2 \qquad \text{Eq. (3)}$$

The elevation pressures gains or losses are added algebratically to the frictional pressure drops.

Flow is considered to be laminar at Reynolds number of 2000 or less, therefor before using the formula for pressure drop, Reynolds number should be determined for regime of flow. The following formula is for pressure loss of laminar flow:

$$\Delta P_{100} = \frac{32\mu 2\mu}{d^2} = 4070 x 10^4 \mu.q_s/d^4 \text{ at flow condition} \qquad \text{Eq. (4)}$$

Where:

ΔP_{100} is the pressure drop in bar per 100 meters.

For a given mass flow rate and physical properties of a single phase fluid in turbulent conditions, P100 can be expressed:

$$\Delta P_{100} = 6253 \frac{f_p.W^2}{d^5.\rho} \qquad \text{Eq. (5)}$$

Alternatively, for a given volumetric rate, ΔP_{100} can be expressed as:

ΔP_{100} = 81055 x 10^7 f_D . q^2/d^5 = bar/100 meter

at flowing conditions (temperature and pressure)

FITTINGS AND VALVES

In case where the coefficient of resistance "K" are to be used, K = K valve + K elbow + K tee...... shall be taken and calculated from Appendices D, E and F. The value "K" is defined as follows:

$$\Delta P = n\frac{4.f.L}{d}\left(\frac{\rho.V_2}{2gc}\right) \text{ (Fanning equation)} \qquad \text{Eq. (7)}$$

$$K = \frac{4.f.L}{d} \qquad \text{Eq. (8)}$$

Pressure drops "$P\Delta_f$" in fittings can be calculated as follows:

$$\Delta P = K\left(\frac{\rho.V^2}{2gc}\right) \qquad \text{Eq. (9)}$$

Where:

ΔP_f is pressure drop in fitting (psi) or (kg/cm^2);
K is coefficient of resistance;
V is velocity is pipe (ft/sec) or (m/s);
P is density (lb/ft^3) or (kg/m^3);
gc is gravity constant 32.17 (lb/ft/sec^2. lb-force), 9.80 (kg.m/s^2. kg-force).

As a result, following equation is obtained:

$$\Delta P_f = K.\ \rho\ .\ V^2 / 196120 = 5.1 \times 10^{-6} K.\ .\rho.V^2 \text{ kg/cm}^2 \qquad \text{Eq. (10)}$$

$$= 5 \times 10^{-4} K.\ \rho.V^2 \text{ kPa} \qquad \text{Eq. (11)}$$

In cases where valves and fittings are to be handled as pipe equivalent lengths, the equivalent lengths shall be taken from Appendix G and added to the actual pipe lengths, from which the pressure drops shall be calculated.

SPECIAL CONDITIONS

1. Water Flow

   The pressure loss for water flow shall be calculated by Hazen- Williams's formula. The Hazen-William's relationship, is one of the most accurate formula for calculation pressure loss in water line (see Appendix C for Hazen-William's constant C). For the design of new water pipelines, constant "C" is taken as "100". The Hazen-s formula is as follows:

$$h_f = 2.25x10^4 L_e\left(\frac{100}{C}\right)^{1.85}\left(\frac{Q_w^{1,85}}{d^{4.8655}}\right) \qquad \text{Eq. (12)}$$

2. Pump Suction Lines

Generally the pressure drops in pump suction lines shall be held below 4.5 kPa/100 m; in the case of liquid at the boiling point and below 7.9 kPa/100 m in the case of liquid below the boiling point.

The maximum velocity of bubble point liquids shall be 1.2 m/s and for sub-cooled liquids shall be 2.4 m/s. For corrosive liquids these values may be reduced by fifty percent.

Allowable pressure drops can be determined by the following formula:

$$\Delta P = 9.835\ S\ [H-(NPSHR + \alpha)] + (P_1-P_v) \qquad \text{Eq. (13)}$$

<u>Where:</u>

ΔP is friction loss in piping to pump inlet, in (kpa);

S is relative density (Water = 1);

H is height from datum to pump centre, in (m) (the term "Datum" refers to the bottom tangent line in the case of vertical vessels and to the bottom level in the case of horizontal vessels);

NPSHR is net positive suction head required, in (m);

α(alpha) is 0.305 m (1 ft) for liquid at boiling point and 0.2134 m for liquid below boiling point;

P_1 is pressure working on suction liquid surface, in (kPa);

P_v is vapor pressure of liquid at suction temperature, in (kPa).

In cases where permanent strainers are to be provided a minimum pressure drop of 3.45 kPa (0.5 psi) shall be added in the case o f dirty service. No addition is required in the case of clean service.
The equivalent length to be used for pressure drop calculations shall be assumed to be 46 m (150 ft).

A suction liquid line to a centrifugal pump should be short and simple. Velocities are usually between 0.3 to 2.13 m/s. Higher velocities and unit losses can be allowed within this range when subcooled liquid is flowing than when the liquid is saturated.

Note that the longer payout times favor larger pipe diameters. Pipe smaller than pump discharge nozzle size is not used.

3. Cooling Water

   Cooling water discharge headers are usually sized with unit pressure losses in decimals of 7 kPa (1 psi). An economical comparison is justified with large diameter piping, where most of the pump pressure is used for pipe and equipment resistance. Of course, piping costs increase with diameter while utility costs decrease. Between alternate design the best size can be determined by adding the total cost of utilities over the period of capital payout to the capital cost of each installation. The lowest over-all figure will give the most economical solution.

4. Limitations Owing to Erosion Preventive Measures

   Velocity of the fluid plays an important role in errosion-corrosion. Velocity often strongly influences the mechanism of the corrosion reactions. Mechanical wear effects at high values and particularly when the solution contains solids in suspension.

   a. Amine solution

      The following limitations should be considered.

      For carbon steel pipe:

      - Liquid 3 m/s
      - Vapor-liquid 30 m/s

      For stainless steel pipe:

      - Liquid 9 m/s
      - Vapor-liquid 36 m/s

   b. Ammonium bisulfate (NH3-H2S-H2O) solution

      Aqueous solutions of ammonium bisulfate produced in the effluent line of hydrocracking, hydrotreating processes often cause rapid erosion- corrosion of carbon steel pipes, especially for nozzles, bend, tees, reducer and air cooler tube inlet parts after water injection points. Care must be taken not to exceed the highest fluid velocity in pipe tubes.

5. Other Matters

   a. Gravity flow

      i) Side cut draw-off

         In cases where no controller is provided for the liquid level in the liquid draw-off tray, the flow velocity in the first 3 meters of the vertical line shall be less than 0.762 m/s. This value is intended for vapor-liquid separation based on the particle diameter 200 micrometers (1000 micron = 1 mm) in cases where the operating pressure is high or the difference between the vapor and liquid densities is small:

$$CV = \frac{Q}{\sqrt{\Delta P / S.G}} \quad \text{Eq. (14)}$$

Where:

CV = The flow coefficient or pressure loss coefficient

Q = flow rate, GPM

ΔP = pressure drop, psi

S.G = specific gravity of the fluid

The line size shall be also checked that the control valve size may not become larger than the line size.

ii) Determination of line sizes in cases where the liquid enters a control valve at the boiling point should be sized on the following basis:

With consideration given to the static head and length of the line from the liquid level to the control valve, the line size shall be determined so that no vaporization may occur at the inlet of control valve. In this case, the following should be satisfied;

$$0.1412\ \rho.H > \Delta P\ (\text{flowmeter}) + \Delta P\ (\text{line friction}) \quad \text{Eq. (15)}$$

Or

$$2.26\ Sh > \Delta P\ (\text{flowmeter}) + P\ (\text{line friction}) \quad \text{Eq. (16)}$$

Where:

ρ(rho) is relative density, in (kg/m3);

H is static head, in (m);

h is static head, in (mm);

S is relative density;

ΔP (flowmeter) = kPa pressure drop in flowmeter;

ΔP (line friction) = kPa pressure drop due to friction.

b. Vacuum tower overhead line

i) The line cost and steam and cooling water consumptions shall be calculated and the line size shall be decided so that the annual cost will become minimum.

ii) Pre-condenser shall be provided in wet cases where the pressure loss in the pre-condenser shall be 4 mm Hg.

c. Steam condensate lines

i) Line from heat exchanger to steam trap or control valve

The pressure drop in this line shall be smaller than 11.3 kPa/100 m (0.1 kg/cm^2/100 m) and shall be checked that no condensate may vaporize therein.

ii) Line from steam trap or control valve to following vessel

- Steam condensate return lines must be sized to avoid excessive pressure loss. Part of the hot condensate flashes into steam when it is discharged into the condensate return system.
- In this case, the flow velocity "V" must be limited to 1524 m/min to prevent erosion.
- The flow velocity shall be calculated by the following equation:

$$V = \frac{354.W_c.V_R(h_c - h_R)}{d^2.L_R} \qquad \text{Eq. (17)}$$

d. Flare headers

Flare headers shall be designed so that the maximum allowable velocity does not exceed 50 percent of critical velocity, a figure mostly practiced by design companies.

SINGLE PHASE GAS FLOW

1. In general when considering compressible flow, as pressure decreases along the line so does the density (assuming isothermal flow). A variation in density implies variation in Reynolds number on which the friction factor is dependent. A rigorous calculation of pressure loss for long pipeline involves dividing it into segments, performing the calculation for each segment (considering variable parameters) and integrating over the entire length. For process piping however, since pipe lengths are generally short, a rigorous calculation would not be necessary and the equation outline below are considered adequate.
2. As mentioned above for estimating pressure drop in shortrun of gas piping such as within plant or battery limit, a simplified formula for compressible fluids is accurate for fully turbulent flow, assuming the pressure drop through the line is not a significant fraction of the total pressure (i.e., no more than 10%).

3. The Darcy formula (Eq. 5) also can be used for calculation of pressure loss in process gas lines as follows:

$$\Delta P_{100} = \frac{62530 f_D . W_g^{\ 2}}{\rho_g . d^5} \text{ bar/100 m} \qquad \text{see Eq. (5)}$$

4. A Practical Way to Calculate Gas Flow in Pipeline

   Here is a short cut way to calculate gas flow in pipelines. It is based on Weymouth formula. At 15.5°C and relative density (specific gravity) of 0.6, the answer will be accurate. For every 5.5°C (10°F) variation in temperature, the answer will be 1% error. For every 0.01 variation in relative density (specific gravity), the answer will be three-fourths percent in error:

   Formula:

$$Q_g = \frac{10.73 x 10^{-4} . d^{8/3} \sqrt{P_1^{\ 2} - P_2^{\ 2}}}{\sqrt{L}} \text{ m}^3\text{/h} \qquad \text{Eq. (18)}$$

   Where:

   Qg is cubic metere of gas per hour, (m3/h);

   d is pipe ID in (mm);

   P_1 is kPa (abs) at starting point;

   P_2 is kPa (abs) at ending point;

   L is length of line in (m).

5. An important factor in handling compressible fluid flow is a phenomenon known as critical flow. As the pressure drop in a pipe (increases) so does the flow. But for compressible flow this increase is limited to the velocity of sound in the fluid at flowing conditions. This limit is called the critical velocity.

   Sonic or critical velocity is the maximum velocity which a compressible fluid can attain in a pipe. For trouble-free operation maintain operable velocities at 0.5 V_c and V_c for ideal gasis is given by:

$$V_c = \sqrt{\frac{K.R.T}{m}} \text{m/s} \qquad \text{Eq. (19)}$$

$$= 31.64 \ V_c \sqrt{\frac{K.P}{m}} \text{m/s} \qquad \text{Eq. (20)}$$

The maximum velocity in piping handling compressible shall be less than ½ of the critical velocity.

6. System operating at pressure less than 7000 N/m² (7 kPa), the Spitzglass equation shall be used for pressure loss calculations:

$$Q = 0.00338\left[\frac{\Delta h_w . d^5}{G.L\left(1+\frac{91.5}{d}+0.00118d\right)}\right]^{0.5} \quad m^3/h \text{ at } 115^oC \qquad \text{Eq. (21)}$$

$$Q = \left[\frac{0.00108\Delta.0\,d^5}{G.L\left(1+\frac{91.5}{d}+0.00118d\right)}\right]^{0.5} \quad m^3/h \text{ at } 115^oC \qquad \text{Eq. (22)}$$

<u>Where:</u>

ΔP is the pressure drop in Pa.

7. Steam Flow

Babcock formula shall be used to calculate pressure drop in steam lines:

$$\Delta P_f = 3.63.10^{-8}\left(\frac{d+3.6}{d^6}\right)\frac{W^2L}{\rho} \qquad \text{(Eq. 23)}$$

<u>Where:</u>

ΔP_f is frictional component of pressure drop, psi

8. Flow Induced Noise

The allowable maximum flow velocities in cases where the maximum sound pressure levels of the piping noises must be kept 8 to 10 dB (A) under the sound pressure level of the background noise, are as follows:

NORMAL BACKGROUND SOUND PRESSURE, dB (A)	MAXIMUM FLUID VELOCITY TO PREVENT NOISE, m/s*
60	30
80	41
90	52

* Obviously these velocity limitations refer to compressible flow.

PROCESS PIPE SIZING FOR PLANTS LOCATED OFFSHORE

SCOPE

This document recommends minimum requirements and guidelines for the sizing of new piping system on production platforms located offshore. The maximum design pressure within the scope of this document is 69000 kPa gage (10000 psig) and the temperature range is -29°C (-20°F) to 343°C (650°F). For applications outside these pressures and temperatures, special consideration should be given to material properties (ductility, carbon migration and etc.). The recommended practices, presented are based on years of experience in developing oil and gas losses. Practically all of the offshore experience has been in hydrocarbon service free of hydrogen sulfide. However, recommendations based on extensive experience onshore are included for some aspects of hydrocarbon service containing hydrogen sulphide.

In determining the transition between risers and platform piping which these practices apply, the first incoming and last outgoing valve which block pipeline flow shall be the limit of this document's application.

SIZING CRITERIA-GENERAL

In determining the diameter of pipe to be used in platform piping systems, both the flow velocity and pressure drop should be considered. The following sections present equations for calculating pipe diameters for liquid lines, singlephase gas lines and gas/liquid two-phase lines, respectively. Many companies also use computer programs to facilitate piping design.

1. When determining line sizes, the maximum flow rate expected during the life of the facility should be considered rather than the initial flow rate. It is also usually advisable to add a surge factor of 20 to 50 percent to the anticipated normal flow rate, unless surge expectations has been more precisely determined by pulse pressure measurements in similar systems or by specific fluid hammer calculation.
2. Determination of pressure loss in a line should include the effects of valves and fittings. Manufacturer's data or an equivalent length as in Appendix G shall be used.
3. Calculated line sizes may need to be adjusted in accordance with good engineering judgment.

SIZING CRITERIA FOR LIQUID LINES

1. General

   Single-phase liquid lines should be sized primarily on the basis of flow velocity. For lines transporting liquids in single phase from one pressure vessel to another by pressure differential, the flow velocity should not exceed 4.6 m/s at maximum flow rates, to minimize flashing ahead of the control valve. If practical flow velocity should not be less than 0.91 m/s to minimize deposition of sand and other solids. At these flow velocities, the overall pressure drop in the piping will usually be small. Most of the pressure drop in liquid lines between two pressure vessels will occur in the liquid dump valve and/or choke.

2. Flow velocities in liquid lines may be calculated using the following derived equation:

$$V = 353.7\,\frac{Q_L}{d_L^2} \qquad \text{Eq. (24)}$$

3. Pressure loss (kPa per 100 meter of flow length) for single-phase liquid lines may be calculated using the following (Fanning) equation:

$$\Delta P_{100} = 62.66x10^8\,\frac{f_m.Q_L^2.S_L}{d_i^5}\ \text{kPa/100 m} \qquad \text{Eq. (25)}$$

4. The Moody friction factor "f" is a function of the Reynolds number and surface roughness of the pipe. The modified Moody diagram, in Appendix A may be used to determine the friction factor when the Reynolds number is known.

PUMP PIPING

1. Reciprocating, rotary and centrifugal pump suction piping systems should be designed so that the available net positive suction head (NPSH) at the pump inlet flange exceeds the pump required NPSH. Additionally provisions should be made in reciprocating pump suction piping to minimize pulsations. Satisfactory pump operation requires that essentially no vapor be flashed from the liquid as it enters the pump casing or cylinder.
2. In a centrifugal or rotary pump, the liquid pressure at the suction flange must be high enough to overcome the pressure loss between the flange and the entrance to the impeller vane (or rotor) and maintain the pressure on the liquid above its vapor pressure. Otherwise cavitation will occur.