ENGINEERING STANDARD

FOR

PROCESS DESIGN OF PIPING SYSTEMS

(PROCESS PIPING AND PIPELINE SIZING)

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0. INTRODUCTION

This Standard covers single phase liquid flow, single phase gas flow, and specific cases requiring special treatment, and a brief discussion of two-phase, two component flow calculations in short process pipes.

"Process Design of Piping Systems (Process Piping and Pipeline Sizing)" are broad and contain variable subjects of paramount importance.

Therefore, this Standard consists of four parts as described below:

STANDARD CODE	**STANDARD TITLE**
IPS-E-PR-440/1 Part One:	"Process Pipe Sizing for Plants Located Onshore-Single Phase"
IPS-E-PR-440/2 Part Two:	"Process Pipe Sizing for Plants Located Offshore"
IPS-E-PR-440/3 Part Three:	"Transmission Pipeline for (1)-Liquid and (2)-Gas"
IPS-E-PR-440/4 Part Four:	"Two-Phase Flow"

The flow of liquid, gases, vapor, two-phase flow and many other fluid systems have received sufficient study to allow definite evaluation of conditions for a variety of process situations for Newtonian fluids, which will be discussed later on. For non-Newtonian fluids considerable data is available. However its correlation is not as broad in application due to the significant influence of physical and reological properties. This presentation is limited to Newtonian system except where noted. Primary emphasis is only given to flow through circular pipe and tubes, since this is the usual means of movement of gases and liquids in process plants.

The basis for fluid flow follows Darcy and Fanning concepts. The exact transmission from laminar or viscous flow to turbulent condition is variously identified or between a Reynolds number of 2000 to 3000.

The correlation included in this Standard are believed to fit average plant design with good engineering accuracy. However other basis and correlations used by designer should be mutually agreed upon.

As a matter of good practice with the exercise of proper judgment, the designer should familiarize himself with the background of the methods presented, in order to better select the conditions associated with a specific problem.

Most published correlations for two-phase prossure drop are empirical and, thus, and limited by range at data for which they were derived. A mathematical models for predicting the flow regime and a procedure for calculating pressure drop in process pipelines are presented in this Standard.

Design conditions may be:

a) Flow rate and pressure drop allowable established, determine pipe diameter for a fixed length.
b) Flow rate and length known determine pressure drop and line size in the range of good engineering practice.

Usually either of these conditions requires a trial approach based upon assumed pipe sizes to meet the stated conditions. Some design problems may require determination of maximum flow for a fixed line size and length; however, this just becomes the reverse of conditions above.

Optimum economic line size is seldom realized in the average process plant. Unknown factors such as future flow rate allowance, actual pressure drops through certain process equipment, etc. can easily over balance any design predicated on selecting the optimum. Certain guides as to order of magnitude of costs and sizes can be established either by one of several correlations or by conventional cost estimating methods. The latter is usually more realistic for a given set of conditions, since generalized equations often do not fit a plant system. Optimum criteria for pipe size should be subject to mutual agreement between Company and designer.

Unless otherwise stated, equations presented here are only used for calculating pressure drop due to friction. Therefore pressure loss or gain due to elevation must be taken into consideration where appropriate .

For supplementary information regarding the mechanical part of design, reference shall be made to Engineering Standard IPS-E-PI-240 for "Plant Piping Systems" as well.

1. SCOPE

This Engineering Standard 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.

Design practice breaks the overall problem into small component parts which allow for simple analysis and solution. This is recommended approach for selection and sizing of process piping.

The economic diameter of a pipe will be the one which makes the sum of amortized capital cost plus operating cost a minimum. It can correlate this total cost (per unit time or production) versus diameter to determine a minimum.

The physical properties of a flowing fluid must be known to predict pressure drop in piping. The two properties entering into the solution of most fluid flow problems are viscosity and density.

2. REFERENCES

Throughout this Standard the following standards and codes are referred to. The editions of these standards and codes that are in effect at the time of publication of this Standard shall, to the extent specified herein, form a part of this Standard. The applicability of changes in standards and codes that occur after the date of this Standard shall be mutually agreed upon by the Company and the Vendor/Consultant:

API (AMERICAN PETROLEUM INSTITUTE)

API Publication 2564
2nd. Ed., December 1980,
Reaffirmed, August 1987 "Manual of Petroleum Measurement Standards; Chapter 15 Guidelines for the Use of International System of Units (SI) in the Petroleum and Allied Industries"

IPS (IRANIAN PETROLEUM STANDARDS)

IPS-E-PR-460 "Process Design of Flare and Blowdown Systems"
IPS-E-PI-240 "Plant Piping Systems"

NACE (NATIONAL ASSOCIATION OF CORROSION ENGINEERS)

NACE RP 0175-75, 1975 "Control of Internal Corrosion in Steel Pipelines and Piping Systems"

GPSA (GAS PROCESSORS SUPPLIERS ASSOCIATION)

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

HYDRAULIC INSTITUTE STANDARD

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

3. 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_2O 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.

4. SYMBOLS AND ABBREVIATIONS

Unless otherwise stated, all symbols used in the Engineering Standard are defined as follows:

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	Enthalpy of condensate at supply pressure, in (J/kg)
h_R	Enthalpy of condensate at return line pressure, in (J/kg)
H	Static head, in (m)
h_1	Initial elevation of pipeline, in (m)
h_2	Final elevation of pipeline, in (m)
K	Ratio of specific heat at constant pressure to the specific heat at constant volume cp/cv, (dimensionless)
K_e	Coefficient of resistance in pipe, fitting, valves and etc., in (m)
L	Length of pipe, in (m)
L_{km}	Length of pipe, in (km)
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 $=\frac{2}{3}\left(\frac{P_1^3 - P_2^3}{P_1^2 - P_2^2}\right)$, in kpa
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 flow = $\left(\frac{\Delta P_{L100}}{\Delta P_{V100}}\right)^{0.5}$

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
σ_L***(sigma)***	Surface tension of liquid, in (dyne/cm = mN/m)

Suscripts:

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

5. UNITS

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

PART ONE
PROCESS PIPE SIZING FOR PLANTS LOCATED ONSHORE-SINGLE PHASE

6. 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.

7. FLUID FLOW

7.1 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.

8. REYNOLDS NUMBER

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

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

9. FRICTION FACTOR

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

$$1/\sqrt{f} = 2\log_{10}\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{R_e\sqrt{f}}\right) \quad \textbf{(Eq. 2)}$$

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

10. FLUID FLOW CALCULATIONS

10.1 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.

10.2 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 ρ is not required and either the down-stream or upstream density can be used.

11. SINGLE PHASE LIQUID FLOW

11.1 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):

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

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

11.2 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:

$$\Delta P_{100} = \frac{32.\mu.V}{d^2} = 4074 \times 10^4\ \mu.q_s/d^4 \text{ at flow condition} \qquad \textbf{(Eq. 4)}$$

Where:

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

11.3 For a given mass flow rate and physical properties of a single phase fluid in turbulent conditions, ΔP_{100} can be expressed:

$$\Delta P_{100} = 62530\ \frac{f_D.W^2}{d^5.\rho}\ bar/100m \qquad \textbf{(Eq. 5)}$$

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

$$\Delta P_{100} = 81055 \times 10^7\ f_D .\ q^2/d^5\ bar/100\ meter$$
$$\text{at flowing conditions (temperature and pressure)} \qquad \textbf{(Eq. 6)}$$

12. 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:

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

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

Pressure drops "ΔP_f" in fittings can be calculated as follows:

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

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:

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

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

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.

13. SPECIAL CONDITIONS

13.1 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:

$$h_f = 2.25 \times 10^4 L_e \left(\frac{100}{C}\right)^{1.85} . \left(\frac{Q_w^{1.85}}{d^{4.8655}}\right) \qquad \textbf{(Eq. 12)}$$

13.2 Pump Suction Lines

13.2.1 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:

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

Where:

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).

13.2.2 In cases where permanent strainers are to be provided a 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).

13.2.3 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.

13.3 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.

13.4 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.

13.4.1 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**

13.4.2 NH_3-H_2S-H_2O solution

Aqueous solutions of ammonium bisulfide produced in the effluent line of hydrocracking, hydrotreating processes often cause rapid errosion- 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.

13.5 Other Matters

13.5.1 Gravity flow

1) 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 vaporliquid 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:

$$V = \sqrt{0.003g(\rho_L - \rho_V).\ D_p / \rho_V} \qquad \textbf{(Eq. 14)}$$

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

2) 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;

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

or

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

Where:

ρ(rhow)	is relative density, in (kg/m^3);
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.

13.5.2 Vacuum tower overhead line

1) 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.

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

13.5.3 Steam condensate lines

13.5.3.1 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.

13.5.3.2 Line from steam trap or control valve to following vessel

1) 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.

2) In this case, the flow velocity "V" must be limited to 1524 m/min to prevent erosion.

3) The flow velocity shall be calculated by the following equation:

$$V = \frac{354.\ W_C\ V_R\ (h_C - h_R)}{d^2.\ L_R} \qquad \textbf{(Eq. 17)}$$

13.5.4 Flare headers

Flare headers shall be designed so that the maximum allowable velocity does not exceed 60 percent of critical velocity, a figure mostly practiced by design companies (refer to Engineering Standard IPS-E-PR-460 for, "Process Design of Flare and Blowdown Systems").

14. SINGLE PHASE GAS FLOW

14.1 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.

14.2 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%).

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

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

14.4 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:

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

Where:

Q_g	is cubic metere of gas per hour, (m^3/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).

14.5 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:

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

$$= 31.64 \sqrt{\frac{K.P}{\rho}} \; m/s \qquad \textbf{(Eq. 20)}$$

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

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

$$Q = 0.00338 \left[\frac{\Delta h_w . d^5}{G.L \left(1 + \frac{91.5}{d} + 0.00118 d\right)} \right]^{0.5} m^3/h \text{ at } 15°C \qquad \textbf{(Eq. 21)}$$

$$Q = 0.00108 \left[\frac{\Delta P . d^5}{G.L \left(1 + \frac{91.5}{d} + 0.00118 d\right)} \right]^{0.5} m^3/h \text{ at } 15°C \qquad \textbf{(Eq. 22)}$$

Where:

ΔP is the pressure drop in Pa.

14.7 Steam Flow

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

$$\Delta P = 1.72 \times 10^{4} \left(\frac{0.0394d + 3.6}{d^{6}} \right) \left(\frac{W^{2} . L}{\rho_{s}} \right) \qquad \textbf{(Eq. 23)}$$

14.8 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.**

PART TWO
PROCESS PIPE SIZING FOR PLANTS LOCATED OFFSHORE

15. 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.

16. 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, single-phase gas lines and gas/liquid two-phase lines, respectively. Many companies also use computer programs to facilitate piping design.

16.1 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.

16.2 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 .

16.3 Calculated line sizes may need to be adjusted in accordance with good engineering judgment.

17. SIZING CRITERIA FOR LIQUID LINES

17.1 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.

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

$$V = 356.2 \frac{Q_L}{d_L^{\,2}} \qquad \textbf{(Eq. 24)}$$

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

$$\Delta P_{100} = 62.66 \times 10^8 \frac{f_m . Q_L^{\,2} . S_L}{d_i^5} \ kPa/100m \qquad \textbf{(Eq. 25)}$$

17.4 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.

18. PUMP PIPING

18.1 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.

18.2 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.

In a reciprocating unit, the pressure at the suction flange must meet the same requirement; but the pump required NPSH is typically higher than for a centrifugal pump because of pressure drop across the valves and pressure drop caused by pulsation in the flow. Similary the available NPSH supplied to the pump suction must account for the acceleration in the suction piping caused by the pulsating flow, as well as the friction, velocity and static head.

18.3 The necessary available pressure differential over the pumped fluid vapor pressure may be defined as net positive suction head available ($NPSH_a$).

It is the total head in meter absolute determined at the suction nozzle, less the vapor pressure of the liquid in meter absolute. Available NPSH should always be equal or exceed the pump's required NPSH. Available NPSH for most pump applications may be calculated using Equation 26.

$$NPSH_a = h_p - h_{vpa} + h_{st} - h_f - h_{vh} - h_a \qquad \textbf{(Eq. 26)}$$

Where:

h_p is absolute pressure head due to pressure, atmospheric or otherwise, on surface of liquid going to suction, meter of liquid;

h_{vpa} is the absolute vapor pressure of the liquid at suction temperature, meter of liquid;

h_{st} is static head, positive or negative, due to liquid level above or below datum line (centerline of pump), meter of liquid;

h_f is friction head, or head loss due to flowing friction in the suction piping, including entrance and exit losses, meter of liquid;

h_{vh} is velocity head, meter of liquid;

h_a is acceleration head, meter of liquid;

V_L is velocity of liquid in piping, meter/second (m/s);

g is gravitational constant (usually 9.81 m/s²).

18.4 For a centrifugal or rotary pump, the acceleration head, h_a, is zero. For reciprocating pumps, the acceleration head is critical and may be determined by the following equation from the Hydraulics Institute.

$$h_a = \frac{L.\, V_L.\, R_P.\, C}{K.g} \qquad \textbf{(Eq. 27)}$$

Where:

h_a is acceleration head, in meter of liquid;
L is length of suction line, in meter (actual length not equivalent length);
V_L is average liquid velocity in suction line, in meter/second (m/s);
R_p is pump speed, in rotations/minute (r/min);
C is empirical constant for the type of pump:

= 0.200 for simplex double acting;
= 0.200 for duplex single acting;
= 0.115 for duplex double acting;
= 0.066 for triplex single or double acting;
= 0.040 for quintuplex single or double acting;
= 0.028 for septuplex single or double acting.

Note:

The constant "C" will vary from these values for unusual ratios of connecting rod length to crank radius.

K is a factor representing the reciprocal of the friction of the theoretical acceleration head which must be provided to avoid a noticeable disturbance in the suction piping:

= 1.4 for liquid with almost no compressibility (deaereted water);
= 1.5 for amine, glycol, water;
= 2.0 for most hydrocarbons;
= 2.5 for relatively compressible liquid (hot oil or ethane);

g is gravitational constant (usually 9.81 m/s^2).

It should be noted that there is not universal acceptance for Equation 27. However, Equation 27 is believed to be a conservative basis which will assure adequate provision for acceleration head.

18.5 when more than one reciprocating pump is operated simultaneously on a common feed line, at time, all crankshafts are in phase and, to the feed system, the multiple pumps act as one pump of that type with a capacity equal to that of all pumps combined. In this case, the maximum instantaneous velocity in the feed line would be equal to that created by one pump having a capacity equal to that of all the pumps combined.

18.6 If the acceleration head is determined to be excessive, the following should be evaluated:

a) Shorten suction line. Acceleration head is directly proportional to line length, L.

b) Use larger suction pipe to reduce velocity. This is very helpful since velocity varies inversely with the square of pipe inside-diameter. Acceleration head is directly proportional to fluid velocity V_L.

c) Reduce required pump speed by using a larger size piston or plunger, if permitted by pump rating. Speed required is inversely proportional to the square of piston diameter. Acceleration head is directly proportional to pump speed Rp.

d) Consider a pump with a larger number of plungers. For example: C = 0.04 for a quintuplex pump. This is about 40% less than C = 0.066 for a triplex pump. Acceleration head is directly proportional to C.

e) Consider using a pulsation dampener if the above remedies are unacceptable. The results obtainable by using a dampener in the suction system depend on the size, type, location and charging pressure used. A good, properly located dampener, if kept properly charged may reduce L, the length of pipe used in acceleration head equation to a value of 5 to 15 nominal pipe diameter. Dampener should be located as close to the pump suction as possible.

f) Use a centrifugal booster pump to charge the suction of the reciprocating pump.

18.7 The following requirements are recommended for designing suction piping:

a) Suction piping should be one or two pipe sizes larger than the pump inlet connection.

b) Suction lines should be short with a minimum number of elbows and fittings.

c) 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.

d) 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.

e) 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.

18.8 Reciprocating, centrifugal and rotary pump discharge piping should be sized on an economical basis. Additionally, reciprocating pump discharge piping should be sized to minimize pulsations. Pulsations in reciprocating pump discharge piping are also related to the acceleration head, but are more complex than suction piping pulsations. The following guidelines may be useful in designing discharge piping:

a) Discharge piping should be as short and direct as possible.

b) Discharge piping should be one or two pipe sizes larger than pump discharge connection.

c) Velocity in discharge piping should not exceed three times the velocity in the suction piping. This velocity will normally result in an economical line size for all pumps, and will minimize pulsations in reciprocating pumps.

d) For reciprocating pumps, include a suitable pulsation dampener (or make provisions for adding a dampener at a later date) as close to the pump cylinder as possible.

The Table 1 below may be used to determine preliminary suction and discharge line sizes.

TABLE 1 - TYPICAL FLOW VELOCITIES

	SUCTION VELOCITY m/s (ft/sec)	DISCHARGE VELOCITY m/s (ft/sec)
Reciprocating pumps		
Speeds upto 250 r/min	0.61 (2 ft/sec)	1.83 (6 ft/sec)
Speeds 251-330 r/min	0.46 (1½ ft/sec)	1.37 (4½ ft/sec)
Speeds above 330 r/min	0.305 (1 ft/sec)	0.91 (3 ft/sec)
Centrifugal pumps	0.61-0.91 (2-3 ft/sec)	1.83-2.74 (6-9 ft/sec)

19. SIZING CRITERIA FOR SINGLE-PHASE GAS LINES

19.1 Process Lines

When pressure drop is a consideration (lines connecting two components operating at essentially the same pressure, etc.) single-phase gas lines should be sized on the basis of acceptable pressure loss.

The pressure drops listed in the following Table 2, have been found by experience to be an acceptable balance for short lines, when capital costs (pipe, compression) and operating cost are considered. When velocities in gas lines exceed 18.3 m/s (60 ft/sec), noise may be a problem.

TABLE 2 - ACCEPTABLE PRESSURE DROPS FOR SINGLE-PHASE GAS PROCESS LINES

OPERATING PRESSURE kPa (ga)	ACCEPTABLE PRESSURE DROP kPa/100m (psi/100 ft)
100-690	1.13-4.30 (0.05-0.19)
696-3447	4.52-11.082 (0.2-0.49)
3454-13790	11.3-27.14 (0.5-1.2)

19.2 The above Table 2 may be used to determine pressure loss; if the total pressure drop is less than 10% of inlet pressure. If the total pressure drop is greater than 10% an equation such as Weymouth's should be used.

19.3 Gas velocity may be calculated using the following derived equation:

$$V = 124.1 \frac{Z.\, Q_g.\, T}{d^2.\, P} \qquad \textbf{(Eq. 28)}$$

19.4 Compressor Lines

Reciprocating and centrifugal compressor piping should be sized to minimized pulsation, vibration and noise. The selection of allowable velocities requires an engineering study for each specific application.

19.5 The following equation (Fanning) may be used when total pressure loss is less than 10% of inlet pressure and is based on friction factor given by the GPSA Data Book.

$$\Delta P = 1.433 \times 10^5 \frac{G.\, L_e.\, f.\, T.\, Q_g^{\,2}}{p.d_i^5}\ kPa \qquad \textbf{(Eq. 29)}$$

Where:

$$f = \frac{1.81}{\left(\frac{d_i \cdot V_g \cdot \rho_g}{\mu_g}\right)^{0.38}} + 0.0025 \qquad \textbf{(Eq. 30)}$$

20. SIZING CRITERIA FOR GAS/LIQUID TWO-PHASE LINES

Erosional velocity, flow lines, production manifolds, process headers and other lines transporting gas and liquid in two-phase flow should be sized primarily on the basis of flow velocity. Flow velocity should be kept at least below fluid erosional velocity. If solid (sand) production is anticipated fluid velocity should be reduced accordingly.

20.1 The velocity above which erosion may occur can be determined by the following empirical equation.

$$V_e = \frac{1.22C}{\sqrt{\rho_m}}\ m/s \qquad \textbf{(Eq. 31)}$$

Where:

V_e is fluid erosional velocity, in (m/s);

C is empirical constant:

= 125 for-non-continuous service;

= 100 for continuous service;

ρ_m is gas/liquid mixture density at operating pressure and temperature, in (kg/m³).

20.2 The density of the gas/liquid mixture may be calculated using the following derived equation:

$$\rho_m = \frac{28829.6 S_L.P + 35.22 R.G.P}{28.82P + 10.12 R.T.Z} \ kg/m^3 \qquad \textbf{(Eq. 32)}$$

Where:

S_L	is relative density of liquid (water = 1), use average gravity for hydrocarbon-water mixtures at Standard Conditions;
R	is gas/liquid ratio, m³ (gas)/m³ (liquid) at MSC;
P	is operating pressure, in (kPa absolute);
T	is operating temperature, in (K);
G	is gas relative density ($\frac{MW}{28.9}$) at Standard Conditions;
$\rho_{m(rho)}$	is gas/liquid mixture density, in (kg/m³).

20.3 Once V_e is known, the minimum cross-sectional area required to avoid fluid erosion may be determined from the following derived equation:

$$A = \frac{277.65 + \frac{97.4\, Z R T}{P}}{V_e} \qquad \textbf{(Eq. 33)}$$

20.4 Minimum Velocity

If possible, the minimum velocity in two-phase lines should be about 3 m/s to minimize slugging of separation equipment. This is particularly important in long lines with elevation changes.

20.5 Pressure Drop

The pressure drop in two-phase steel piping system may be estimated using a simplified Darcy equation.

$$\Delta P_{100} = 6.254 \times 10^6 \frac{f_m . W_T^2}{d^5 . \rho_m} \ kPa/100m \qquad \textbf{(Eq. 34)}$$

Where:

W_T	is total liquid plus vapor rate, in (kg/h);
$\rho_m(rho)$	is gas/liquid density at flowing temperature and pressure, in (kg/m³) (calculated as shown in Equation 32).

The use of the equation should be limited to a 10% pressure drop due to inaccuracies associated with changes in density.

W_T may be calculated using the following derived equation:

$$W_T = 1.2225\, Q_g . G + 1000\, Q_L . S_L \ (kg/h) \qquad \textbf{(Eq. 35)}$$

PART THREE
TRANSMISSION PIPELINES FOR: 1) LIQUID 2) GAS

21. SCOPE

The transmission line as related to the requirements of this Engineering Standard is a pipeline transporting gas or liquid and also two-phase flow from oil fields to the ship loading points or production Units such as refineries and natural gas plants. This Standard presents different methods for economical calculations of pressure loss, required diameter, for transmission of a specific quantity of crude oil, products and natural gas to the terminal under consideration.

22. SIZING CRITERIA

Although pressure loss is primary criterion in determining line size, the following points should be taken into consideration when designing a pipeline.

22.1 Design consideration should be given to flow velocity within a range which will minimize corrosion. The lower limit of the flow velocity range should be that velocity which will keep impurities suspended in the commodity, thereby minimizing accumulation of corrosion matter within the pipeline.

The upper limit of the velocity range should be such that erosion-corrosion cavitation, or impingement attack will be minimal.

22.2 Intermittent flow conditions should be avoided where possible. If operating criteria dictate the need for intermittent flow, design consideration should be given to obtaining an operating velocity which will pick up and sweep away water or sediment that accumulates in lower places in the line during periods of no flow.

22.3 If water, sediment or other corrosive contaminates are expected to accumulate in the pipeline, design should include loading and receiving pigtraps. Operating procedures should be developed and implemented for adequate cleaning (see also NACE RP 0175-75).

23. CRUDE OIL PIPELINES

The following formula have been found to give results corresponding fairly closed to those observed in operation for typical Iranian crude oil properties.

23.1 For pipe sizes less than or equal to DN 750, pressure loss shall be calculated using the Service Pipeline Co. Formula.

For turbulent flow up to Re = 170000

$$\Delta P = 26.83 \times 10^{8}\ \frac{Q_L^{\,1.748} . \gamma^{0.2518} . S}{d^{4.748}}\ kPa/100m \qquad \text{(Eq. 36)}$$

23.2 For pipe sizes greater than DN 750 the SHELL/MIT formula shall be used for pressure loss calculations. But application of SHELL/MIT formula for diameter of less than DN 750 also gives an acceptable results:

$$\Delta P_{100} = 2.54 \times 10^{10} \times \frac{f.S.Q^2_L}{d^5}\ kPa/100m \qquad \text{(Eq. 37)}$$

Where:

For viscous flow (laminar) $f = 0.00207\ \frac{1}{R_{em}}$

For turbulent flow $$f = 0.0018 + 0.00662 \left[\frac{1}{R_{em}}\right]^{0.355}$$

$$R_{em} = \frac{45.65.Q_L.S}{\mu.d} \qquad \textbf{(Eq. 38)}$$

Viscous flow	R_{em} = 0.1 to 0.135
Turbulent	R_{em} > 0.4
Indeterminate region	R_{em} = 0.135 to 6.4

Where:

R_{em}	is Reynolds number modified Re/7742;
Q_L	is flow rate, in (m³/h);
S	is relative density, (dimensionless);
μ*(mu)*	is absolute viscosity, in centipoise (cP);
d	is inside diameter, in (mm).

23.3 Crude oils generally tend to carry free water along the pipeline, this water is a potential source of corrosion. If the velocity of crude oil is too low, the water stratifies to the bottom of pipe and corrosion may occur. It is advisable to maintain a certain minimum velocity in order to keep the water from stratification.

25. NATURAL GAS LIQUIDS (NGL) PIPELINES

25.1 Natural gas liquids is generally in natural gas processing plants (dew point depression process). Its components typically range from C_2 to C_9 and its relative density is typically about 0.55.

25.2 Pressure drop calculations should be made using an appropriate method as described for liquid handling.

25.3 In transmission of NGL by pipe, pressure loss should not cause vaporization and consequently create two-phase flow. Therefore in the time of calculations of pressure loss; actual temperature & pressure of line should be regarded.

25.4 Due to its higher vapor pressure, a two-phase flow conditions must be avoided by maintaining an adequately, high minimum pressure along the pipelines.

26. NATURAL GAS PIPELINES

26.1 For gas pipelines up to about DN 300 the Weymouth and Panhandle formulas have been found to give satisfactory results. The Weymouth formula is applicable where the operating pressure is less than 450 kPa, Weymouth formula:

$$Q_g = 0.00494 \left[\frac{T_o}{P_o}\right] . d^{8/3} \left[\frac{P_1^2 \quad P_2^2}{G.T_{ave}Z_{ave}L}\right]^{0.5} \qquad \textbf{(Eq. 39)}$$

26.2 For line pressures greater than 450 kPa the Panhandle revised or "B" equation should be used:

$$Q_g = 14.14 \times 10^{3} \left[\frac{T_o}{P_o}\right]^{1.02} . d^{2.53} \left[\frac{P_1^2 \quad P_2^2}{G^{0.961}.T_{ave}Z_{ave}L}\right]^{0.5} . E \qquad \textbf{(Eq. 40)}$$

The efficiency factor "E" decreases with increasing flow rate for fully turbulent flow. It generally varies between 0.88-0.94.

26.3 The compressibility factor "Z" is calculated at pipeline average pressure which is given by:

$$P_{ave} = 2/3 \left[P_1 + P_2 - \frac{P_1 P_2}{P_1 + P_2} \right] = 2/3 \left(\frac{P_1^3 - P_2^3}{P_1^2 - P_2^2} \right) \qquad \textbf{(Eq. 41)}$$

26.4 For natural gas pipelines with diameter greater than DN 300 the IGT/AGA formula should be used:

$$Q_g = 75.75 \times 10^{-5} \left[\frac{T_o}{P_o} \right] \left[\frac{P_1^2 - P_2^2 - \dfrac{0.06834\, G\, (h_2 - h_1)\, P_{ave}^2}{T_{ave} Z_{ave}}}{G.L.Z_{ave} T_{ave}} \right]^{0.5} \times d^{2.5} \times \left[4 \log_{10} \frac{3.7d}{\varepsilon} \right] \qquad \textbf{(Eq. 42)}$$

26.5 In gas transmission lines changes in elevation may seem to have a negligible contribution to the overall pressure drop, but it turns out that, particularly in high pressure lines this contribution could be appreciable.

26.6 When corrosion inhibitor is being injected into gas transmission line, particular attention must be paid to gas velocity. High gas velocities tend to decrease the effectiveness of corrosion inhibitors. At design stage, it would be helpful to consult the inhibitor manufacturer for limiting velocities.

26.7 In long gas transmission lines when excessive pressure drop is encountered, the final temperature might even drop below ambient. This phenomenon called "Joule-Thomson effect in pipelines" should be watched for, particularly when the gas contains water vapor and hydrate formation is suspected.

26.8 Due to operating problems normally a transmission line is not designed to handle two-phase (gas-liquid).

26.9 Exceptions are flow and gas lines between oil and gas wells and separation Unit or system. Sometimes rich gas gathering networks also exhibit two-phase behavior.

26.10 To get a reasonable evaluation one must resort to computer application. In recent years several computer methods have been developed for predicting the behavior of two-phase flow in pipelines. But, due to complex nature of two-phase phenomenon, inter phase changes along the line, the effect of elevation changes, etc.; a general agreement on the best methods available does not exist. Each method has it relative merits in its particular applications. Although it is not intended to present calculation methods for two-phase flow in transmission pipelines in this Standard, the following points are noteworthy as for as process design of these systems is concerned.

26.10.1 The effect of liquids accumulating in low sections of natural gas pipelines should be taken into account in the process design stage. consideration should be given to the incorporation of liquid knock-out traps where desirable and where permitted.

26.10.2 Particular attention should be given to the additional pressure required if it is intended to remove accumulated liquids or internal deposits by pigs. The factor can be critical in hill country.

PART FOUR
TWO PHASE FLOW

27. TWO PHASE FLOW SIZING PROCESS PIPING

27.1 Two-phase flow resistance is calculated in two main steps:

a) A possible flow pattern is selected by calculating coordinates of flow region chart.

b) Unit pressure losses are determined by calculating vapor-phase unit loss only, corrected by applicable two-phase flow correlation.

Where ΔP_{100} (liquid) is calculated assuming only liquid is flowing in the pipe and ΔP_{100} (vapor) is calculated assuming only vapor is flowing in the same size pipe. X remains constant for one set of flow conditions.

27.1.1 Two-phase flow regions

The two-phase flow patterns are shown in Table 3 first column. The selection of one of these flow patterns is made using Fig. 1. The borders of the various flow pattern regions in Fig. 1 are shown as lines. In reality these boarders are rather broad transition zone.

27.1.2 Baker parameters

A particular flow region can be determined by the Baker parameters B_x and B_y. Using data supplied or usually available to the process piping designer, the Baker parameters can be expressed as follows:

$$B_y = K \frac{W_V}{A\sqrt{\rho_L \cdot \rho_V}}$$

K = 7.1 × 10⁶ SI

K = 2.16 English **(Eq. 43)**

"B_y" depends on the vapor phase flow rate, on vapor and liquid densities, and on the pipe size. The practical significance of the later variable is that by changing pipe diameters, the type of flow might also be changed, which is in turn, also changes friction losses in pipe.

$$B_x = K\,(W_L / W_V)\,(\sqrt{\rho_L \cdot \rho_V} / \rho_L^{2/3})\,(\mu_L^{1/3} / \sigma_L)$$

K = 212 SI

K = 531 English **(Eq. 44)**

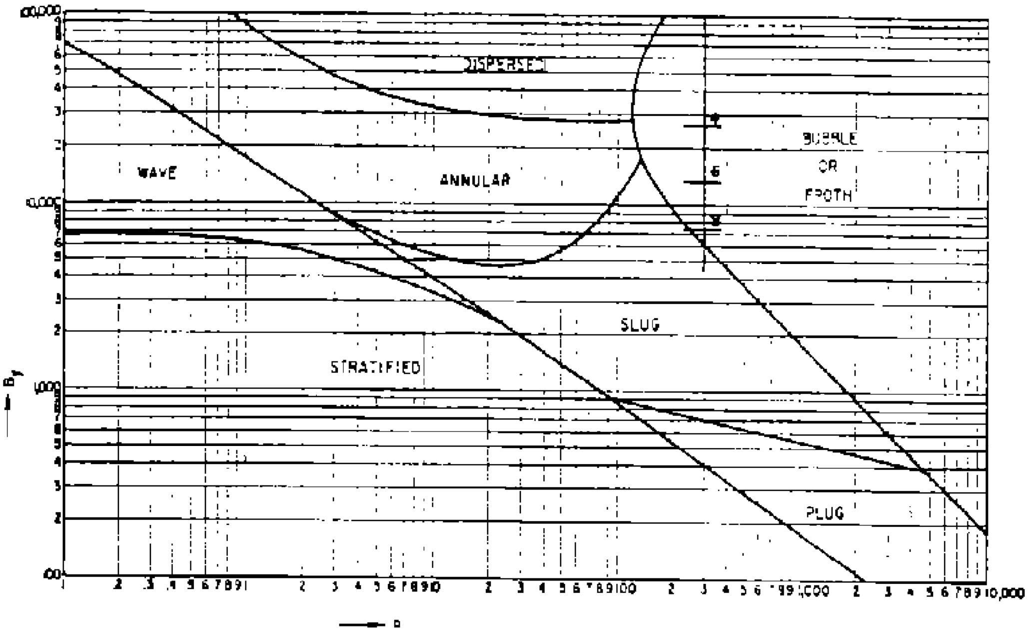
Note:

Percent liquid/percent vapor, can be substituted for W_L/W_v and $\sqrt{\rho_L \rho_V}/\rho_L^{2/3} = (\rho_V^{0.5} \sqrt{\ } \rho_L^{0.166})$ and also $\mu_L^{1/3} = \mu_L^{0.33}$.

As Equation 44 shows "B_x" depends on the mass-flow ratio and the physical properties of the liquid and vapor phase. Once calculated, it does not change with alternative pipe diameters. The position of the B_x line in Fig. 1 changes only if the liquid-vapor mixture proportion changes and, to a much lesser extent, if the physical properties of the concurrently flowing liquid and vapor changes. This can occur in long pipe lines where relatively high friction losses reduce the pressure. Consequently, the vapor content of the mixture in equilibrium increases with corresponding decrease in vapor density. The B_x line will shift somewhat to the left.

The intersection of B_x and B_y on Fig. 1 determines the flow region for the calculated liquid-vapor proportion and physical properties of the liquid and vapor. With increasing vapor content, the intersection point moves up and to the left.

It is suggested that the designer calculate B_y first. If $B_y \geq 80{,}000$ the flow will fall in dispersed flow region for hydrocarbon liquids of normal viscosities, then, the long multiplication of B_x does not have to be calculated. This can be expected when the vapor content is 25 percent or more of the total mass flow rate.



TWO-PHASE FLOW REGION SELECTION GRAPH
Fig. 1

27.2 Two-Phase Flow Unit Loss

There are different methods to calculate Unit losses for vapor-liquid mixtures. At this stage the Lockhard-Martinelli method widly used in the chemical industries mostly for horizontal Pipes. In oil & gas transmission lines there are different correlations methods which the BBM (Begg's-Brill-Moody) is the most popular. The selected method for calculations should be approved by the Company.

However, the calculations of the unit losses for vapor-liquid mixtures is based on the method of Messrs. Lockhart and Martinelli. Only the essential necessary relationship are repeated here and used with the customary data of practical process piping design. The general equation for calculating two-phase flow unit losses is:

$$\Delta P_{100}\,(two\text{-}phase) = \Delta P_{100}\,(vapor)\,(\phi^2) \qquad \textbf{(Eq. 45)}$$

Calculate the pressure drop of the vapor phase, assuming that there is only vapor flowing in the pipeline. It is assumed that the two-phase flow is isothermal and turbulent in both liquid and vapor phases and that the pressure loss is less than 10% of the absolute upstream pressure.

In Equation 45 Φ is the two-phase flow modulus, is a function of the Lockhart-Martinelli two-phase modulus X as follows:

$$X = \left(\frac{\Delta P_{L100}}{\Delta P_{V100}}\right)^{0.5} \qquad \textbf{(Eq. 46)}$$

Correct the calculated ΔP_{100} (vapor) with the correlations listed in Table 3 second column.

The form of the correlations are identical:

$$\Phi = a.x^b \qquad \text{(Eq. 47)}$$

Where "a" includes the vapor-phase flow rate and the pipe cross section and "b" is a constant, except for annular flow where in "a" and "b" only pipe diameters appear as variants.

27.2.1 As mentioned previously X is the Lockhart-Martinelli, two-phase modulus:

$$X^2 = \Delta P_{100}\,(liquid)/\,\Delta P_{100}\,(vapor) \qquad \text{(Eq. 48)}$$

Inserting Darcy's (or Wiesbach's) equation in the numerator and denominator of Equation 48 (deleting the identical constants and symbols), the two-phase flow modulus will be equal to:

$$X^2 = (W_L/W_v)^2\,(\rho_v/\rho_L)\,(f_L/f_v) \qquad \text{(Eq. 49)}$$

$$X = (W_L/W_v)\,(\rho_v/\rho_L)^{0.5}\,(f_L/f_v)^{0.5} \qquad \text{(Eq. 50)}$$

f_L and f_v is the liquid and vapor-phase friction factor. It can be obtained directly by calculating the liquid vapor phase Reynolds numbers and using the Moody friction factor diagram for commercial steel pipes (see Appendix A).

Usually both phases fall in transitional turbulent zone, where the friction factor varies with varying Reynolds numbers, f_L/f_v increases with the increasing vapor content of liquid.

27.2.2 Reynolds number are calculated separately for the vapor and liquid-phase using the same diameter, corresponding flow rates and viscosities.

$$R_e = 353.7W/d.\mu \qquad \text{(Eq. 51)}$$

A convenient form of Darcy's equation for unit pressure loss calculations for liquid or vapor as previously stated is:

$$\Delta P_{100} = 62530\,(f_D.W^2)/d^5.\rho \qquad bar/100\ m$$

Use the same diameter for liquid and vapor-phase and corresponding phase flow rate, density and friction factor.

27.2.3 As with all line sizing procedures pipe sizes must be estimated first. After pipe size selection, flow region coordinates can be calculated and the flow type determined. After finding the vapor-phase unit loss and applicable two-phase flow correlation (in Table 3), two-phase flow Unit losses can be calculated by Equation 45. Φ can be also found through Fig. 3 or Table 3.

27.2.4 The initial diameter of two-phase flow in short process pipes can be sized through the vapor-liquid reboiler return line formula:

$$dmm = 18.73\,\left(\frac{W_V}{\rho_V}\right)^{0.42}.\left(\frac{\rho_V}{\rho_L.x}\right)^{0.167} \qquad \text{(Eq. 52)}$$

27.2.5 The over-all friction loss in the pipe between two points will be:

$$\Delta P = \Delta P_{100}\,(two\text{-}phase)(L/100) \qquad \text{(Eq. 53)}$$

Where L is the equivalent length of the pipe and fittings in meter (Appendix G). Knowing the pipe configurations, L can be computed conveniently using Appendix G.

28. TWO-PHASE FLOW PATTERNS

In determining the type of flow in a process pipeline designers generally refer to as Table 3 which is known as Baker map in Fig. 1. The seven types of flow in horizontal pipe are shown in Table 3. These patterns (in order of increasing gas flow rate at a constant liquid flow rate) are as follows.

28.1 Bubble or Froth Flow

This pattern, characterized by bubbles of gas moving along the upper part of the pipe at approximately the same velocity as the liquid, develops when bubbles of gas dispersed throughout the liquid. It occurs for liquid superficial velocities of about 1.5-4.6 m/s (5-15 ft/sec) and gas superficial velocities of about 0.305-3.05 m/s (1-10 ft/sec).

28.2 Plug Flow

Alternate plugs of liquid and gas move along the upper part of the pipe and liquid moves along the bottom of the pipe. Plug flow occurs for liquid velocities less then 0.61 m/s (2 ft/sec) and gas velocities less then about 0.91 m/s (3 ft/sec).

28.3 Stratified Flow

The liquid-phase flows along the bottom of the pipe while the gas flows over a smooth liquid-gas interface. It occurs for liquid velocities less then 0.15 m/s (0.5 ft/sec) and gas velocities of about 0.61-3.05 m/s (2-10 ft/sec).

28.4 Wave Flow

Wave flow is similar to stratified flow except that the gas is moving at a higher velocity and the gas-liquid interface is distributed by waves moving in the direction of flow. It occurs for liquid velocities less than 0.305 m/s and gas velocities from about 4.6 m/s.

28.5 Slug Flow

This pattern occurs when waves are picked up periodically by the more rapidly moving gas. These form frothy slugs that move along the pipeline at a much higher velocity than the average liquid velocity. This type of flow causes severe and in most cases dangerous vibration in equipment because of the impact of the high-velocity slugs against fittings.

28.6 Annular Flow

In annular flow liquid forms around the inside wall of the pipe and gas flows at a high velocity through the central core. It occurs for gas velocities greater than 6.1 m/s (20 ft/sec).

28.7 Dispersed Spray or Mist Flow

Here all of the liquid is entrained as fine droplets by the gas-phase. Dispersed flow occurs for gas velocities greater than 61 m/s (200 ft/sec).

29. VELOCITY LIMITATIONS

Depending on the flow regime, the liquid in a two-phase flow system can be accelerated to velocities approaching or exceeding the vapor velocity. In some cases, these velocities are higher than what would be desirable for process piping. Such high velocities lead to a phenomena known as "erosion-corrosion", where the corrosion rate of material is accelerated by an erosive material or force (in this case, the high-velocity liquid).

29.1 An index based on velocity head can indicate whether erosion-corrosion may become significant at a particular velocity and can be used to determine the range of mixture densities and velocities below which erosion-corrosion should not occur.

This index is:

$$\rho_{ave} \cdot V^2_{ave} \leq 10000 \quad \text{English} \quad \text{(Eq. 54)}$$
$$\leq 14800 \quad \text{SI}$$

Where the mixture or average density:

$$\rho_{ave} = \frac{W_L + W_G}{W_L / \rho_L + W_G / \rho_G} \quad \text{(Eq. 55)}$$

$$V_{ave} = V_G + V_L \quad \text{(Eq. 56)}$$

$$= W_G / (3600\ \rho_G .A) + W_L / (3600\ \rho_L .A)$$

For erosional velocity refer to Clause 20 in Part two herein.

29.2 The corrosional velocity limitations may be determined exprimentaly. The limitation for corrosional velocity is based on the inhibitor film resistance and experiments. It is normally less than erosional velocity and is basis for design velocity in pipelines.

30. MAINTAIN THE PROPER REGIME

In addition to keeping the velocity-density product within the acceptable range, one must also maintain the proper flow regime.

30.1 Most importantly, slug flow must be avoided. Slug flow unit losses in process piping are generally not calculated, because it causes various mechanical and process problems.

First water hammer may occur as the slug of liquid impinges on pipe and equipment walls at every changes of flow direction. This could result in equipment damage due to erosion-corrosion. Second, if slug flow enters a distillation column, the alternating composition and density of the gas and liquid slugs cause cycling of composition and pressure gradients along the length of the column. The cycling causes problems with product quality and process control.

30.2 Slugs can form in variety of ways. They may be created as a result of wave formation of the liquid-gas interface in a stratified flow. As the liquid waves grow large enough to bridge the entire pipe diameter, the stratified flow pattern breaks down into slug flow. Slugs can also form due to terrain effects, such as liquid collecting at a sag in a pipeline and blocking the gas flow.

The pressure in this blocked gas rises until it forces the accumulated liquid downstream in the form of a slug. Changes in the inlet flow rate can cause slugs as well. When the flow rate increases, the liquid inventory in the pipeline decreases and the excess liquid forms a slug or a series of slugs. Pigging-the removal of water from the line to minimize corrosion-can cause very large slugs as the line’s entire liquid inventory is swept a head of the pig.

30.3 Slug Flow Can be Avoided in Several Ways

30.3.1 By reducing lines sizes to a minimum permitted by available pressure differentials.

30.3.2 By designing parallel pipe lines that will increase flow capacity without increasing the overall friction loss.

30.3.3 By using valves auxiliary pipe runs to regulate alternative flow rates and avoid slug flows.

30.3.4 By using a low point effluent drain or by-pass or other solutions.

30.3.5 By arranging the pipe configurations to protect against slug flow.

31. DESIGN CONSIDERATIONS

The significance of the two-phase flow theories and experiments to the process piping designer is threefold. First, it has been shown that if the liquid content of a vapor line increases, the friction loss can be many times its original single phase pressure loss. Second, it has been shown that for a given set of vapor liquid properties and physical properties a characteristics flow pattern develops. Third between the various flow patterns, unit losses can differ.

- The flow patterns shall be checked for normal flow, max. flow and turn-down flow.

- Two phase calculations normally are done by computer packages. They use different correlations for flow patterns and equilibrium of states equations.

- The package and the selected correlations should be confirmed by the Company.

31.1 Two-phase flow piping design through rational and empirical steps. Limitations, generalizations and simplifications have been introduced for providing practical methods of design. The assumptions connected with this Standard are: that two-phase flow is isothermal, that two-phase flow is turbulent in both the liquid and vapor phase, that the flow is steady (liquid and vapor move with the same velocity), that the pressure loss is not more than 10 percent of the absolute downstream pressure.

31.2 In long lines, vapor moves faster than the liquid. Thus there are varying densities along the pipe length. In vertical lines, the static head back pressure will not be the same as that calculated with average densities.

31.3 Process piping systems have flexibility in the distribution of pressure loss and control valves can operate within a wide range of available pressure differentials.

31.4 Designer can capitalize on the characteristics to obtain the optimum piping and component size and layout.

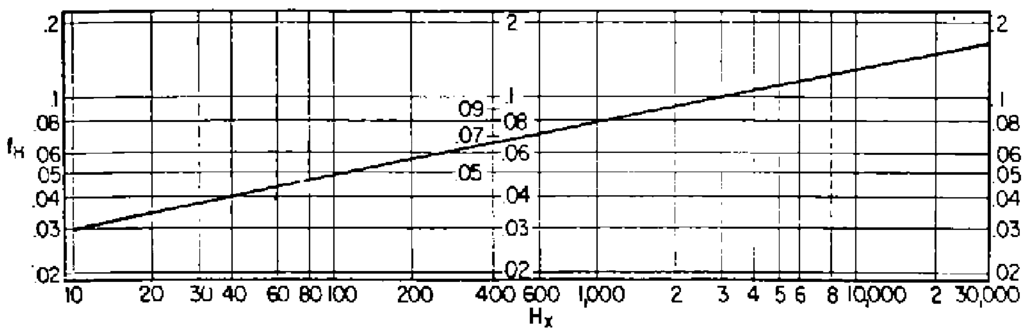
31.5 In general, the criterion for selecting a suitable line size is that the pipe diameter must be sufficiently small to have the highest possible velocity, but larger enough to stay within available pressure differentials and allowable pressure drops.

31.6 Normally slug flow is undesirable in two-phase flow pipelines. Since flow velocity is one of the factors which influence the flow regime, due consideration should be given to this phenomenon at design stage.

31.7 Pipe size over design must be particularly watched for. Low velocities tend to increase the chances for the slug flow.

31.8 Two-phase flow pipelines are unstable. When changes are made to the pressures or to flow rates, the pipeline readjusts itself gradually and equilibrium may not be re-established for several days. A two-phase pipeline is both a pipeline and extremely long storage tank; changes in flow conditions cause the liquid to go into or come out of storage.

The liquid in two-phase pipes passing over hills tends to run back down hill and accumulate in the valleys. Increasing the velocity in the uphill portion of the line reduces the liquid holdup in the valleys and lowers the pressure drops.

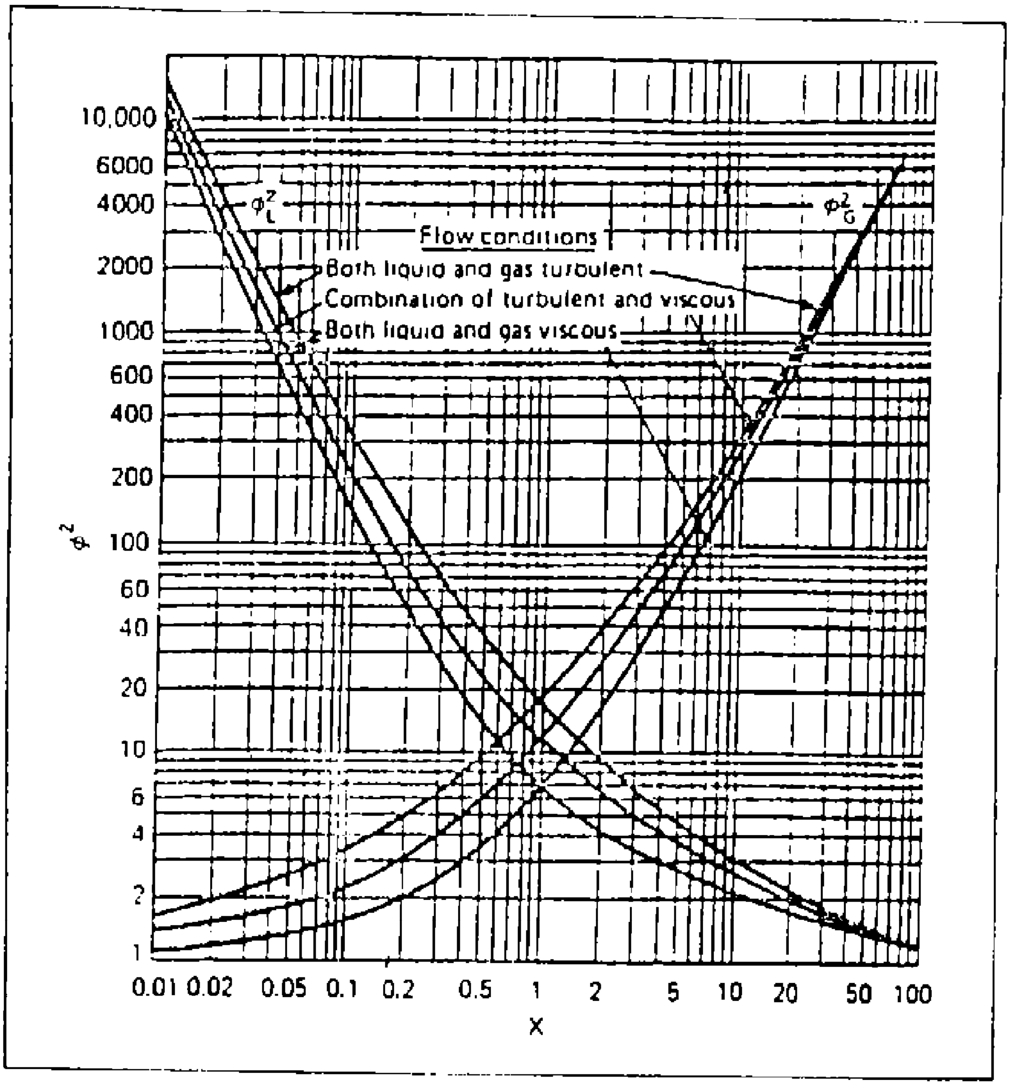


WAVE FLOW UNIT LOSS
Fig. 2

According to Table 3 Item 5 for the wave regime, pressure drop calculations are as follows:

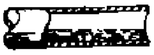
Wave flow unit loss. procedure:

1) Calculate Huntington Correlation; $H_x = (W_l/W_v)(\mu_l/\mu_v)$
2) with H_x enter the graph and read the friction factor, ρf_H;
3) calculate the unit loss, ΔP_{100} *(wave)* $= 62530\, f_H\, (W_v)^2/d^5\ \rho_v$.



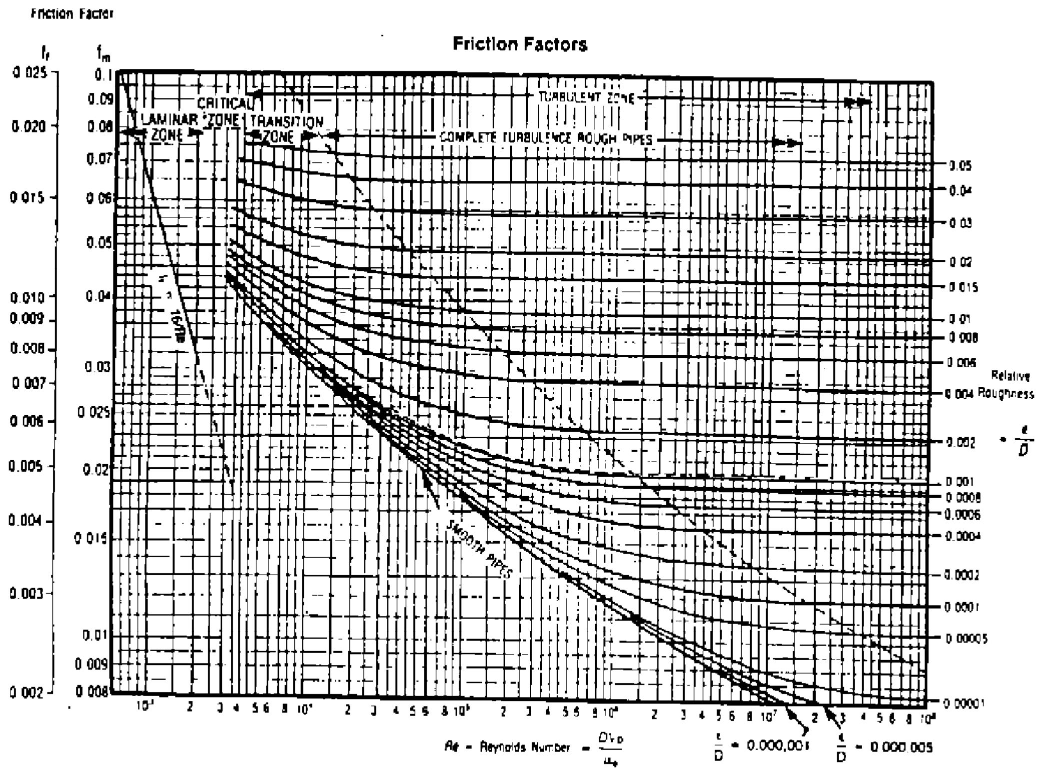
LOCKHART-MARTINELLI CORRELATIONS CAN BE USED TO DETERMINE TWO-PHASE PRESSURE DROP
Fig. 3

TABLE 3 - SUMMARY OF TWO-PHASE FLOW UNIT FRICTION LOSS CALCULATIONS

TWO-PHASE FLOW PATTERNS	TWO-PHASE FLOW CORRELATIONS
  1. DISPERSED NEARLY ALL OF THE LIQUID IS ENTRAINED AS SPRAY BY THE GAS	$Q=\exp(A_0+A_1.\ln x+A_2.\ln x^2+A_3.\ln x^3)$ $A_0=1.4659$ $A_1=0.49139$ $A_2=0.04887$ $A_3=-0.000349$
  2. ANNULAR LIQUID FORMS A FILM AROUND THE INSIDE WALL OF PIPE AND GAS FLOWS AT A HIGH VELOCITY AS A CENTRAL CORE	$\phi = aX^b$ $a=4.8 - 0.0123d$ $b=0.343 - 0.000826$ d=PIPE INSIDE DIA.,mm d=250 FOR 300 mm AND LARGER SIZES
  3. BUBBLE BUBBLES OF GAS MOVE ALONG AT ABOUT THE SAME VELOCITY AS THE LIQUID	$\phi = \frac{4.18X^{0.75}}{(W_l/A)^{0.1}}$
 4.STRATIFIED LIQUID FLOWS ALONG THE BOTTOM OF PIPE AND THE GAS FLOWS ABOVE OVER A SMOOTH GAS-LIQUID INTERFACE	$\phi = \frac{86.782X110^{-2}X}{(W_l/A)^{0.8}}$ FOR LONG HORIZONTAL PIPES
 5.WAVE SIMILAR TO STRATIFIED EXCEPT THE INTERFACE IS DISTURBED BY WAVES MOVING IN THE DIRECTION OF FLOW	USE SCHNEIDER-WHITE-HUNTINGTON CORRELATION FOR LONG HORIZONTAL PIPES CALCULATION ON FIG.2
  6.SLUG WAVES ARE PICKED UP PERIODICALLY IN FORM OF FROTHY SLUG WHICH MOVE AT A MUCH GREATER VELOCITY THAN THE AVERAGE LIQUID VELOCITY	$\phi = \frac{2.6294X^{0.815}}{(W_l/A)^{0.5}}$ AVOID SLUG FLOW
 7.PLUG ALTERNATIVE PLUGS OF LIQUID AND GAS MOVE ALONG THE PIPE	$\phi = \frac{3.4156X^{0.855}}{(W_l/A)^{0.17}}$

APPENDICES

APPENDIX A
MOODY FRICTION FACTOR CHART



(to be continued)

APPENDIX A.1

A.1 Method of Solution

The well-behaved nature of Colebrook function-no maxima-no minimal or inflection points-suggests Newton-Raphson interpolation, as a method of solving this equation.

A.1.1 The Newton-Raphson method is applied with convergence to ±0.0001, this requires differentiating the objective function. The Newton-Raphson method is of the form:

$$X_{n+1} = X_n - \frac{f(X_n)}{f'(X_n)} \quad \textbf{(Eq.A.1)}$$

Where:

$n = 1, 2, 3, 4, \ldots\ldots n_{max}$

x_n is the guessed root of equation given by $f(x_n) = 0$. $f(x_n)$ is the objective function. $f'(x_n)$ is the value of differentiate of the objective function. The n is iterative counter and n_{max} is the maximum iteration.

For this purpose, one defines a function:

$$y = x + 2 \log_{10} [A + B_x] \quad \textbf{(Eq. A.2)}$$

Where:

$$X = 1/\sqrt{f} \quad \textbf{(Eq. A.2-a)}$$
$$A = \varepsilon/3.7D \quad \textbf{(Eq. A.2-b)}$$
$$B = 2.51/R \quad \textbf{(Eq. A.2-c)}$$

The problem then is to find an x such that y = 0 . To use the Newton- Raphson method the first derivation of y with respect to x is required:

$$y' = \frac{d_y}{d_x} = 1 + \frac{2.B.\log_{10} e}{A + B_x} \quad \textbf{(Eq. A.3)}$$

A.1.2 Successive values of x are then obtained from Equation (A.1) until the value of $y(x_n)$ is sufficiently close to zero. The corresponding value of x is then substituted into Equation (A.2-a) in order to compute the friction factor.

A.1.3 As an initial guess of the value x, the friction factor used for the case of completely turbulent flow (flat parts of curves in Moody chart where f is independent of R_e). This value may be computed explicity from the form of Colebrook equation which is valid in this region:

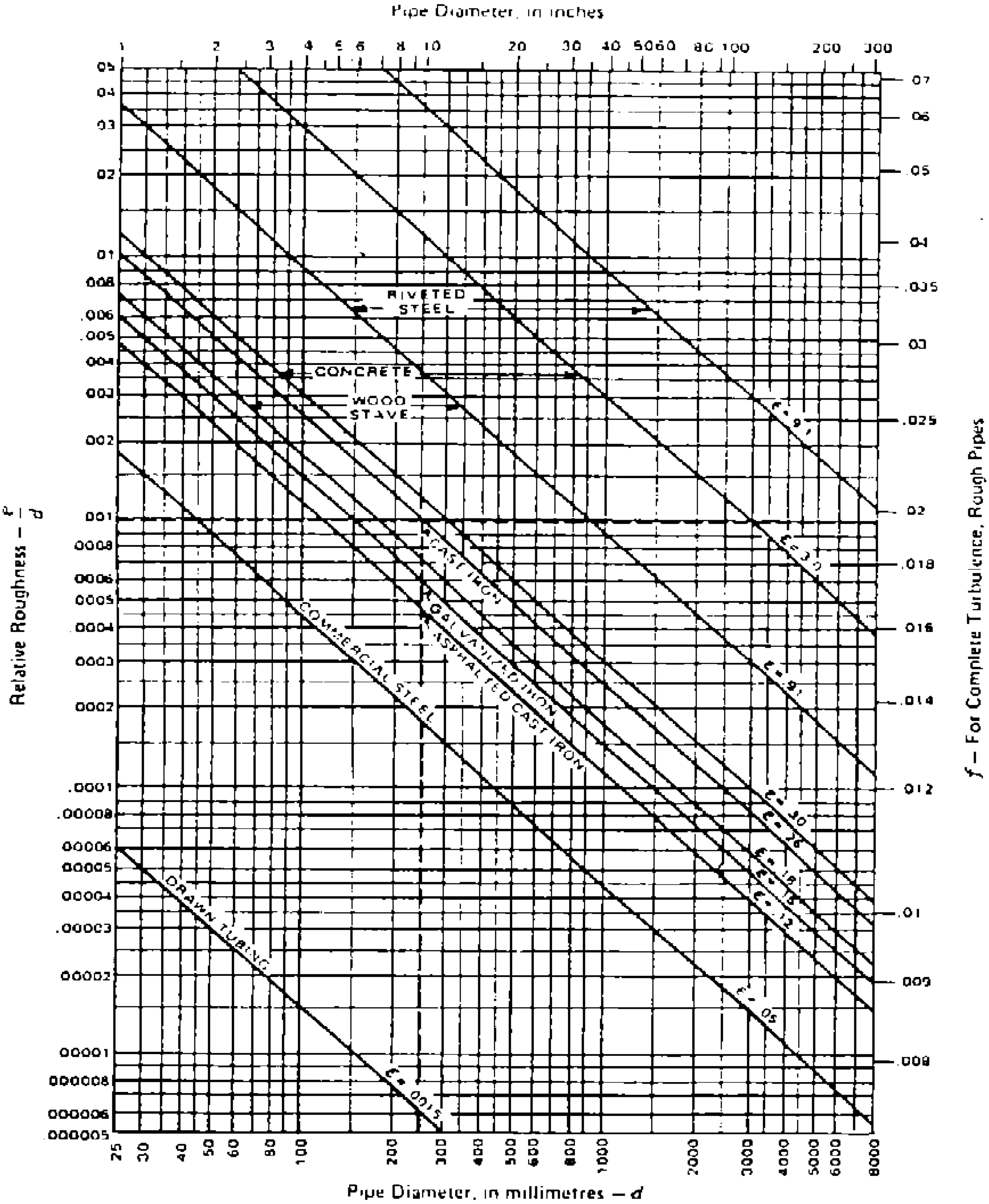
$$X = 1/\sqrt{f} = -2 \log_{10} (\varepsilon/3.7d) \quad \textbf{(Eq. A.4)}$$

A.1.4 For the liquid when $R_e < 2000$, friction factor can be found through Hagen-Poiseuille’s equation:

$$f_D = 64/R_e \quad \textbf{(Eq. A.5)}$$
$$f_f = 16/R_e \quad \textbf{(Eq. A.6)}$$
$$f_D = 4f_f \quad \textbf{(Eq. A.7)}$$

APPENDIX B
RELATIVE ROUGHNESS CHART

Relative roughness of pipe materials and friction factors for complete turbulence.



(Absolute Roughness ε is in millimeters)

Problem: Determine absolute and relative roughness, and friction factor, for fully turbulent flow in a cast iron pipe, 250 mm int. diam.

Solution: Absolute roughness (ε) = 0.26 .... Relative roughness (ε /d) = 0.001 .... Friction factor at fully turbulent flow (f) = 0.0196.

APPENDIX C
HAZEN-WILLIAMS COEFFICIENT (FRICTION FACTOR) "C"

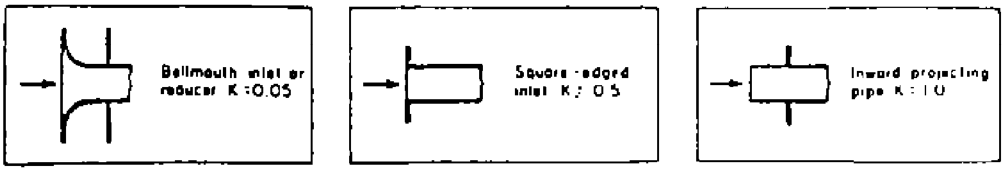
The Factor "C" shall not be confused with the Darcy-Weisbach-Colebrook friction factor "f", these two friction factors are not in any way related to each other.

TYPE OF PIPE	VALUES OF C		
	RANGE-HIGH = BEST SMOOTH WELL LAID-LOW = POOR OR CORRODED	AVERAGE VALUE FOR CLEAN NEW PIPE	COMMONLY USED VALUE FOR DESIGN PURPOSES
Cement-Asbestos	160-140	150	140
Fiber	—	150	140
Bitumastic-Enamel-Lined iron or steel centrifugally applied	160-130	148	140
Cement-Lined iron or steel centrifugally applied	—	150	140
Copper, brass, lead, tin or glass pipe and tubing	150-120	140	130
Wood-Stave	145-110	120	110
Welded and seamless steel	150-80	130	100
Intenor riveted steel (no projecting rivets)	—	139	100
Wrought-Iron, cast-iron	150-80	130	100
Tar-Coated cast-iron	145-50	130	100
Girth-Riveted steel (projecting rivets in girth seams only)	—	130	100
Concrete	152-85	130	100
Full-Riveted steel (projecting rivets in girth and horizontal seams)	—	115	100
Vitrified, spiral-riveted steel (flow with lap)	—	110	100
Spiral-Riveted steel (flow against lap)	—	100	90
Corrugated steel	—	60	60

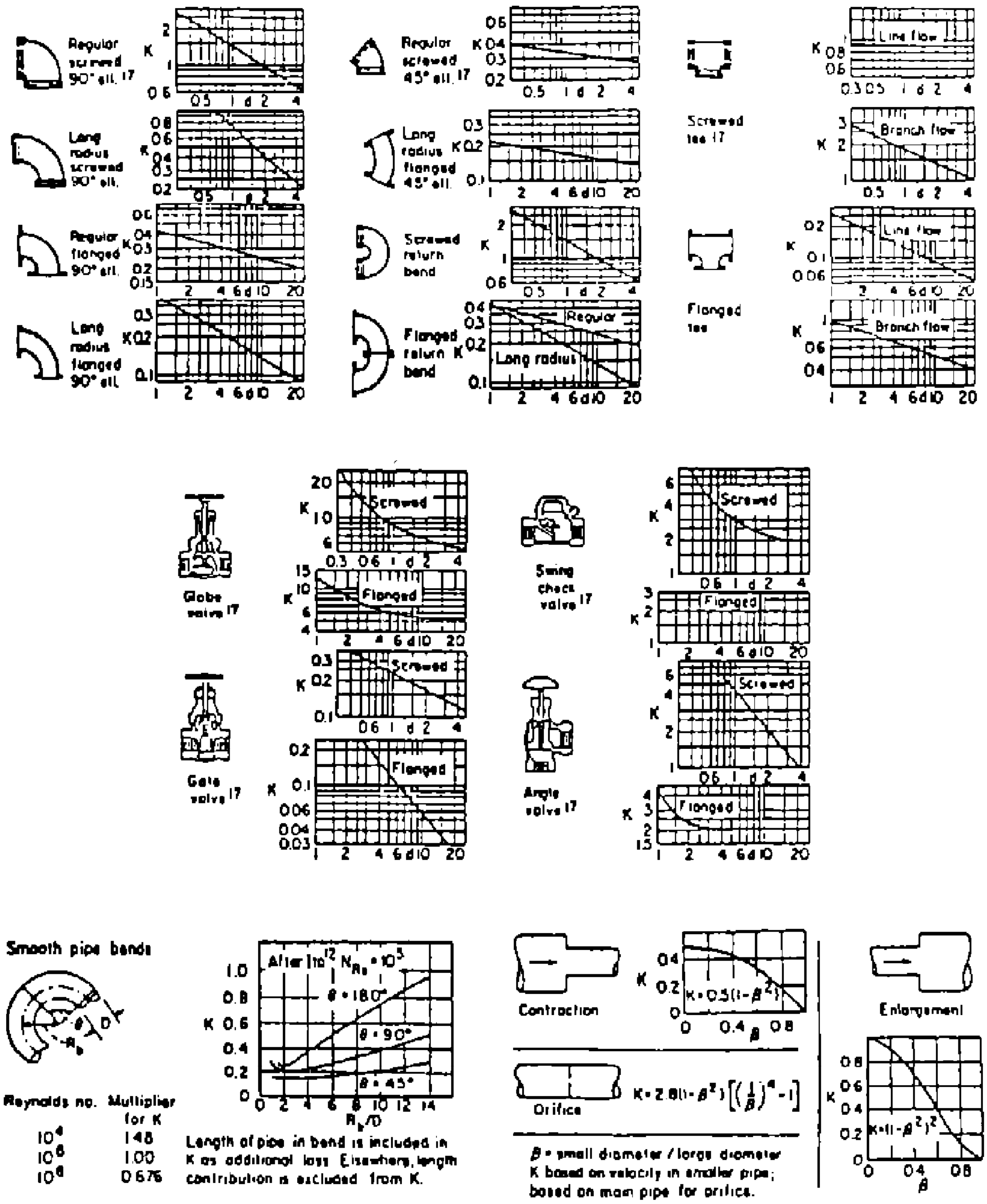
VALUE OF C	150	140	130	120	110	100	90	80	70	60
$(100/C)^{1.85}$	0.47	0.54	0.62	0.71	0.84	1.00	1.22	1.50	1.93	2.57

Multiplier (Basis C = 100)

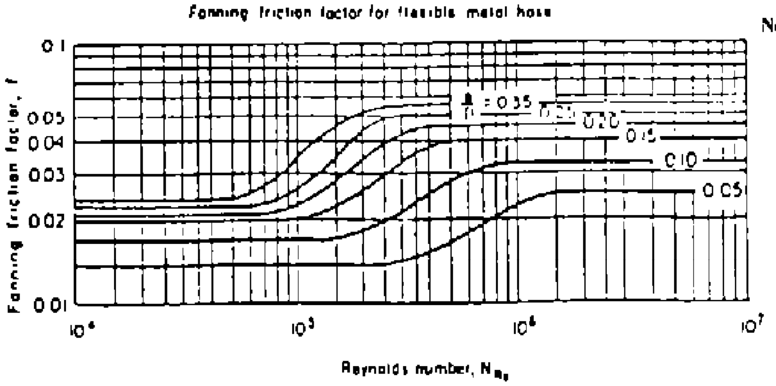
APPENDIX D
RESISTANCE COEFFICIENTS FOR VALVES AND FITTINGS



Note: K decreases with increasing wall thickness of pipe and rounding of edges

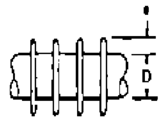


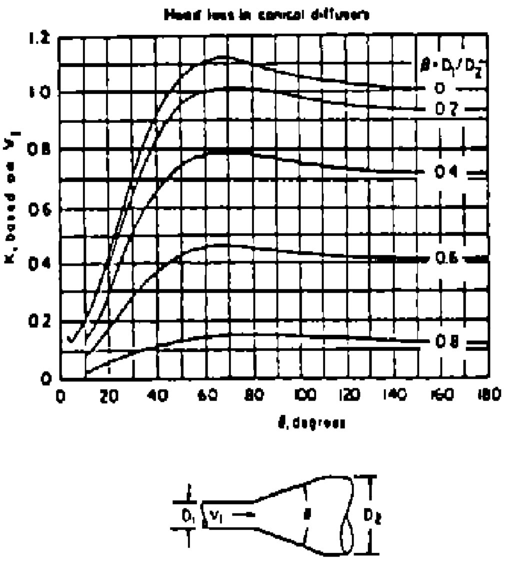
APPENDIX E
RESISTANCE COEFFICIENTS FOR VALVES AND FITTINGS

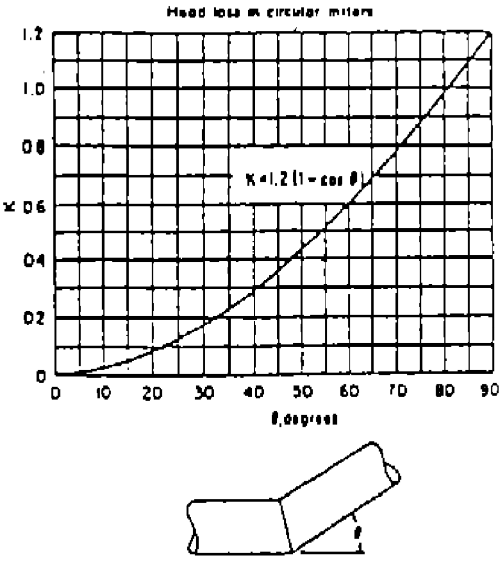


Typical dimensions

Nominal Size (in)	ID mm	ε/D
1/2	13.2	0.250
3/4	19.1	0.275
1	25.4	0.187
1-1/2	38.1	0.200
2	50.8	0.161
3	75.2	0.143
4	101.6	0.081



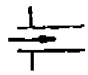
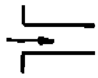
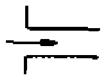
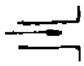




		K	
Plug cock valve	θ = 5°	0.05	θ is angle between pipe axis and plug cock axis
	10°	0.29	
	20°	1.56	
	40°	17.3	
	60°	206.0	

		K	
Butterfly valve	θ = 5°	0.24	θ is angle between pipe axis and flopper plate
	10°	0.52	
	20°	1.54	
	40°	10.8	
	60°	118.0	

APPENDIX F
RESISTANCE DUE TO PIPE ENTRANCE AND EXIT

SHAPE	K	SERVICE
	0.78	INWARD PROJECTING PIPE ENTRANCE
	0.50	SHARP EDGES ENTRANCE
	0.23	SLIGHTLY ROUNDED ENTRANCE
	0.04	WELL ROUNDED ENTRANCE
	1.0	PROJECTING PIPE EXIT
	1.0	SHARP ENDED EXIT
	1.0	ROUNDED EXIT

APPENDIX G
EQUIVALENT LENGTHS OF VALVES AND FITTINGS

Representative equivalent length in pipe diameters (L/D) of various valves and fittings.

DESCRIPTION OF PRODUCT				EQUIVALENT LENGTH IN PIPE DIAMETERS (L/D)
Globe valves	Conventional	With no obstruction in flat, bevel, or plug type seat	Fully open	340
		With wing or pin guided disc	Fully open	450
	Y-Pattern	(No obstruction in flat, bevel, or plug type seat)		
		- With stem 60 degrees from run of pipe line	Fully open	175
		- with stem 45 degrees from run of pipe line	Fully open	145
Angle valves	Conventional	With no obstruction in flat, bevel or plug type seat	Fully open	145
		With wing or pin guided disc	Fully open	200
Gate valves	Conventional		Fully open	13
			Three-Quarters open	35
	wedge disc, double disc or plug disc		One-Half open	160
			One-Quarter open	900
Gate valves	Pulp stock		Fully open	17
			Three-Quarters open	50
			One-Half open	260
			One-Quarter open	1200
	Conduit pipe line		Fully open	3**
Gheck valves	Conventional swing	3.4 ♣...	Fully open	135
	Clearway swing	3.4 ♣...	Fully open	50
	Globe lift or stop	13.8 ♣...	Fully open	same as globe
	Angle lift or stop	13.8 ♣...	Fully open	same as angle
	In-Line ball	17.2 vertical & 1.7 horizontal ♣...	Fully open	150
Foot valves with strainer		With poppet life-type disc 2.1 ♣...	Fully open	420
		With leather hinged disc 2.8 ♣...	Fully open	75
Butterfly valves (DN 150 and larger)			Fully open	20
Cocks	Straight through	Rectangular plug port area equal to 100% of pipe area	Fully open	18
	Three-Way	Rectangular plug port area equal to 80% of pipe area (fully open)	Flow straight through	44
			Flow through branch	140

(to be continued)

APPENDIX G (continued)

Fittings	90 Degree standard elbow 45 Degree standard elbow 90 Degree long radius elbow		30 16 20
	90 Degree street elbow 45 Degree street elbow Square corner elbow		50 26 57
	Standard tee	With flow through run With flow through branch	20 60
Close pattern Return bend			50
Enlargement*	Sudden	d/D = ¼ d/D = ½ d/D = ¾	37 24 8
	Standard reducer	d/D = ½ d/D = ¾	30 8
Contraction*	Sudden	d/D = ¼ d/D = ½ d/D = ¾	76 30 11
	Standard reducer	d/D = ½ d/D = ¾	16 3

Note:

*** Equivalent lengths are in terms of small diameter.**

*** Values applicable up to DN 600.**

**** Exact equivalent length is equal to the length between flange faces or welding ends.**

♣ Minimum calculated pressure drop (kPa) across valve to provide sufficient flow to lift disc fully.

| For limitations and effect of end connections.