10

FLOW OF FLUIDS

This chapter is an overview of a more detailed discussion in the book *Petroleum Fluid Flow Systems*, also published by Campbell Petroleum Series.[10.1]

The flow of any fluid through a line containing no heat input or output may be considered isothermal and adiabatic. One may therefore combine the First and Second Laws of Thermodynamics, using these assumptions to write an equation.

$$\int V dP + \frac{g}{g_c} \Delta X + \frac{(\Delta v)^2}{2\, g_c} = -W_f - W \qquad (10.1)$$

Where:
- V = volume of the fluid
- P = pressure of the fluid
- ΔX = change in elevation of the fluid
- Δv = change in velocity of the fluid
- W_f = work lost due to friction
- W = work done by the system
- g = gravitational force
- g_c = mass-force conversion constant

Equation 10.1 is as far as one can go using thermodynamics alone, for it provides no way of solving for the term W_f, lost work.

The total irreversible effects (W_f) are usually attributed to friction. The work done in overcoming friction through a distance dL is proportional to the surface in contact with the fluid, approximately proportional to the square of velocity and proportional to the fluid density. By including a proportionality constant, f, the equation may be written.

$$\text{frictional resistance} = (f)(dL)(\pi d)\left(v^2/\,2\, g_c\right)(\rho) \qquad (10.2)$$

where the definition of variables for Equations 10.2-10.4 may be found on page 4.

The weight of fluid in the pipe is the length dL multiplied by the cross-sectional area and the fluid density. Any frictional work would be represented by the frictional resistance moved through distance dL. Combining these concepts into Equation 10.2 then yields dW_f:

$$dW_f = \frac{(f)(dL)(\pi d)\left(v^2/\,2\, g_c\right)(\rho)(dL)}{(\pi/4)(d^2)(\rho)(dL)} \qquad (10.3)$$

Simplifying:

$$dW_f = \frac{2\, f\, v^2\, dL}{g_c\, d} \tag{10.4}$$

which integrated between the limits of 0 to W_f and 0 to L gives

$$W_f = \frac{2\, f\, L\, v^2}{g_c\, d} \tag{10.5}$$

Equation 10.5 is called the *Fanning friction factor* equation. Other forms of this equation have been published which differ only in the value of the coefficient ahead of "f." In this book the Fanning equation will be used in all correlations.

Combination of Equations 10.1 and 10.5 then gives

$$\int VdP + \frac{g}{g_c}\Delta X + \frac{(\Delta v)^2}{2\, g_c} = -\frac{2\, f\, L\, v^2}{g_c\, d} - W \tag{10.6}$$

which is the basic equation for flow often called the Bernoulli equation.

When solving a problem for a line containing equipment like pumps, the line is divided into sections as shown below.

The pump sections (1-2 and 3-4) are solved by the equation

$$\int VdP = -W_{theor.} \tag{10.7}$$

The potential energy, kinetic energy and friction drop terms are incorporated into a single efficiency term (E) that replaces these terms.

For a pump,

$$W_{actual} = \frac{W_{theor.}}{E}$$

For a hydraulic expander,

$$W_{actual} = (E)(W_{theor.})$$

For the line section 2-3 no work is done. So, W is eliminated from an equation like 10.6 to solve for the pressure drop – flow rate behavior, giving Equation 10.8:

$$\int VdP + \frac{g\, \Delta X}{g_c} + \frac{\Delta\, v^2}{2\, g_c} = -\frac{2\, f\, L\, v^2}{g_c\, d} \tag{10.8}$$

EVALUATION OF FRICTION FACTOR

The friction factor "f" must be evaluated empirically. One common correlation is the Moody plot shown in Figure 10.1.[(10.2)]

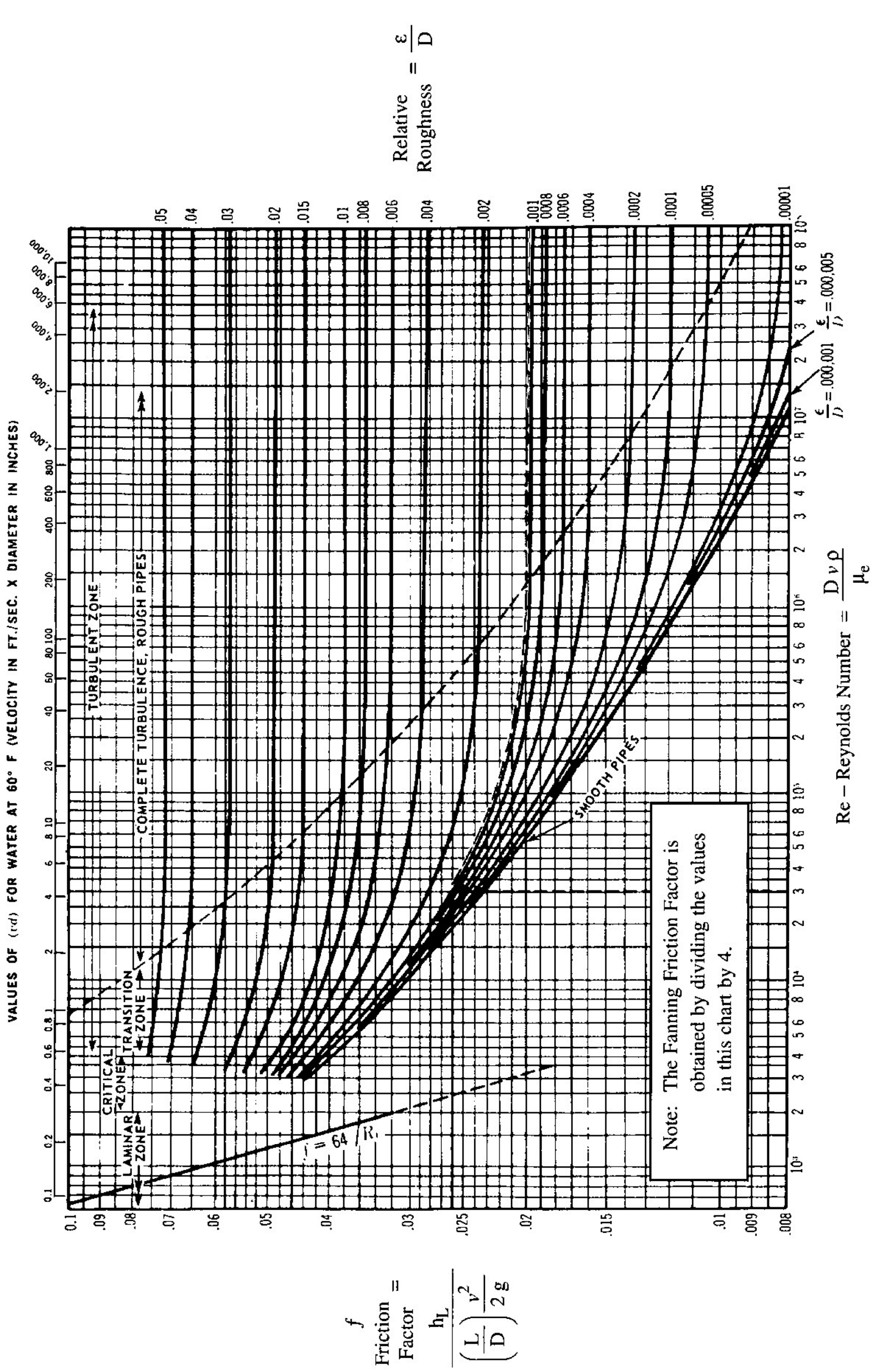


Figure 10.1 Friction Factors Using Moody Correlation[(10.4)]
(which combines the Colebrook transition between the smooth pipe equation and the fully turbulent zone)

Factor "f" depends on pipe roughness and diameter, and the fluid involved. The dimensionless Reynolds number on the abscissa of Figure 10.1 may be written in various forms.

$$Re = \frac{d\,v\,\rho}{\mu} = \frac{q\,\rho}{0.785\;d\,\mu} = \frac{m}{0.785\;d\,\mu} \qquad (10.9)$$

Any units that make Re dimensionless may be used. For fluid flow applications the following units are most common.

Where:			Metric	English
	d	= internal line diameter	m	ft
	v	= velocity	m/s	ft/sec
	ρ	= density	kg/m^3	lbm/ft^3
	μ	= viscosity	kg/m·s	lbm/ft-sec
	P	= pressure	Pa	lbf/ft^2
	g_c	= mass/force conversion constant	$1.0\left(\frac{kg\cdot m}{N\cdot s^2}\right)$	$32.17\left(\frac{lbm-ft}{lbf-sec^2}\right)$
	g	= gravitational force	$9.81\ m/s^2$	$32.17\ ft/sec^2$
	L	= length of line	m	ft
	ΔX	= change of elevation	m	ft
	m	= mass flow rate	kg/s	lbm/sec
	q	= volumetric flow rate	m^3/s	ft^3/sec
	f	= dimensionless factor		
	0.785	= $\pi/4$		
	1 cp	= 0.001 kg/m·s = $6.72 \times 10^{-4}\ lb_m/ft\text{-}sec$		

This type of correlation applies only to *Newtonian fluids.* Gas and almost all liquids of concern in this book are Newtonian in nature.

Non-Newtonian Liquids

One often assumes (erroneously) that the liquids being used are Newtonian fluids. In fact, many fluids do not truly fall within this category. There are several general classifications of liquids based on their rheological behavior: *Newtonian, Bingham Plastic, Pseudoplastic* and *Dilatant.*

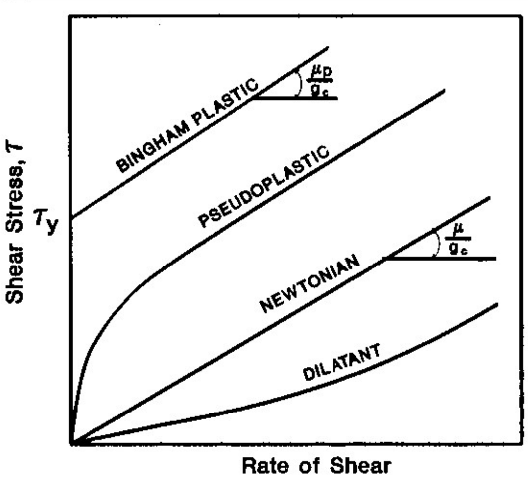


Figure 10.2 Shear Curves of Newtonian and Non-Newtonian Fluids

The general curves for these are shown in Figure 10.2.

Of these, the Bingham plastic and Newtonian liquids are the most common models employed. For a Newtonian liquid, the relationship is expressed by the equation

$$\tau = \frac{\mu}{g_c}\left(-\frac{dv_r}{dr}\right) \qquad (10.10)$$

Where: μ = absolute viscosity
$-(dv_r/dr)$ = shear rate for laminar flow in a circular pipe

As shown in Figure 10.2, this equation produces a straight line on cartesian coordinates starting at the origin. The slope of the line is μ/g_c.

A Bingham plastic, on the other hand, is a straight line that does not pass through the origin. It is represented by the equation

$$\tau - \tau_y = \frac{\mu_p}{g_c}\left(-\frac{dv_r}{dr}\right) \tag{10.11}$$

Where: τ_y = the intercept
μ_p = plastic viscosity, a value found from the line slope

A Bingham plastic will not flow until the shear stress exceeds a value represented by τ_y, known as the *yield point.* Drilling fluids consisting of colloidal clay particles suspended in liquid approximate Bingham plastic behavior.

Pseudoplastic and *dilatant* fluids possess no yield point and exhibit nonlinear behavior. In addition, there are fluids where behavior is time dependent. A *thixotropic* fluid will produce a double curve. When increasing shear rates are imposed it will break down with time. If shear rate is then reduced no further, breakdown occurs. After exposure to high shear rates, this type of liquid will regain its original consistency after a period of time. This property is useful in special applications.

A *rheopectic* liquid is one where at constant shear rate the shear stress increases with time, the opposite of a thixotropic liquid.

It is common to characterize non-Newtonian liquids by an empirical power law formula

$$\tau = K\left(-\frac{dv_r}{dr}\right)^n \tag{10.12}$$

Where: K = an overall measure of the liquid viciousness
= μ/g_c for Newtonian fluid

n = 1.0 for Newtonian fluid
= 0.0 – 1.0 for pseudoplastic fluids
= >1.0 for dilatant fluids

Equation 10.12 is a straight line on a log-log plot. The deviation of "n" from 1.0 (in either direction) is a measure of the degree of non-Newtonian behavior.

The detailed calculation of the behavior of non-Newtonian liquids is beyond the scope of this book, but more details are available in Reference 10.3 and many references covering rheology of liquids.

Most oils may be treated as Newtonian liquids, particularly those with a relative density below 0.9. However, some emulsions of oil and water may exhibit non-Newtonian characteristics.

NEWTONIAN LIQUID FLOW

The Moody chart (Figure 10.1) is a very sound correlation if one can estimate roughness. Unfortunately, one never knows what it is at any one time, and it changes with time. Initially the pipe may contain mill scale from its manufacture. After use this mill scale may be removed by liquid action and the pipe wall becomes smoother. On continued use erosion/corrosion, scale formation and the like may increase roughness.

Figure 10.3 is based on field tests. The smooth curve is for 8 in. pipe or larger, or smooth tubing. As diameter increases, roughness has less effect.

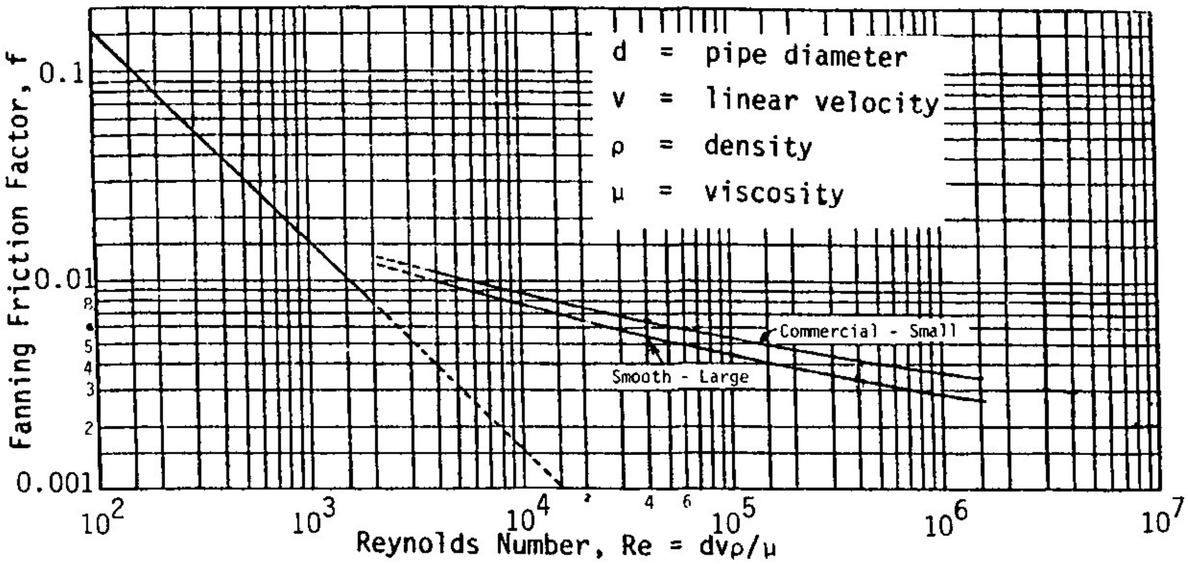


Figure 10.3 Simplified Correlation for Fanning Friction Factor

The equations of the three lines in Figure 10.3 are represented closely by the following equations.

$$\text{Re} < 2000 \;,\quad f = 16/\text{Re} \tag{10.13}$$

$$\text{Re} > 4000 \;,\quad f = 0.042/\text{Re}^{0.194} \quad \text{large pipe} > 20 \text{ cm [8 in.]} \tag{10.14}$$

$$\text{Re} > 4000 \;,\quad f = 0.042/\text{Re}^{0.172} \quad \text{small pipe} \le 20 \text{ cm [8 in.]} \tag{10.15}$$

Figures 10.1 and 10.3 have served for years as a basis in evaluation of friction factors. In the world of computers, equations are more convenient. Churchill and Usagi have proposed a single equation to replace the f vs. Re plots shown.

$$f = 2\left[\left(\frac{8}{\text{Re}}\right)^{12} + \frac{1}{(C_1 + C_2)^{1.5}}\right]^{0.0833} \tag{10.16}$$

Where: $C_1 = \{2.457 \ln[1/((7/\text{Re})^{0.9} + 0.27\,(\varepsilon/d))]\}^{16}$
$C_2 = (37530/\text{Re})^{16}$

For the flow of an incompressible liquid in a section of line of fixed diameter where no work is done, Equation 10.6 may be written

$$\Delta P = (P_2 - P_1) = \left(-\frac{2\,f\,L\,v^2\,\rho}{g_c\,d}\right) - \rho\,\Delta X\,(g/g_c) \tag{10.17}$$

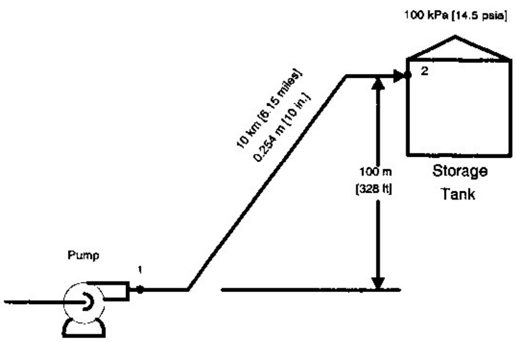
$$= (-\Delta P_f) - \rho\,\Delta X\,(g/g_c)$$

It often is more convenient to express flow rate in volumetric or mass terms rather than velocity. The v^2 in Equation 10.17 may be substituted for as follows:

$$v^2 = \frac{q^2}{(0.785)^2\,d^4} = \frac{m^2}{\rho^2\,(0.785)^2\,d^4} \tag{10.18}$$

As with Reynolds number, the substitution of these terms in the Fanning friction factor equation to obtain ΔP_f enables one to write Equation 10.17 in terms of volumetric or mass flow.

Example 10.1: A 0.254 m [10 in.] crude oil line handles 6400 m^3 [40 200 bbl] per day of a 0.83 Rel. ρ oil. The oil viscosity is 9.5 cp. The flow line is 10 km [6.15 miles] long and discharges to a 100 kPa [14.5 psi] storage tank. The tank battery is 100 m [328 ft] higher than the pump discharge. Determine the necessary pump discharge pressure.



(This type of problem is frequently found in new design work. No attempt is made to calculate losses caused by pipe fittings. Such losses are relevant, however, in real world situations and should be calculated. Normally they are added as an equivalent length to the actual length of the pipeline. See later text. The ΔP_f, then, should usually include any such fitting friction losses.)

Metric:

$$d = 0.254 \text{ m}$$

$$v = \frac{6400}{(86400)(0.785)(0.254)^2} = 1.46 \text{ m/s}$$

$$\mu = (9.5)(0.001) = 0.0095 \text{ kg/m·s}$$

$$\rho = (0.83)(1000) = 830 \text{ kg/m}^3$$

$$\frac{d\, v\, \rho}{\mu} = \frac{(0.254)(1.46)(830)}{0.0095} = 3.2 \times 10^4$$

From Figure 10.3, f = 0.0058

$$\Delta P_f = \frac{(2)(0.0058)(10\ 000)(1.46)^2(830)}{(1.0)(0.254)} = 8.08 \text{ x } 10^5 \text{ N/m}^2 = 8.08 \text{ x } 10^5 \text{ Pa}$$

$$\rho\, \Delta X\, g/g_c = (830)(100)(9.81) = 814\ 230 \text{ N/m}^2$$

$$(100\ 000 - P_1) + 814\ 230 = -8.08 \text{ x } 10^5$$

$$P_1 = 1.72 \text{ x } 10^6 \text{ Pa} = 1.72 \text{ MPa(a)}$$

English:

$$\Delta P_f = \frac{(2)(0.0058)(6.15)(5280)(4.79)^2(51.8)}{(32.2)(0.833)} = 16\ 690 \text{ lbf/ft}^2$$

$$\rho\, \Delta X = (51.8)(328)\ \ 16\ 990 \text{ lbf/ft}^2$$

$$(2088 - P_1) + 16\ 990 = -16\ 690 \quad , \quad P_1 = 35\ 678 \text{ lbf/ft}^2$$

In Example 10.1 ΔX is positive since the liquid is going uphill. If the liquid had been flowing downhill, ΔX would have been a negative number. Also, in this example no effort was made to specify exact pipe ID, the nominal diameter was used. In reality, pipe ID varies from the values shown in tables, since some pipe specifications as much as 12.5% deviation in pipe wall thickness. As a blunt, practical matter, the use of exact diameters causes no real increased precision in predicting long line performance.

Calculation of Pipe Diameter

Situations often occur where it is desired to calculate the line diameter (d) needed for a given flow rate and pressure drop. The friction factor, f, and Reynolds number, Re, are a function of diameter and velocity terms. Therefore, the use of the correlation of f plotted against Re involves a trial-and-error solution if diameter is desired.

A direct solution for diameter is possible by algebraic manipulation of the basic equations.

Solving for "d" yields

$$d = 1.265\, q^{0.4} \left[\frac{f\, L\, \rho}{(\Delta P_f)\, g_c} \right]^{0.2} \qquad (10.19)$$

The friction factor in Equation 10.19 can be found by many methods. It is convenient to use Equations 10.14 and 10.15. Substituting these equations for "f" into Equation 10.19 gives

For small pipe,

$$d = 0.649\, q^{0.379}\, \rho^{0.172}\, \mu^{0.036} \left[\frac{L}{(\Delta P_f)\, g_c} \right]^{0.207} \qquad (10.20)$$

For large pipe,

$$d = 0.647\, q^{0.376}\, \rho^{0.168}\, \mu^{0.041} \left[\frac{L}{(\Delta P_f)\, g_c} \right]^{0.208} \qquad (10.21)$$

where units are defined as those listed on page 4.

Example 10.2: Determine the diameter for an oil flow rate of 0.0416 m^3/s [1.46 ft^3/s], relative density of 0.79, a total pressure drop allowance of 500 kPa [72.5 psi] over 20 km [12.4 miles], and a viscosity of 10 cp. Assume large pipe ($d \geq 10$ inches):

Metric:

$$d = 0.6474\, (0.0416)^{0.376}\, [0.79\, (1000)]^{0.168}\, [10\, (0.001)]^{0.041} \left[\frac{20\,000}{(500\,000)\; 1.0} \right]^{0.208}$$

$$= 0.2547 \text{ m}$$

English:

$$d = 0.6474\, (1.46)^{0.376}\, [0.79\, (62.4)]^{0.168}\, [10(0.000672)]^{0.041} \left[\frac{(12.42)(5280)}{(72.5)(144)(32.2)} \right]^{0.208}$$

$$= 0.834 \text{ ft} = (10 \text{ inches})$$

Calculation of Line Capacity

The calculation of the capacity of a line with a given pressure drop is trial-and-error when using a plot of friction factor vs. Re. However, a direct solution is possible. Starting with the same basic equations as in the calculation of the diameter, eliminating q instead of d, and using Equations 10.14 and 10.15 for "f,"

For small pipe,

$$q = \frac{3.127\, d^{2.64}}{\rho^{0.453}\, \mu^{0.094}} \left[\frac{(\Delta P_f)\, g_c}{L} \right]^{0.547} \qquad (10.22)$$

For large pipe,

$$q = \frac{3.180\, d^{2.661}}{\rho^{0.446}\, \mu^{0.107}} \left[\frac{(\Delta P_f)\, g_c}{L} \right]^{0.553} \qquad (10.23)$$

Calculation of Pressure Drops

A similar derivation approach for the remaining variable normally required in field problems, ΔP_f, yields the following equations for normal pipe friction.

$$\text{Small Pipe,} \quad \frac{\Delta P_f}{L} = \frac{q^{1.828}\,\rho^{0.828}\,\mu^{0.172}}{8.038\ g_c\ d^{4.828}} \qquad (10.24)$$

$$\text{Large Pipe,} \quad \frac{\Delta P_f}{L} = \frac{q^{1.806}\,\rho^{0.806}\,\mu^{0.194}}{8.081\ g_c\ d^{4.806}} \qquad (10.25)$$

Example 10.3: $q = 0.0416\ m^3/s$ [1.46 ft^3/sec] $\mu = 10$ cp
$\rho = 790\ kg/m^3$ [49.3 lb_m/ft^3] L = 20 km [12.42 miles]
d = 0.254 m [0.833 ft]

Assume fittings increase L by 5% and elevation changes insignificant

Metric:
$$\Delta P_f = \frac{(0.0416)^{1.806}\,(790)^{0.806}\,[10\,(0.001)]^{0.194}\,(20\,000)(1.05)}{8.081\ (1.0)(0.254)^{4.806}} = 535\ 418\ \text{Pa}$$

English:
$$\Delta P_f = \frac{(1.46)^{1.806}\,(49.3)^{0.806}\,[10\,(0.000\,672)]^{0.194}\,(12.42)(5280)(1.05)}{8.081\ (32.17)(0.833)^{4.806}}$$
$$= 11\ 071\ \text{psf} = 76.9\ \text{psi}$$

Valves and Fittings

Equations 10.20-10.25 adequately describe flow of Newtonian fluids in straight runs of commercial steel pipe under conditions of fully turbulent flow. The pressure drop through valves and fittings must be addressed separately. The most common method used to account for this additional pressure drop is the use of equivalent lengths (L_e). Each fitting (e.g. valve, tee, el, etc.) is assigned an L_e based on its size and geometry. These are documented extensively in fluid flow literature. Reference 10.4 provides an excellent review of the theory of this approach and Reference 10.5 tabulates L_e's for the most common valves and fittings encountered in the hydrocarbon industry. Figure 10.4 is a handy nomograph for estimating Le for various types of fittings.

Once the L_e of a fitting has been determined it is simply added to the length of the straight pipe sections. The pressure drop calculation is then performed on the total pipe length including the L_e additions.

In the planning mode, how does one estimate the pressure drop through a pipe section before the actual pipe layout has been determined. References 10.16 and 10.7 detail a method whereby pressure drops can be estimated before detailed drawings have been completed. This is useful for predicting pumping and compression requirements. The correlation accounts for a typical number of pipe fittings and valves installed in a piping system.

$$L_e/L = 1 + (0.347\ d^{1/2} + 0.216)\ F_c \qquad (10.26)$$

Where: L_e/L = total equivalent lengths per length of straight pipe
d = nominal pipe diameter, inches
F_c = complexity factor (see table at right)

	F_c
Very complex piping manifolds	4
Manifold type piping	2
Normal piping	1
Long reasonably straight runs	0.5
Utility supply lines, outside battery limits	0.25

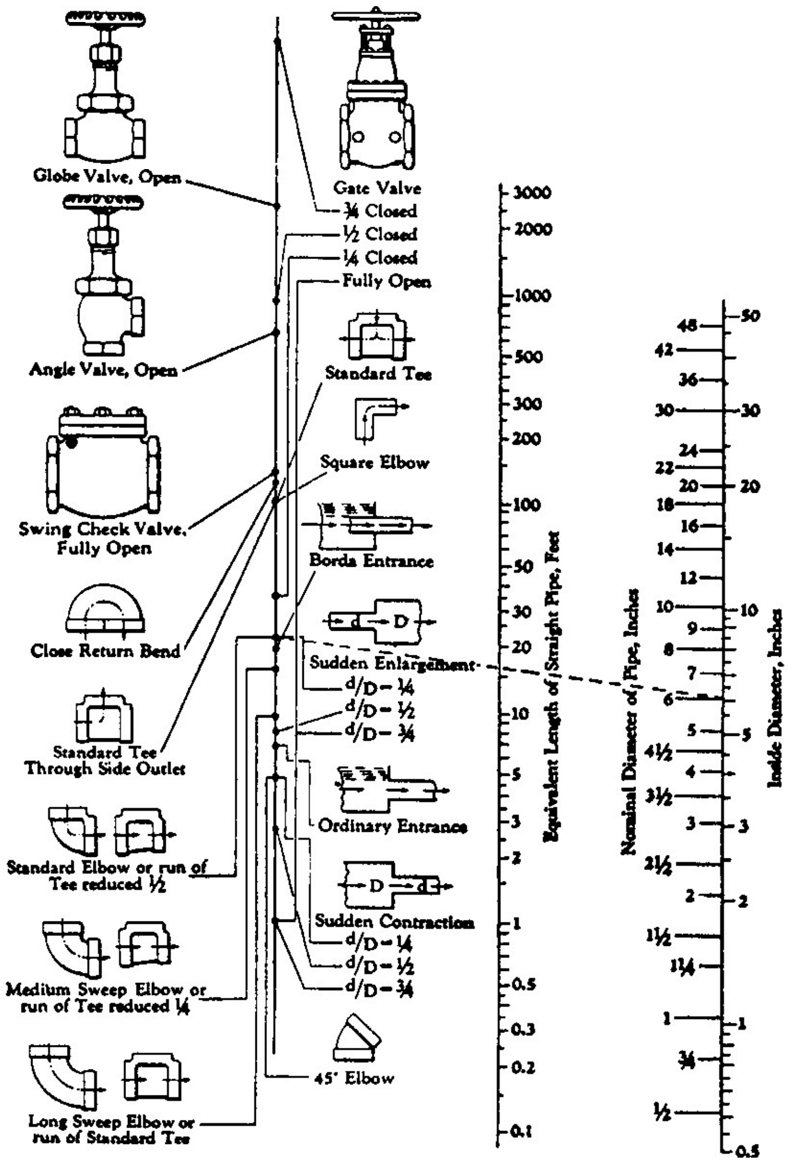
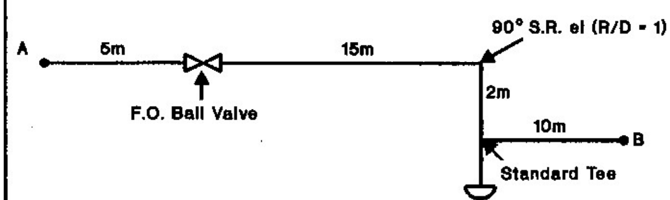


Figure 10.4 Resistance of Valves and Fittings to Flow of Fluids

Example 10.4: Calculate the effective length of piping in the system shown below. Pipe size is 0.254 m [10 in.].



From Figure 10.4

10" F.O. Ball Valve	L_e = 2.1 m [7 ft]
10" 90° S.R. el	L_e = 5.5 m [18 ft]
10" tee (branch flow)	L_e = 14.3 m [47 ft]

For ΔP calculation, the effective length of the system

$$= 5 + 2.1 + 15 + 5.5 + 2 + 14.3 + 10 = 53.9 \text{ m } [177 \text{ ft}]$$

Economic Pipe Diameter

Equations 10.20-10.25 simply provide the relationship between flow rate, pressure drop and pipe size. They do not provide line sizing criteria. In sizing a line one is always faced with a compromise of two factors. For a given flow rate of a given fluid, as pipe diameter is increased piping cost increases. But, pressure loss decreases, which reduces potential pumping or compression costs.

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

Equation 10.27 may be used to estimate this economic diameter. It is derived from cost data and must be considered an approximation only.

$$d = \frac{A\, m^{0.45}}{\rho^{0.31}} \qquad (10.27)$$

Where:		Metric	English
d =	economic diameter	m	in.
m =	mass flow rate, in thousands of	kg/h	lb/hr
ρ =	fluid density	kg/m^3	lb/ft^3
A =	constant	0.189	2.2

Equation 10.27 is derived primarily for relatively short lengths of piping in processing and production installations where pressure drop is not particularly large. It is applicable to both vapor and liquid lines. Remember also that the most economic diameter may not be the one you should choose. There are mechanical considerations which are of concern.

Many companies provide pipe sizing quidelines in terms of velocity or pressure drop per unit length. The following guidelines have proven useful for preliminary sizing of process piping.

Liquid Lines	Recommended Velocity	
	m/s	ft/s
Noncorrosive liquid flow	2-3	7-10
Corrosive liquid flow (sour condensate, glycol, amine)	0.7-1.0	2-3
Centrifugal pump suction	0.7-1.0	2-3
Reciprocating (plunger) pump suction	0.3	1
Vapor Lines	**Recommended ΔP/L**	
	Pa/m	psi/100 ft
Natural gas: 0-700 kPa(g) [0-100 psig]	110-230	0.5-1.0
700-3500 kPa(g) [100-500 psig]	230-450	1.0-2.0
3500-14 000 kPa(g) [500-2000 psig]	450-1130	2.0-5.0

Maximum Velocity

Some company specifications limit *maximum continuous* liquid velocity to

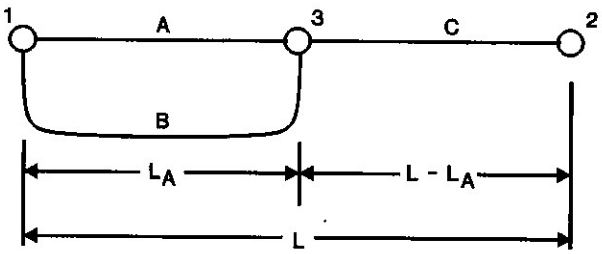
$$v_{max} = \frac{K\, A}{\rho^{0.5}} \qquad (10.28)$$

Where:		Metric	English
v =	velocity	m/s	ft/sec
ρ =	density	kg/m^3	lb/ft^3
A =	conversion constant	1.23	1.0
K =		100	100

API 14E recommends three values of K; 100 for normal design calculation, 125 for intermittant service, 150 for the "never exceed" value. These values are, in general, based on erosion corrosion limitations in bends, elbows, and fittings. Some companies have found considerably higher values of K are acceptable for straight piping with "clean" fluids.

DESIGNING LOOP SYSTEMS

The need often arises for increasing the flow rate per unit pressure drop in a system. This is normally encountered where it is desired to increase the capacity or where the flow rate is to remain the same with a lower pressure. The former is a common need in expanding systems, while the latter usually stems from older systems wherein the pipe or pumping equipment has to be derated for pressure.



The usual and most economical solution to the problem is to place one or more lines parallel to the original, either partially or throughout the entire length. The figure at left shows a schematic view of a simple loop system that may be used to illustrate the principles involved.

The loop and the line it parallels are usually equal in length throughout their common length, although conditions sometimes dictate a loop of different length.

Loop Capacity

It is often desirable to calculate the capacity of an existing looped system. The most straightforward method of analyzing this problem is to convert the loop into a single line having the length of the looped section with a diameter (d_e) which gives the line the same capacity as the looped section. An alternative approach is to fix the diameter of the line and calculate the equivalent length L_e of the line so that the line capacity is equal to that of the loop.

This method can be derived from Equation 10.22 or 10.23. Equation 10.23 will be used as an example:

$$q_A = \frac{3.180\, d_A^{2.661}}{\rho^{0.446}\, \mu^{0.107}} \left[\frac{\Delta P_f}{L_A} g_c \right]^{0.553}$$

A similar equation can be written for q_B. Now envision a line with a diameter d_e and length L_e which has the same capacity as the looped section ($q_A + q_B$).

$$q_A + q_B = \frac{3.180\, d_e^{2.661}}{\rho^{0.446}\, \mu^{0.107}} \left[\frac{\Delta P_f}{L} g_c \right]^{0.553} \qquad (10.29)$$

Remember that $(\Delta P_f)_A = (\Delta P_f)_B = (\Delta P_f)_e$. Substituting Equation 10.23 into 10.29 for q_A and q_B and canceling like terms gives

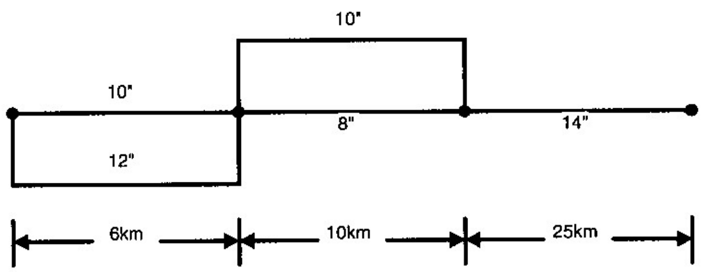
$$\frac{d_e^{2.661}}{L_e^{0.553}} = \frac{d_A^{2.661}}{L_A^{0.553}} + \frac{d_B^{2.661}}{L_B^{0.553}} \qquad (10.30)$$

By an identical approach, for small pipe,

$$\frac{d_e^{2.641}}{L_e^{0.547}} = \frac{d_A^{2.641}}{L_A^{0.547}} + \frac{d_B^{2.641}}{L_B^{0.547}} \tag{10.31}$$

Equation 10.30 can be used to convert complex looped systems into simple systems for capacity calculations.

Example 10.5: Calculate the capacity of the following system handling a 32°API gravity crude oil with a viscosity of 3 cp. Allowable pressure drop is 1000 kPa [145 psi].



Convert the first looped section into an equivalent length, Le, of 14 in. pipe –

$$\frac{14^{2.661}}{L_e^{0.553}} = \frac{10^{2.661}}{6^{0.553}} + \frac{12^{2.661}}{6^{0.553}} \quad , \quad \text{solve for } L_e \qquad L_e = 5.3 \text{ km}$$

Similarly for the second looped section –

$$\frac{14^{2.661}}{L_e^{0.553}} = \frac{10^{2.661}}{10^{0.553}} + \frac{8^{2.661}}{10^{0.553}} \quad , \quad \text{solve for } L_e \qquad L_e = 22.8 \text{ km}$$

Total effective length of system = 5.3 + 22.8 + 25 = 53.1 km

Calculate q from Equation 10.23 –

$$\gamma = 141.5/(131.5 + 32) = 0.865 \qquad \rho = 865 \text{ kg/m}^3 \ [54 \text{ lbm/ft}^3]$$

$$\mu = 0.003 \text{ kg/m·s} \ [2.02 \times 10^{-3} \text{ lbm/ft-sec}]$$

$$\Delta P/L = 1\,000\,000/53\,100 = 18.83 \text{ Pa/m} \ [633 \text{ (lbf/ft}^2\text{)/ft}]$$

$$d = 0.356 \text{ m} \ [1.17 \text{ ft}]$$

$$q = \frac{(3.180)(0.356)^{2.661}}{(865)^{0.446}\,(0.003)^{0.107}}\,[18.83]^{0.553} = 0.094 \text{ m}^3\text{/s} \ [51\,100 \text{ bpd}]$$

Loop Length

Another common application of the loop calculation is to estimate the length of loop required to increase the capacity of the system a given amount. This calculation can be made by establishing a pressure balance for the system

$$\left(\frac{\Delta P_f}{L}\right)_A (L_A) + \left(\frac{\Delta P_f}{L}\right)_C (L - L_A) = (\Delta P_f)_{total} \tag{10.32}$$

If we define $X = L_A/L$ (fraction of the system looped) we can write the equation

$$\left(\frac{\Delta P_f}{L}\right)_A (X) + \left(\frac{\Delta P_f}{L}\right)_C (1 - X) = \left(\frac{\Delta P_f}{L}\right)_{total} \qquad (10.33)$$

Equations 10.24 and 10.25 can be used to calculate $(\Delta P_f/L)_A$ and $(\Delta P_f/L)_C$. The right hand term $(\Delta P_f/L)_{total}$ is fixed by allowable pressure drop. Equation 10.33 can be solved for X.

Direct substitution of Equations 10.24 and 10.25 into Equation 10.33 yields the equations shown in Table 10.1.

TABLE 10.1
Equations for Lines in Parallel

	Lines in Parallel	
	Large Pipe	Small Pipe
Equivalent Diameter	$\frac{d_e^{2.661}}{L_e^{0.554}} = \frac{d_A^{2.661}}{L_A^{0.554}} + \frac{d_B^{2.661}}{L_B^{0.554}}$	$\frac{d_e^{2.641}}{L_e^{0.547}} = \frac{d_A^{2.641}}{L_A^{0.547}} + \frac{d_B^{2.641}}{L_B^{0.547}}$
Looping Requirements*	$X = \frac{1-(q/q_1)^{1.806}}{1-\left[\frac{d_A^{2.661}}{d_A^{2.661}+d_B^{2.661}}\right]^{1.806}}$	$X = \frac{1-(q/q_1)^{1.828}}{1-\left[\frac{d_A^{2.641}}{d_A^{2.641}+d_B^{2.641}}\right]^{1.828}}$
Entire Line Looped*	$\frac{q_1}{q} = \left[1+\left(\frac{d_B}{d_A}\right)^{2.661}\right]$	$\frac{q_1}{q} = \left[1+\left(\frac{d_B}{d_A}\right)^{2.641}\right]$
Diameter of Original and Parallel Lines the Same*	$X = 1.40\left[1-\left(\frac{q}{q_1}\right)^{1.806}\right]$	$X = 1.392\left[1-\left(\frac{q}{q_1}\right)^{1.828}\right]$
*Assumes that the length of all lines in the loop are the same length.		

For Lines in Parallel:

Where:
- d_e = diameter of a single line that is equivalent to a group of parallel lines
- d_A = diameter of the original line before looping
- d_B = diameter of a single loop line (or equivalent diameter of a group of loop lines)
- L_e = length of equivalent single line corresponding to d_e
- L_A = length of loop line d_A
- L_B = length of loop line d_B
- q = original flow rate before line is looped
- q_1 = flow rate after looping line
- X = fraction of length of original line that is looped

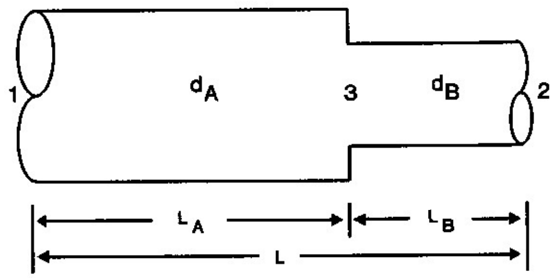
Example 10.6: It is desired to increase the capacity of an existing pipeline system by 60%. The diameter of the existing line is 12 inches. What percentage of the pipeline system must be looped if the diameter of the loop is 14 inches?

$$X = \frac{1 - (1.0/1.6)^{1.806}}{1 - \left[\dfrac{12^{2.661}}{12^{2.661} + 14^{2.661}}\right]^{1.806}} = \underline{0.706} = \underline{70.6\%}$$

Complex Liquid Gathering Systems

The situation often arises when it is convenient to use several diameters of pipe in series. Fluid principles also may be applied in this case after proper analysis of the conditions present.

In the sketch at right, L and d represent the length and diameter, respectively, of two different size lines joined in series. Therefore:



$$q_A = q_B = q_{total} \quad \text{and} \quad \Delta P_{fB} + \Delta P_{fA} = \Delta P_{f\,total}$$

However, no one equation may be written between points (1) and (2) because of the varying diameter. It should be possible to write an equation for section 1-3 and one for 3-2, solve each for q_L, and permit one to solve for the remaining unknown P_3.

A more flexible and convenient solution is afforded by determining a single-diameter line that is equivalent to the system shown. An equation in the form of 10.24 or 10.25 may be written for one section. Then the equation is written for a hypothetical section having the same diameter as the other section but of such length that the ΔP_f is the same as that first written. Then:

$$\Delta P_f = \frac{C\,L_e}{d_A^{5-n}} = \Delta P_{fB} = C\left(\frac{L_B}{d_B^{5-n}}\right) \tag{10.34}$$

where L_e is defined as the length of line of diameter d_A that will give the same friction loss as length L_B of line d_B. C is defined as all the terms of equation 10.24 or 10.25 not related to length or diameter. If the physical properties of the fluid remain essentially unchanged throughout the line the C terms are equal. If the constants are canceled and Equation 10.34 rearranged:

$$L_e = L_B\left(\frac{d_A}{d_B}\right)^{5-n} \tag{10.35}$$

where n is the exponent in Reynolds number when the friction factor is expressed in the form of Equation 10.14 or 10.15.

If d_e is defined as the diameter of line of length L_A that will give the same friction loss as L_B feet of line d_B, Equation 10.34 may be rearranged to yield:

$$d_e = d_B\left(\frac{L_A}{L_B}\right)^{\frac{1}{(5-n)}} \tag{10.35a}$$

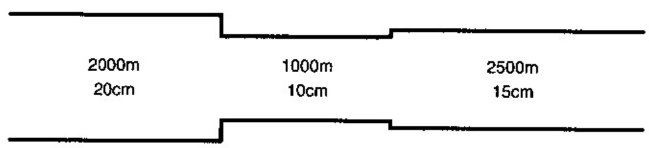
Equation 10.35 is particularly useful, for the system shown then may be regarded as consisting of L_e + L_A feet of line of diameter d_A. Once this equivalence is established, the problem may be solved in the conventional manner.

Equation 10.33 is summarized for large pipe and small pipe in Table 10.2.

TABLE 10.2
Equations for Lines in Series

	Lines in Series	
	Large Pipe	Small Pipe
Equivalent Diameter	$d_e = d_B\left(\frac{L_A}{L_B}\right)^{0.208}$	$d_e = d_B\left(\frac{L_A}{L_B}\right)^{0.207}$
Equivalent Length	$L_e = L_B\left(\frac{d_A}{d_B}\right)^{4.806}$	$L_e = L_B\left(\frac{d_A}{d_B}\right)^{4.828}$

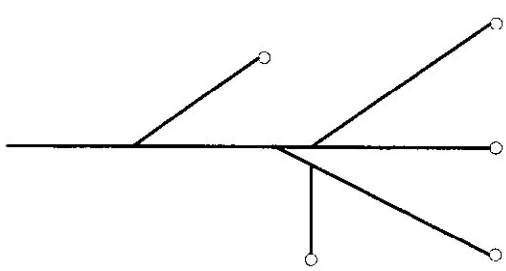
Example 10.7: Express the equivalent length of the following system in terms of 10 cm pipe, assuming that the nominal diameter is the actual inside diameter.



From Equation 10.35, $L_{10} = (2000)(10/20)^{4.828} + (2500)(10/15)^{4.828} = 423 \text{ m}$

Total length $= 423 + 1000 = 1423$ m

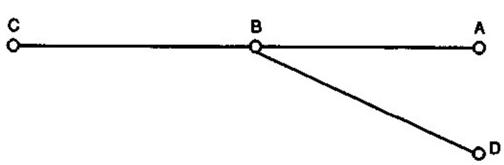
Therefore, the system shown may be considered as equivalent to one consisting of 1423 m of 10 cm pipe. Capacity or pressure-drop calculations then may be made on the latter system using the correlations previously presented.



Most gathering systems contain one or more branch lines. If the system is very complex, one has a large number of calculations to make. As a practical matter, these will be made on a computer.

This essentially is a trial-and-error approach, although some short cuts are possible. What one will do in principle is to use the fundamental equations discussed previously and combine these with the concepts of series and parallel flow.

The total system is made up of a series of branch lines coming together at a common point. In the sketch



$$q_{CB} = q_{AB} + q_{BD}$$

and P_B is common to all three lines. In the usual case, the flow rate from each well or production facility is estimated from reservoir analysis and the design of the down-hole equipment. The problem is to establish line sizes and line arrangement to optimize initial cost and pressure loss.

The proper approach is to plan the total gathering system at the time field development commences. Spacing and deliverability can be estimated within reason to set up an expected grid. Some modification will be needed usually, but the system will be more efficient than one which is expanded in a rather haphazard manner. Too often the resultant is an operating nightmare and is quite inefficient.

The procedure used for system design may be varied. One approach may be illustrated using the simple sketch shown previously.

1. Establish q_{AB}, q_{BD}, P_A and P_D. This fixes q_{CB}.
2. Establish a reasonable pressure for Point C.
3. Guess a pressure for Point B, between pressures established at A, C and D. One method is to prorate on distance.
4. Calculate d for each line, using methods outlined previously.
5. Repeat the calculation for various combinations of pressures, flow rates and line configurations until a grid is found that is satisfactory.

For preliminary studies we sometimes take a short cut by assuming a line velocity of 2–3 m/s [6–10 ft/sec] for each well flow rate to establish a diameter for its line. With these d's and map lengths a tentative grid may be drawn. ΔP is found for each leg and one can "work through" the grid to find the pressure distribution. In most cases, this simplified approach will check more complex methods rather well.

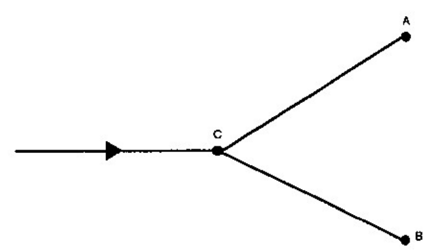
In most liquid grids the oil properties are relatively constant, as are wellhead pressures. If this is the case

$$\Delta P_f \propto (d, v, L)$$

One can set up a series of relative numbers for calculation of the system. If P_A and P_D are at the same pressure and at about the same elevation, lines A-B and B-D behave as loop lines and an equivalent diameter may be found.

We can use the principles to algebraically develop many working equations that describe the specific system being studied.

Flow Splitting



A common problem also encountered is where the flow from a single line is split into two or more parts, as in the figure at the right. In the rare case where the pressures and elevations at A and B are equal, this may be handled as a loop problem – in all others the calculation of flow is theoretically a trial-and-error process. However, this tedious approach usually may be eliminated by the following method.

1. Assume a pressure at Point C that is high.
2. Calculate ΔP_f for lines CA and CB.
3. Calculate q_L for both lines.
4. Knowing the total flow rate at C and the relative q_L's from above, calculate the actual flow rate for each line.
5. Using the actual flow rates, calculate ΔP_f for lines CA and CB using a correlation of f vs. Re.
6. Calculate pressure at C for both lines.
7. If the two pressures at C check within a few percent, the actual flow rates calculated in (4) above may be used. If not, a new value must be assumed for the pressure at C and the calculation repeated.

In most instances the above process is only necessary one time, particularly if the pressure at C is assumed high enough. Often it will work even if the assumed pressure and that calculated in Step 6 differ widely. As a rule-of-thumb, the assumed pressure should be about 15 times the maximum elevation change, expressed in pressure terms.

GAS FLOW

Principles

The fundamental thermodynamic equations used for liquid flow also may be used for gases. In order to conveniently handle Equation 10.6, several assumptions are often made:

$$\int VdP + \frac{g}{g_c}\Delta X + \frac{(\Delta v)^2}{2\,g_c} = -\frac{2\,f\,L\,v^2}{g_c\,d} - W$$

1. That no external work is done by or on the system, i.e., W = 0.
2. That the flow is isothermal.
3. That the changes in elevation on a long pipeline are negligible, i.e., X = 0.
4. The flow is steady state.

These three assumptions allow Equation 10.6 to be rewritten as:

$$\int VdP = \frac{-2\,f\,L\,v^2}{g_c\,d} \tag{10.36}$$

To evaluate the integral term the following assumption is made

$$v = \frac{n\,R\,T\,z_m}{P} \tag{10.37}$$

where z_m is not a function of pressure over the integration range.

Substitution of Equation 10.37 in Equation 10.36 and integration gives the "Basic Equation" shown below:

$$q_{sc} = K\left[\frac{T_{sc}}{P_{sc}}\right]\left[\frac{(P_1^2 - P_2^2)\,d^5}{f\,\gamma\,L\,T_m\,z_m}\right]^{0.5}$$

All of the above assumptions are usually satisfactory on the typical long pipeline. However, those equations derived on this basis usually contain an efficiency factor E to correct for these inaccuracies. (See Table 10.3) The assumption of isothermal flow has little effect on the final accuracy, which may be proven by assuming adiabatic conditions, the opposite extreme.

The use of a mean compressibility factor z_m is convenient. It may be determined at the average pressure P_m, which in turn is found from the equation:

$$P_m = 2/3\left[\frac{P_1^3 - P_2^3}{P_1^2 - P_2^2}\right] = 2/3\left[(P_1 + P_2) - \left(\frac{P_1\,P_2}{P_1 + P_2}\right)\right] \tag{10.38}$$

where P_1 and P_2 are the inlet and outlet absolute pressures, respectively. The mean temperature can be found from the equation:

$$T_m = \left[\frac{T_1 - T_2}{\ln\left(\frac{T_1 - T_g}{T_2 - T_g}\right)}\right] + T_g \tag{10.39}$$

Where: T_g = temperature of pipe surroundings

TABLE 10.3
Summary of Gas Flow Equations

	Metric	English
Basic, Eq. 10.40 $q_{sc} = K\left[\frac{T_{sc}}{P_{sc}}\right]^{1.000}\left[\frac{(P_1^2 - P_2^2)\,d^5}{f\gamma L T_m z_m}\right]^{0.5}$ (E)	$K = 5.62 \times 10^5$	$K = 38.774$
Weymouth, Eq. 10.41 $q_{sc} = K\left[\frac{T_{sc}}{P_{sc}}\right]^{1.000}\left[\frac{(P_1^2 - P_2^2)\,d^{5.333}}{\gamma L T_m z_m}\right]^{0.5}$ (E)	$K = 1.162 \times 10^7$ $f = \frac{0.00235}{d^{0.33}}$	$K = 433.49$ $f = \frac{0.008}{d^{0.33}}$
Panhandle A, Eq. 10.42 $q_{sc} = K\left[\frac{T_{sc}}{P_{sc}}\right]^{1.0788}\left[\frac{(P_1^2 - P_2^2)\,d^{4.854}}{\gamma^{0.8541} L T_m z_m}\right]^{0.5394}$ (E)	$K = 1.198 \times 10^7$ $f = \frac{0.0189}{(q\gamma/d)^{0.1461}}$	$K = 435.87$ $f = \frac{0.0192}{(q\gamma/d)^{0.1461}}$
Panhandle B, Eq. 10.43 $q_{sc} = K\left[\frac{T_{sc}}{P_{sc}}\right]^{1.02}\left[\frac{(P_1^2 - P_2^2)\,d^{4.961}}{\gamma^{0.961} L T_m z_m}\right]^{0.51}$ (E)	$K = 1.264 \times 10^7$ $f = \frac{0.0057}{(q\gamma/d)^{0.03922}}$	$K = 737$ $f = \frac{0.00359}{(q\gamma/d)^{0.03922}}$
AGA, Eq. 10.44	$K = 5.622 \times 10^5$	$K = 38.774$
	Partially Turbulent	Fully Turbulent
$q_{sc} = K\left[\frac{T_{sc}}{P_{sc}}\right]^{1.000}\left[\frac{(P_1^2 - P_2^2)\,d^5}{\gamma L T_m z_m}\right]^{0.5}$ (F_t)	$F_t = F_f\sqrt{1/f_{SPL}}$ $= F_f\left[4\log\left(\frac{Re}{\sqrt{1/f}}\right) - 0.6\right]$	$F_t = 4\log(3.74\,d/\varepsilon)$

Where:

		Metric	English
q_{sc}	= gas rate at T_{sc}, P_{sc}	m^3/d	scf/d
P	= absolute pressure	kPa	psia
P_{sc}	= pressure, standard conditions	kPa	psia
T_m	= mean absolute temperature of line	K	°R
T_{sc}	= temperature, standard conditions	K	°R
T_g	= ground temperature	K	°R
d	= inside diameter of pipe	m	in.
ε	= absolute roughness	m	in.
L	= pipe length	m	mile
μ	= viscosity	Pa·s	lb/ft-sec
γ	= gas relative density	–	–
z_m	= mean compressibility factor	–	–
f	= Fanning friction factor	–	–
E	= pipeline efficiency	–	–
Re	= Reynolds number	–	–
F_t	= transmission factor ($\sqrt{1/f}$)	–	–
F_f	= drag factor	–	–

Standard Equations

Table 10.3 summarizes several standard equations derived from the basic flow equation based on the above mean conditions. Equation 10.40 is the basic equation containing a friction factor "f." The Weymouth and Panhandle A and B equations are merely the basic Equation 10.40, using the friction factor correlations shown. The Weymouth equation assumes "f" depends only on pipe diameter; the Panhandle equations use different correlations for "f" as a function of gas flow rate, relative density and pipe diameter. The AGA equation uses the "f" values shown in Table 10.3.

The use of a *transmission factor* $(1/f)^{0.5}$ is common in gas transmission. It is a function of Re. Figure 10.5 shows the basic relationship for the flow conditions shown. The Weymouth equation is a horizontal line for fully turbulent flow. The smooth pipe law (no roughness) and Panhandle A parallel the AGA equation for partially turbulent flow. The AGA equation for fully turbulent flow is a series of horizontal lines depending on pipe roughness (ε). This will vary, but a value of 46 μm [1800 μin.] is suitable for typical steel pipe.

Although Figure 10.5 shows a sharp change between partially turbulent and turbulent flow, there really is a transition zone. If one wishes to use an AGA equation, use Equation 10.45 to determine which one.

$$\mathrm{Re} = 20.91 \left(\frac{d}{\varepsilon}\right)\left[\log\left(\frac{3.7\,d}{\varepsilon}\right)\right] \tag{10.45}$$

If the actual Re is less than that calculated from Equation 10.45, use the partially turbulent form; if greater, use the turbulent flow equation. For most large diameter natural gas transmission lines operating dry in a non-corrosive environment, ε = 15 – 33 μm [600 – 1300 μin.] with 19 μm [750 μin.] a reasonable average.

In Example 10.8 the range of answers obtained applies only to this example, but is indicative of the spread usually obtained. In the final analysis, the usefulness of a given equation will depend on experience with it in varying circumstances. The Weymouth equation is used quite often on gas gathering system design since it maximizes pipe diameter needs for a given flow and pressure drop.

Example 10.8: Determine flows using the Basic, Weymouth, Panhandle A and B, and AGA fully turbulent equations (Table 10.3) from the following information.

				Metric	**English**
Where:	P_1	=	absolute pressure	3448 kPa	500 psia
	P_2	=	absolute pressure	3103 kPa	450 psia
	d	=	inside diameter of pipe	0.3048 m	12 in.
	γ	=	gas relative density	0.7	0.7
	L	=	pipe length	16 100 m	10 miles
	T_m	=	mean absolute temperature of line	300 K	541 °R
	z_m	=	mean compressibility factor	0.96	0.96
	μ	=	viscosity	0.01 cp	0.01 cp
	E	=	pipeline efficiency	0.92	0.92
Solutions:	q_{sc}	=	gas rate at T_{sc}, P_{sc}	10^6 std m^3/d	MMscfd
			Basic (using Figure 10.1)	1.3	45
			Weymouth	1.1	39
			Panhandle A	1.3	44
			Panhandle B	1.4	50
			AGA (maximum)	1.3	47

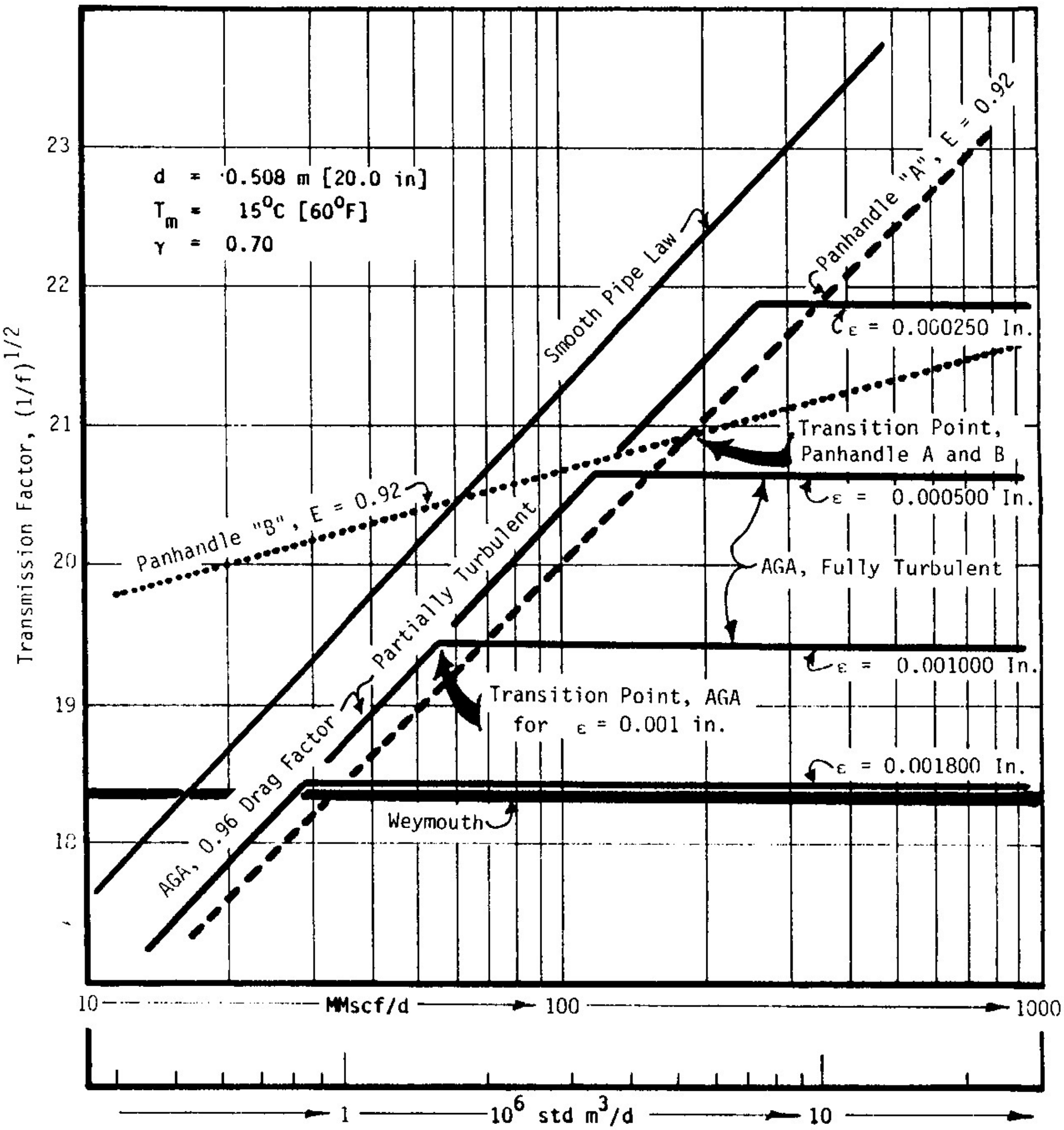


Figure 10.5 **Transmission Factor Comparison**

Maximum Velocity

The maximum allowable velocity in a gas line is governed by noise, pressure loss and surge considerations. Equation 10.46 is a guideline for establishing any maximum velocity limits. Design velocity always should be less than this maximum velocity.

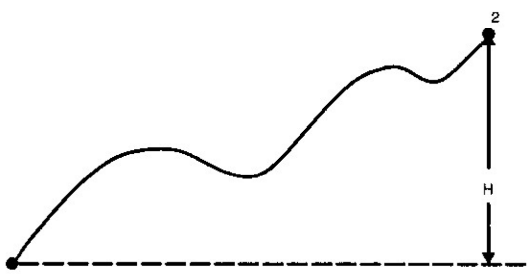
$$v = \frac{A}{(\rho)^{0.5}} \tag{10.46}$$

Where:		Metric	English
v =	velocity	m/s	ft/sec
ρ =	density	kg/m^3	lb/ft^3
A =	constant	146	120

At a pressure of about 7.0 MPa [1000 psia] Equation 10.46 will show a maximum velocity of about 17-18 m/s. At about 14.0 MPa the maximum velocity will be about two-thirds of this amount. See page 2211 for additional comments.

Static Pressure (Head) in Flow Lines

All equations like those shown in Table 10.3 are based on the assumption that the line is horizontal; i.e., there is no potential energy change affecting P_1 and P_2. In actual practice the line may be going uphill or downhill.



For single phase flow some correction must be made for the relative elevation between Points (1) and (2). Before using these equations some correction must be made for P_1 or P_2, to convert the ΔP to what it would have been if the line were horizontal.

Since the gas is compressible and temperature varies, many models may be used. The simplest equation uses an average temperature and compressibility, ignores any kinetic energy changes and assumes a constant friction factor. For these assumptions, from the basic energy equations, one may derive the equation

$$\frac{H\,\gamma}{A\,T_m\,z_m} = \ln\left(\frac{P_A}{P_B}\right) \tag{10.47}$$

Where:		Metric	English
H =	head	m	ft
γ =	gas relative density	–	–
T_m =	mean gas temperature	K	°R
z_m =	mean gas compressibility	–	–
P_A =	pressure at bottom of static column	MPa	psia
P_B =	pressure at top of static column	MPa	psia
A =	constant	29.28	53.34

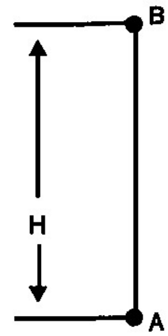
Equation 10.47 also can be written in the form

$$P_A = P_B\,e^s \tag{10.48}$$

Where: $s = \gamma/AT_mz_m$ and $e = 2.718$

There are two ways in which one may handle this correction for head, depending on which is more convenient for the calculation involved. Both require knowledge about the value of H, the relative elevation between (1) and (2).

What we are doing is converting the line from the actual profile shown previously to that shown at right. If line pressure P_1 is fixed, the pressure at (2) for a given circumstance depends on whether the line is horizontal, goes uphill or downhill.

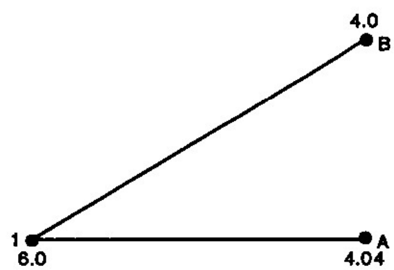


We use the subscripts A and B to show the pressure change due to head only. Do not confuse these with measured (or specified) pressures P_1 and P_2. In the uphill case, P_2 for use in the horizontal flow equations will be greater than the actual P_2. In Equation 10.48, actual P_2 will be inserted as P_B. The value of P_A solved for will be the new P_2 for use in the horizontal flow equations. For downhill flow the effective horizontal P_2 will be less than the actual P_2 and the process with Equation 10.48 is reversed.

One simple approach for a nonhorizontal line is to convert the actual P_2 (if known) to the equivalent P_2 for a horizontal line which can then be used in the type of equations shown in Table 10.3.

Example 10.9: A gas line runs uphill. The following data apply:

$H = 100$ m, $T_m = 300$ K, $z_m = 0.90$, $\gamma = 0.70$,
$P_2 = 4.0$ MPa, $P_1 = 6.0$ MPa
$s = (100)(0.70)/(29.28)(300)(0.9) = 0.00885$
$P_A = (4.0)e^{0.00885} = 4.04$ MPa (which is equivalent P_2 for horizontal flow)



If the line had been horizontal, the total pressure drop would have been less and P_2 would have been higher, or P_2 in the above example would have been 4.04 MPa. This value for P_2 would then be used in an equation of the type shown in Table 10.3. If the line goes downhill (the net H is minus), the equivalent horizontal P_2 would be less than the actual P_2. The basic rule – use as P_2 in the pipeline equations the value that should occur if the line is horizontal to relate q, d and pressure drop as shown by the equation chosen.

Also remember that length L should be established from the line profile and not merely the geographical distance between (1) and (2).

The alternative is to incorporate the pressure and length corrections directly into the flow equations. Probably, the method shown is easier and clearer.

For most actual gas pipelines the correction for head and length is very small. If in doubt, make it. The value of the calculation depends upon its use in the decision process.

SINGLE-PHASE VERTICAL GAS FLOW

The correction for potential energy (head) in the previous section is the simplest possible approach but is nevertheless suitable for situations where the magnitude of H is not large. It is thus suitable for the calculation of static pressure in shallow gas wells.

I cannot recommend the equations of Table 10.3 for flow in a wellbore. Each of these equations is based on the premise that the friction factor is primarily a function of diameter, a reasonable conclusion in relatively large diameter piping. Most tubing and casing possess small diameters compared to pipelines. Thus, other equations are often used for the application.

Static Bottom Hole Pressure

Equations 10.47 and 10.48 serve as a simple model for estimating bottom hole static pressure for a gas column. P_B would be the wellhead pressure and P_A the bottom hole pressure, under static conditions.

This is a trial-and-error calculation since z_a cannot be calculated until P_A is known. The first step is to assume a value of P_A to find z_a. Solve Equation 10.47 or 10.48 for P_A using this z_a. Use this new P_A to find a new z_a. After 3-4 trials the P_A assumed will converge on the P_A calculated.

What about the first assumption for P_A? A reasonable first guess is to assume a 600 kPa [90 psi] per 300 m [1000 ft] of depth.

An average temperature will have to be assumed for use with this simple approach. The use of any average introduces error. One way to compromise this is to subdivide total depth into several sections for calculation purposes. The P_A for the top section is the P_B for the next section, and so on down. In this approach, averages are not taken over as wide an interval of P and T and are thus more representative of the gradient.

There are two ways to find average quantities. One is to average P and T and then find the quantities at these average conditions. The second is to calculate the value of the desired quantities at each known P and T and then average each quantity. The effect of the averaging method on results is negligible in most instances.

Cullender and Smith[(10.8)], Poettmann[(10.9)] and others have proposed solutions for static bottom-hole pressure that are represented by Equation 10.49

$$\int_{P_1}^{P_2} \frac{zdP}{P} = \frac{\gamma H}{AT_m} = \int_{P_{r1}}^{P_{r2}} \frac{zdP_r}{P_r} = \int_{0.2}^{P_{r2}} \frac{zdP_r}{P_r} - \int_{0.2}^{P_{r1}} \frac{zdP_r}{P_r} \tag{10.49}$$

The left-hand terms of Equation 10.49 are equivalent to Equation 10.47. The value of the integrals on the right can be found from Table 10.4.

TABLE 10.4
Integrals of $(z/P_r)dP_r$ versus P_r

Pseudo-Reduced Pressure, P_r	Pseudo-Reduced Temperature, T_r										
	1.05	1.10	1.20	1.30	1.40	1.50	1.6	1.8	2.0	2.4	3.0
0.2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
0.4	0.62	0.62	0.63	0.63	0.63	0.64	0.64	0.64	0.64	0.64	0.64
0.6	0.96	0.97	1.00	1.02	1.04	1.05	1.05	1.05	1.05	1.05	1.05
0.8	1.18	1.21	1.26	1.30	1.32	1.34	1.35	1.36	1.36	1.36	1.36
1.0	1.33	1.38	1.46	1.50	1.53	1.55	1.57	1.59	1.60	1.61	1.61
1.2	1.43	1.50	1.60	1.65	1.68	1.71	1.74	1.76	1.78	1.80	1.81
1.4	1.49	1.59	1.71	1.77	1.81	1.85	1.88	1.91	1.94	1.95	1.98
1.6	1.53	1.65	1.80	1.88	1.92	1.96	1.99	2.04	2.06	2.08	2.11
1.8	1.56	1.69	1.87	1.96	2.01	2.06	2.09	2.14	2.17	2.20	2.23
2.0	1.59	1.73	1.92	2.03	2.09	2.14	2.18	2.23	2.27	2.30	2.33
2.5	1.66	1.82	2.04	2.18	2.26	2.32	2.36	2.44	2.48	2.51	2.56
3.0	1.73	1.89	2.14	2.29	2.40	2.47	2.51	2.60	2.64	2.69	2.74
3.5	1.80	1.96	2.23	2.39	2.52	2.58	2.64	2.74	2.79	2.85	2.90
4.0	1.86	2.03	2.31	2.47	2.62	2.69	2.75	2.86	2.92	2.98	3.04
4.5	1.93	2.10	2.38	2.56	2.70	2.78	2.85	2.97	3.03	3.10	3.16
5.0	2.00	2.17	2.45	2.63	2.78	2.86	2.94	3.07	3.13	3.21	3.27
5.5	2.07	2.24	2.52	2.70	2.86	2.94	3.02	3.15	3.22	3.30	3.37
6.0	2.13	2.30	2.59	2.77	2.93	3.02	3.10	3.24	3.30	3.39	3.46
7.0	2.26	2.43	2.71	2.90	3.06	3.15	3.24	3.39	3.46	3.56	3.62
8.0	2.39	2.56	2.84	3.03	3.18	3.29	3.37	3.52	3.60	3.70	3.77
9.0	2.51	2.68	2.96	3.15	3.31	3.41	3.49	3.64	3.73	3.82	3.90
10.0	2.63	2.80	3.08	3.26	3.42	3.52	3.61	3.76	3.84	3.94	4.02

Example 10.10: A 0.74 relative density natural gas is produced from a reservoir with a subsurface depth of 1768 m [5800 ft]. The shut-in surface pressure of this well is 12.4 MPa [1800 psia]. From well logs the reservoir temperature is 53°C [128°F]; the wellhead temperature is 20°C [68°F]. Estimate the bottom hole pressure.

$$P_c = 4.58 \text{ MPa } [665 \text{ psia}] \quad \text{and} \quad T_c = 214 \text{ k } [385°R]$$

English:

$$T_m = (68 + 128)/2 = 98°F$$

$$(H\gamma)/(AT_m) = (5800)(0.74)/(53.34)(98 + 460) = 0.144$$

$$P_r = 1800/665 = 2.71, \quad T_r = 558/385 = 1.45$$

From Table 10.4 (by interpolation) the value of the integral from 0.2 to P_{r1} = 2.35

From Equation 10.49,

$$\int_{0.2}^{P_{r2}} (z/P_r)\, dP_r - 2.35 = 0.144 \quad ,$$

$$\int_{0.2}^{P_{r2}} (z/P_r)\, dP_r = 2.49$$

By interpolation from Table 10.4, P_{r2} = 3.3

$$P_2 = (665)(3.3) = 2195 \text{ psia}$$

Relatively simple methods are available for calculating vertical gas flow. References 10.8-10.10 summarize these. One approach is to use Equation 10.40 (Table 10.3) with the value for "f" found from the equation following (from Cullender and Binckley).

$$f = A\left[\frac{\mu^{0.065}}{q^{0.065}\, d^{0.058}\, \gamma^{0.065}}\right] \tag{10.50}$$

Where:		Metric	English
μ =	gas viscosity	cp	cp
q =	gas flow rate	10^6 std m/d	MMscf/d
d =	pipe diameter	m	ft
γ =	gas specific gravity	–	–
A =	constant	0.005 715	0.007 73

An excellent review of these methods is provided in a series by Aziz.(10.10)

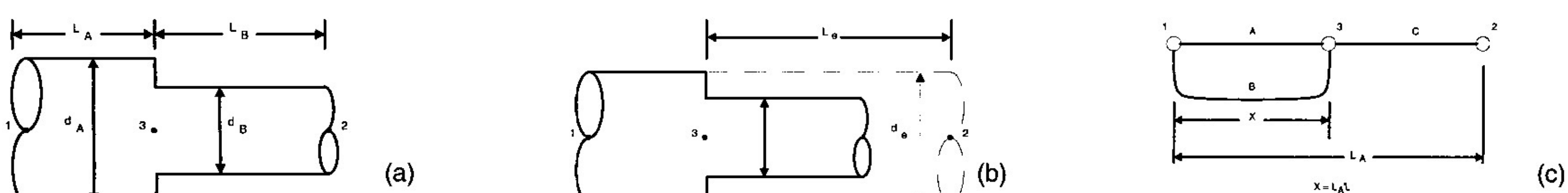
Complex Gas Flow Systems

The principles outlined for liquid flow also apply to steady-state gas flow. Table 10.5 summarizes the resulting equations. The sketches below the table outline the nomenclature used. The subscript (e) denotes a single line equivalent in length or diameter to another line in series or looped lines.

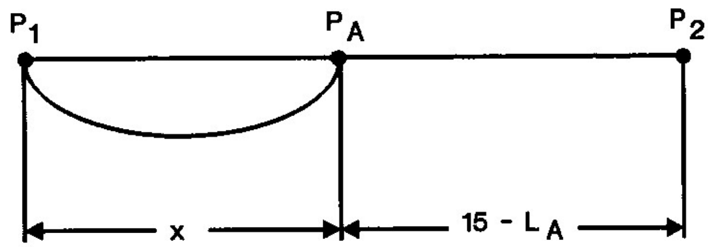
Examples 10.11-10.12 will serve to illustrate the application of the gas flow equations. For simplicity, the Weymouth equation, uncorrected for gas compressibility, will be used as an illustration of the principles involved.

TABLE 10.5
Equations for Complex Gas Flows

	Basic	Weymouth	Panhandle
	For Two Lines in Series		
Equivalent Diameter, d_e	$d_B\left(\frac{L_A}{L_B}\right)^{1/5}\left(\frac{f_A}{f_B}\right)^{1/5}$	$d_B\left(\frac{L_A}{L_B}\right)^{3/16}$	$d_B\left(\frac{L_A}{L_B}\right)^{0.2060}$
Equivalent Diameter, L_e	$L_B\left(\frac{d_A}{d_B}\right)^{5}\left(\frac{f_B}{f_A}\right)$	$L_B\left(\frac{d_A}{d_B}\right)^{16/3}$	$L_B\left(\frac{d_A}{d_B}\right)^{4.854}$
	For Two Lines in Parallel		
Equivalent Diameter or Length - Loops d_e or L_e	$\frac{d_e^{2.5}}{(f_e\,L_e)^{0.5}} = \frac{d_A^{2.5}}{(f_A\,L_A)^{0.5}} + \frac{d_B^{2.5}}{(f_B\,L_B)^{0.5}}$	$\frac{d_e^{8/3}}{L_e^{1/2}} = \frac{d_A^{8/3}}{L_A^{1/2}} + \frac{d_B^{8/3}}{L_B^{1/2}}$	$\frac{d_e^{2.618}}{L_e^{0.5394}} = \frac{d_A^{2.618}}{L_A^{0.5394}} + \frac{d_B^{2.618}}{L_B^{0.5394}}$
Loops - Diameters and Flows Vary X = Fraction Looped $d_R = \left(\frac{1/f_B}{1/f_A}\right)^{0.5}\left(\frac{d_B}{d_A}\right)^{5/2}$	$X = \frac{1-\left(\frac{q}{q_1}\right)^{0.5}}{1-\frac{1}{(1+d_R)^2}}$	$X = \frac{1-\left(\frac{q}{q_1}\right)^{2}}{1-\left[\frac{d_A^{8/3}}{d_A^{8/3}+d_B^{8/3}}\right]^{2}}$	$X = \frac{1-\left(\frac{q}{q_1}\right)^{1.86}}{1-\left[\frac{d_A^{2.618}}{d_A^{2.618}+d_B^{2.618}}\right]^{1.86}}$
Entire Line Looped		$\frac{q_1}{q} = 1+\left(\frac{d_B}{d_A}\right)^{8/3}$	$\frac{q_1}{q} = 1+\left(\frac{d_B}{d_A}\right)^{2.618}$
Diameters of Original and Parallel Lines are the Same X = Fraction Looped	$X = 4/3\left[1-\left(\frac{q}{q_1}\right)^{2}\right]$	$X = 4/3\left[1-\left(\frac{q}{q_1}\right)^{2}\right]$	$X = 4/3\left[1-\left(\frac{q}{q_1}\right)^{1.86}\right]$



Example 10.11: A portion of a large gas-gathering system consists of a 15.41 cm [6.067 in.] line 15 km [9.4 miles] that is handling 206 x 10^3 std m^3/d [7.6 MMscfd] with an average Rel. ρ of 0.64. The pressure at the upstream end of this section is 2.58 MPa gauge [375 psig] and the average delivery pressure is 2.07 MPa gauge [300 psig]. The average temperature is 23°C [73°F].



Due to new well completion, it is desired to increase the capacity of this line 20% by looping with additional 15 cm line. What length is required? Let L_A represent the length of the loop section.

Metric Solution

New flow rate $q_1 = 1.2\ (206 \times 10^3) = 274.2 \times 10^3$ std m^3

The loop may be represented by a single line d_e.

$$d_e = [(15.41)^{8/3} + (15.41)^{8/3}]^{3/8} = 19.98 \text{ cm} = 0.1998 \text{ m}$$

A Weymouth equation for each section may now be written.

$$247.2 \times 10^3 = 1.162 \times 10^7 \left(\frac{288}{100}\right)\left[\frac{(2680)^2 - P_A^2}{L_A\,(0.64)(296)}\right]^{1/2} (0.1998)^{8/3}$$

$$247.2 \times 10^3 = 1.162 \times 10^7 \left(\frac{288}{100}\right)\left[\frac{P_A^2 - (2170)^2}{(15 - L_A)(0.64)(296)}\right]^{1/2} (0.1541)^{8/3}$$

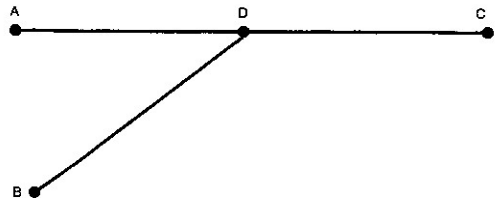
(Note: 100 kPa has been added to each gauge pressure.) Since there are two equations and two unknowns these equations may be solved algebraically. For this example $L_A = 6.0$ km.

Using the equations shown in Table 10.5 for looping requirements represents an easier solution method.

$$X = \frac{1 - \left(\frac{206}{247.2}\right)^2}{1 - \left[\frac{(15.41)^{8/3}}{(15.41)^{8/3} + (15.41)^{8/3}}\right]^2} = 0.407$$

$$L_A = 0.407\ (15 \text{ km}) = 6.1 \text{ km}$$

Example 10.12 The system at left is to be designed for the gathering of gas. Well "A" is a flowing gas well that will deliver 81.33 x 10^3 std m^3/d [3 MMscfd]. Lease "B" contains high gas-oil ratio oil wells. It is necessary to determine what compressor discharge pressure is necessary in order to deliver 47.44 x 10^3 std m^3/d [1.75 MMscfd] from point B. The pressure at the transmission line (point C) is 2.07 MPa gauge [300 psig]. Average flowing temperature will be 29°C [85°F]. The gravity of the flowing gas is 0.63 and that of the casing head gas is 0.71. The line diameters shown have been arbitrarily fixed because of pipe availability.



It is obvious that once the pressure at point D is found, P_B may be found; furthermore, the average specific gravity in line CD is estimated.

$$\frac{81.33(0.63) + 47.44(0.71)}{128.77} = 0.66$$

Line CD

$$128.77 \times 10^3 = 1.162 \times 10^7 \left(\frac{288}{100}\right)\left[\frac{P_D^2 - (2170)^2}{(16100)(0.66)(302)}\right]^{1/2} (0.1521)^{8/3}$$

$$P_D = 2409 \text{ kPa}$$

Line BD

$$47.44 \times 10^3 = 1.162 \times 10^7 \left(\frac{288}{100}\right)\left[\frac{P_B^2 - (2409)^2}{(4830)(0.71)(302)}\right]^{1/2} (0.1024)^{8/3}$$

$$P_B = 2490 \text{ kPa}$$

UNSTEADY STATE GAS FLOW

All of the discussion to this point is based on steady state flow. Unsteady state flow is, however, an important consideration. This cannot be analyzed thoroughly and conveniently by manual calculations but there are some simple approximations that may prove useful.

Transient Pipe Line Flow

Any time the rate of withdrawal of gas from a line differs from the input, unsteady state flow is occurring. During routine operations this often occurs because said line is used for storage to dampen out peak loads. During peak demand, output exceeds input and pressure is declining throughout the line. Between peak loads input exceeds output and the pressure builds up again.

To predict behavior of the line one must start with the basic energy equations and then develop a solvable model that will approximate the behavior of a moving pressure front with time. This is strictly a computer modeling game. Some models I see are not too applicable. Examine a model carefully to ascertain it if fits your specific system situation.

Blow Down and Purge

Occasions occur when it is necessary to blow down and purge a gas line. This is a special case of unsteady state flow.

One may estimate the blowdown time by some simple equations.[10.11,10.12]

For the *critical flow* case (when the ratio of the higher pressure to lower pressure is larger than two):

$$t = \left(\frac{BV}{C_d A_v}\right)\left(\frac{\gamma}{zT}\right)^{0.5} \ln\left(\frac{P_1}{P_2}\right) \tag{10.51}$$

Where:		Metric	English
t =	blowdown time	s	sec
B =	constant	0.09	5.3
V =	actual system volume	m^3	ft^3
C_d =	valve discharge coefficient	–	–
A_v =	valve area	m^2	$in.^2$
γ =	gas specific gravity	–	–
z =	average gas compressibility factor	–	–
T =	average gas temperature	K	°R
P_1 =	initial system pressure	kPa	psia
P_2 =	final system pressure	kPa	psia

The gas temperature and compressibility factor upstream of the blowdown valve change during the blowdown period. Thermodynamically the blowdown process falls somewhere between isentropic and isenthalpic. Estimating the average temperature and compressibility factor requires some knowledge of the thermodynamic path which is not known exactly. For most blowdown cases use of the initial temperature and compressibility factor is a reasonable assumption.

Equation 10.51 was developed for critical (sonic) flow. Once the system pressure falls below about 200-300 kPa [30-50 psia] the flow is probably subcritical. Fortunately the time required to completely depressure a system to atmospheric does not usually have to be calculated. When it does, Equation 10.51 has been found to give a reasonable estimate of total blowdown time if P_2 is set to about 80% of atmospheric pressure.

Pressure Surges on Closing a Valve

When a valve is closed on a line a pressure surge occurs. This is required thermodynamically. The kinetic energy of the fluid is converted to internal energy when flow stops. A wave (surge) travels back through the line countercurrent to the fluid still flowing forward, but slowing down. Each section of the line behaves in a different manner. Since flow does not stop upstream when stoppage occurs at some point downstream, the pressure rise decreases as the wave front travels upstream.

Some of the energy is absorbed by the expansibility of the fluid and the pipe. The more energy absorbed, the less the pressure rise. Thus, compressibility of the fluid is a primary factor. Since liquids are essentially incompressible, they represent the primary problem. By the same token, high pressure gas will exhibit more pressure rise than low pressure gas.

A thorough analysis is definitely a computer solution. Reference 10.13 provides an excellent summary of the problem and some important references.

For a liquid line an approximate equation for predicting pressure rise on a valve closing is

$$\Delta P = A\left(\frac{vL}{t}\right) \tag{10.52}$$

Where:		Metric	English
v =	liquid velocity	m/s	ft/sec
L =	length of pipe preceding valve	m	ft
t =	valve closing time	s	sec
ΔP =	pressure rise above steady state flowing pressure	kPa	psi
A =	constant	5.2	0.070

Pressure Testing

Gas pressure testing of a line is a form of unsteady flow in the sense that pressure must be held for sufficient time to assure that there is no leak. The time necessary for such testing may be estimated by the equation

$$t_m = \frac{Ad^2L}{P_1} \tag{10.53}$$

Where:		Metric	English
t_m =	time necessary, minimum	h	h
d =	internal pipe diameter	cm	in.
L =	length of pipe section	km	miles
P_1 =	initial test pressure	MPa	psig
A =	constant	0.002	3.0

When a line has been shut in for at least the above amount of time, it may be considered "tight" if the following pressure loss has not been exceeded.

$$\Delta P_{max} = \frac{tP_1}{Ad} \tag{10.54}$$

Where:		Metric	English
ΔP_{max} =	acceptable pressure loss	kPa	psi
t =	shut-in time	h	h
d =	pipe diameter	cm	in.
A =	constant	0.372	949
P_1 =	initial test pressure	MPa	psig

TEMPERATURE CHANGES IN PIPING

Most of the equations used commonly for pressure loss calculations require the use of a single value of temperature. This, of necessity, is some kind of an average temperature. In a numerical (computer) solution one divides the line into a series of constant temperature sections; an analytical (manual) solution may treat the entire line as one section. However one does it, the temperature used should be representative of what is anticipated. The value used has a direct impact on the physical property values used.

Prediction of the temperature distribution in the flowing system may be as important as the pressure profile. It affects many design considerations surrounding the line. Crude oil pour points, gas hydrates, vapor-liquid phase behavior and water content of gas are all temperature sensitive. A reliable temperature prediction is an early step in the calculation procedure.

The temperature at any point is predictable through use of a first law thermodynamic energy balance. If potential and kinetic energy changes are ignored – and work equals zero in the pipe section involved – this first law balance for a steady state system reduces to $\Delta H = Q$.

The enthalpy of a gas increases with decreasing pressure; it is almost independent of pressure for a liquid. So, the enthalpy at any point in the line depends both on pressure at that point and the heat energy lost or gained through the pipe wall.

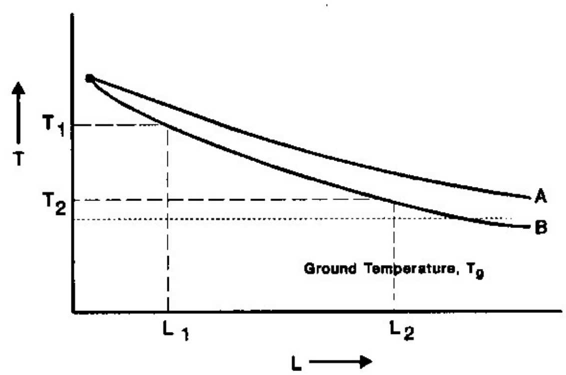
From heat transfer principles, $Q = UA\Delta t_m$. For a circular pipe this may be written as:

$$Q = (U)(\pi dL)(\Delta T_m)$$

Where:
- U = overall heat transfer coefficient
- π = 3.1416
- d = pipe diameter
- L = pipe section length = $L_2 - L_1$
- ΔT_m = log mean $\Delta T = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$
- Q = heat lost in line section of length L
- $\Delta T_1 = T_1 - T_g$
- $\Delta T_2 = T_2 - T_g$
- T_g = normal ground or water temperature at that point (far enough away from pipe to be undisturbed by it)

The general relationship is shown at right. If the fluid temperature is greater than T_g, heat transfer will be negative as shown; if it is less than T_g, the Q will be positive and the curves will approach T_g from below.

Line A is for the case where the effect of pressure on enthalpy is ignored, as per the discussion following. This is a simple heat transfer process and the fluid temperature cannot drop below T_g. Line B reflects the additional change in temperature due to the effect of pressure on enthalpy. In commercial practice this line normally would not drop over 2-3°C below T_g for ordinary pressure drop situations.



If no phase changes occur in the line section, the following equation applies.

$$\Delta h = \int_{T_1}^{T_2} mC_p dT + \int_{P_1}^{P_2} [V - T(\delta V/\delta T)_p]\, dP$$

This equation relates enthalpy to specific heat (C_p) and P, V and T. The second term is zero, or essentially so, for an ideal gas and a liquid. For a gas the equation shows that the enthalpy increases as the pressure declines.

If the pressure drop in the line segment (dP) is small compared to the absolute pressure, the second term of the equation is small compared to the first term. For a gas line at a pressure above 3.5 MPa [500 psi], ignoring this second term may not be important. There are thus three basic solutions for the Δh equation.

1. Use an enthalpy-PVT equation of state program.
2. Approximate the second term by assuming a series of Joule-Thomson expansions.
3. Ignore the second term completely.

Method (1) is a routine computer solution. Method (3) is an easy manual solution. The practical accuracy is as good as for (1) if a gas line is calculated in segments so that the pressure drop per segment does not exceed about 20% of the initial pressure. Method (2) is a carryover from pre-computer days as a manual method to calculate the effect of pressure on temperature (A, B). It could be programmed but is not as convenient as Method (1). Both Methods (1) and (2) require iterative solutions.

One can approximate Method (2) by translating the change of enthalpy with pressure into a temperature change per unit length. If this is done, Equation 10.55 results.

$$\ln\left[\frac{(T_1 + JL_1) - (T_g + J/a)}{(T_2 + JL_2) - (T_g + J/a)}\right] = a\,L \qquad (10.55)$$

Where:			Metric	English
	C_p =	heat capacity	kJ/kg·°C	Btu/lb-°F
	L_1 =	distance from initial point	m	ft
	L_2 =	distance from initial point	m	ft
	L =	$L_2 - L_1$	m	ft
	T_1 =	temperature at L_1	°C	°F
	T_2 =	temperature at L_2	°C	°F
	γ =	gas sp gr (air = 1.0)	–	–
	T_g =	ground or water temperature	°C	°F
	J =	Joule-Thomson coefficient	°C/m	°F/ft
	d =	outside pipe diameter	m	ft
	U =	overall heat transfer coefficient	kJ/h·m^2·°C	Btu/hr-ft^2-°F
	q =	gas flow rate, std volumes	10^3m^3/h	Mscf/h
	B =	constant	408	24.4
	a =	$dU/Bq\gamma C_p$		

Because of the Joule-Thomson effect, it is possible for the gas temperature to be less than that of the ground. This is only likely to occur with long lines possessing a large pressure drop or at regulator stations. For lean, pipeline-quality natural gases the cooling due to Joule-Thomson expansion is approximately 0.004 to 0.005 °C/kPa [0.05-0.06 °F/psi] at pipelines operating near 70 bar.

With Method (3) the quantity "J" is zero and Equation 10.56 results.

$$\ln\left(\frac{T_1 - T_g}{T_2 - T_g}\right) = \frac{d\,U\,L}{B\,q\,\gamma\,C_p} \qquad (10.56)$$

The above equations normally are solved for T_2. It is obtained by taking the antilog of the left-hand side. This means that any errors in the data are amplified.

Overall Heat Transfer Coefficient (U). This is the single number that represents all of the resistances in *series*. In a buried pipe, all of the following resistances to heat flow can occur:

1. Film coefficient between fluid and pipe wall
2. Inner pipe wall conductivity (tubing)
3. Annular space between inside and outside pipe
4. Pipe wall
5. Pipe coating
6. Insulation
7. Concrete layer or bond
8. Sand backfill
9. Native soil near pipe at temperature above normal due to heat gain from pipe

A line buried in soil normally would have resistances 1, 4, 5, 8 and 9; buried on the sea bottom 1, 4, 5 and 7 may apply. In a wellbore with fluid flowing inside the tubing 1, 2, 3, 4, 7 and 9 would be in series. The use of insulation can be justified only in very special situations.

Resistance 9 is necessary to correct for the temperature rise around the pipe above Tg, the temperature of the bulk of the ground at that depth. Temperature surveys show it is common for the ground temperature to be higher than Tg up to 5 m [15 ft] from the line. This is an effective resistance reducing heat flow.

As with a heat exchanger, the most reliable values of "U" are found by test. A study of uninsulated gathering and transmission lines shows values of U from 5-20 kJ/h·m2·K [0.25-1.0 Btu/hr-ft2-°F]. Most of the values measured are in the lower third of this range. Data on steam flood injection wells and geothermal production wells often will show higher "U" values because resistance (1) is lower.

Heat loss rates in bodies of water containing substantial currents or in tidal flats are higher because of convection losses due to water movement. A given line may cross several different heat loss environments.

The most practical approach is to recognize that a calculation based on a single value of "U" is basically invalid. Pick a likely range of values from test data to arrive at a range of answers. Then design for the worst case.

Temperature Tg. – This is not a constant quantity but depends on air temperature to some degree. At a burial depth below the "frost line" the soil temperature generally will vary from 2-16°C [35-60°F] seasonally. Maximum ground temperature will lag air temperature by a month or two. About the same temperature range will be encountered with burial in water over 30 m [100 ft] deep. In a body of water with no pronounced currents, the fluctuation will be less. Some lakes will have a bottom water variation of no more than 3°C [5°F] year round.

Temperature surveys should be a part of planning if the temperature of line contents is a design factor other than pressure loss calculations. For very preliminary considerations the minimum temperature seldom will be less than 0-1°C [32-35°F].

Because of changes in T_g, the value of "U" will vary seasonally also. In a temperate climate with discernible seasons, the "U" may be 50-60% higher in winter than in summer. In theory this should not occur, "U" being independent of T_g, but it has been noted in tests. One must recognize that the models only approximate the real world.

Above Ground Lines. – The above equations do not apply for above ground lines; no T_g is applicable. The maximum temperature of fluid in the line is the result of gain of daytime heat from the sun by radiation and the corresponding loss by convection to the air. The relative rate of these would determine the rate of heating with distance. The temperature depends on

Time of day	Wind velocity
Atmospheric conditions	Color and character of pipe surface
Air temperature (T_a)	Flow rate and properties of fluid in pipe

There are no absolutely rigorous performance prediction methods, but data are available from line tests.(10.35)

The maximum line temperature may be estimated by the equation

$$T = \left[\frac{R}{(\pi\, h_a)}\right] + T_a \qquad (10.57)$$

Where:
- T = max temp of fluid, °F
- R = solar radiation absorbed, Btu/ft^2/hr
- h_a = air film coefficient for convection, Btu/hr-ft2-°F
- π = 3.1416
- T_a = ambient air temperature, °F

Values of R and h_a may be estimated from the table below.

Pipe Surface	R	Wind Velocity, miles/h	h_a
Highly oxidized steel	300	0	2.0
Oxidized	230	2	2.6
Normal	180	5	3.5
Bright	110	10	4.5
Aluminum Paint	90	15	5.0
White Paint	75	20	5.2

Equation 10.57 shows the *maximum temperature* possible with the sun directly overhead, a clear sky and a long enough line to achieve thermal equilibrium between radiation energy gain and convection loss. In a typical case, the rise in temperature above ambient will not exceed 10-16°C [18-30°F].

The actual temperature probably will be lower than the maximum unless the exposed line is very long. Fluid properties, velocity, pipe length, season of the year (sun position) and geographical position all influence the actual radiation effect. Since not all of these are constant with time, neither is performance.

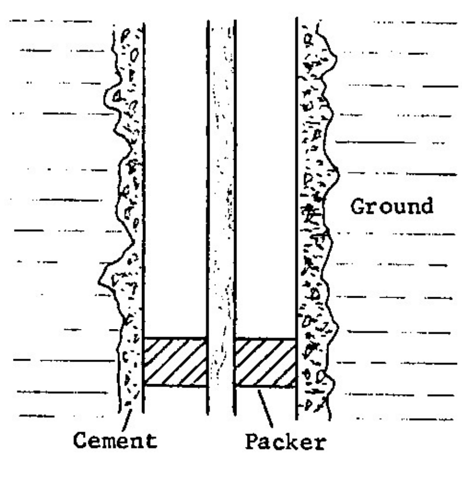
At night, even in a tropical climate, radiation losses to a clear, dark sky may be enormous. So ... rapid temperature changes can occur. The best one can hope to do is estimate the magnitude of such changes for decision purposes.

Insulation. – The purpose of insulation is to reduce the rate of gain or loss of energy, to or from a pipe, in a cost-effective manner. When one has a group of resistances in series, the one with the lowest thermal conductivity controls total heat loss.

The amount and type of insulation is governed by mechanical limitations and economics. Some very effective insulations are not suitable for buried line applications. Some of the polyurethane formulations have proven satisfactory in lines containing heated heavy oils and liquefied natural gas. Overall values of "U" of the order of 0.01 Btu/hr-ft^2-°F are achievable.

Wellbore Temperatures

Prediction of wellbore temperatures parallels that for pipelines. In this case, though, ground temperatures around the pipe may vary substantially with depth. A series of models has been proposed for injection, production and geothermal wells.



The prediction of surface flowing temperatures is critical to surface facility design. None of the models is completely adequate because of the inability to define all variables quantitatively. The work of Ramey is most often used as the basic equation. Actual wellbore then is used to adjust the constants. Effective values of "U" may vary from 0.25-2.0 Btu/hr-ft^2-°F. They tend to be higher than for pipelines because of higher fluid velocities and film coefficients.

The models used for calculation of well-bore temperatures are designed for computer usage. The accuracy of the output is contingent on the amount of hard data available for similar wells in the same area. In the absence of such data, the accuracy of such prediction may be no better than 15-25%. One of the common mistakes in surface facilities design is not allowing for a range of wellhead temperatures and designing for the worst condition.

Temperature depends on time. It takes finite time for the temperature profile to stabilize. This is true particularly in wellbore and gathering lines as wells are continually shut in and opened or flow rates change. With steam injection, geothermal wells and wells in permafrost or ice islands, insulation may be required to stabilize the temperature-time relationship.

References 10.14-10.24 and 10.38-10.46 address temperature performance.

GAS-LIQUID (TWO-PHASE) FLOW

The simultaneous flow of liquid and gas in a line is most important in modern operations. For many installations the use of two-phase lines is the most economical solution. A two-phase line reduces metal needed and reduces capital cost 20-25% from that of two single-phase lines. It is also practical from a mechanical standpoint.

All models are based on the same thermodynamic principles and fluid flow principles discussed previously herein. As with single-phase flow, a prediction of the "lost work" (friction drop) term is essentially empirical. Thus, any one method must be limited by the pipe sizes, fluid characteristics, flow conditions and geometry, and gas-liquid ratios used to determine a given correlation.

There is no single best correlation for universal use. When using a given model one should examine the kinds of systems on which it is based. This is why data on existing systems is vitally important. Is the model and its supporting data physically compatible with your proposed system? Like all semi-empirical equations, extrapolation beyond the range of data used to develop said equation is extremely hazardous.

For example, some correlations have been based on data from small diameter, horizontal pipes. Their application to large diameter pipelines with uphill and downhill profiles is limited. Their primary application would be in process piping. Table 10.6 from Reference 10.32 indicate the range of data taken from some of the more popular correlations.

TABLE 10.6
Experimental Information for the Two-Phase Pressure Drop Correlations

Correlation	Date	Basis	Pipe Size(s)	Fluids
Vertical Flow				
Duns & Ross	1961	Laboratory and field data	wide range	oil, gas, water
Angel-Welchon-Ross	1964	Field data	large diameter tubing and annuli	gas, water
Hagedorn & Brown	1965	Laboratory and field data	1 in. - 4 in.	oil, gas, water
Orkiszewski	1967	Review and modifica-tions of other methods	wide range	oil, gas, water
Aziz & Govier	1972	Laboratory and field data	wide range	oil, gas, water
Beggs & Brill	1973	Laboratory data	1 in., 1.5 in.	gas, water
Gray	1974	Field data	< 3.5 in.	gas condensates
Horizontal Flow				
Lockhart-Martinelli	1949	Laboratory data	0.0586 in. - 1.1017 in.	
Eaton	1966	Laboratory and field data	2 in., 4 in.	
Dukler	1969	Data and similarity analyses	wide range	oil, gas, water
Inclined Flow				
Mukherjee-Brill	1983	Laboratory data	1.5 in.	kerosene, lube oil, gas

In the real world there are three systems of basic concern – a pipeline with uphill and downhill components, vertical pipes in wellbores and risers, and combinations of the two. Correlations from these kinds of systems are of primary importance.

The discussion herein is merely an overview of the principles involved, to provide basic understanding of the principles. Chapter 8 of Reference 10.1 provides a more detailed discussion and references.

A large amount of work has been done in transparent piping so that behavior could be photographed. A number of flow regimes have been observed which help our understanding of the problem.

Figure 10.6 shows various forms of flow noted. Different words might be used to describe each type, and a different number of types could be used, but the seven regimes shown cover the range observed.

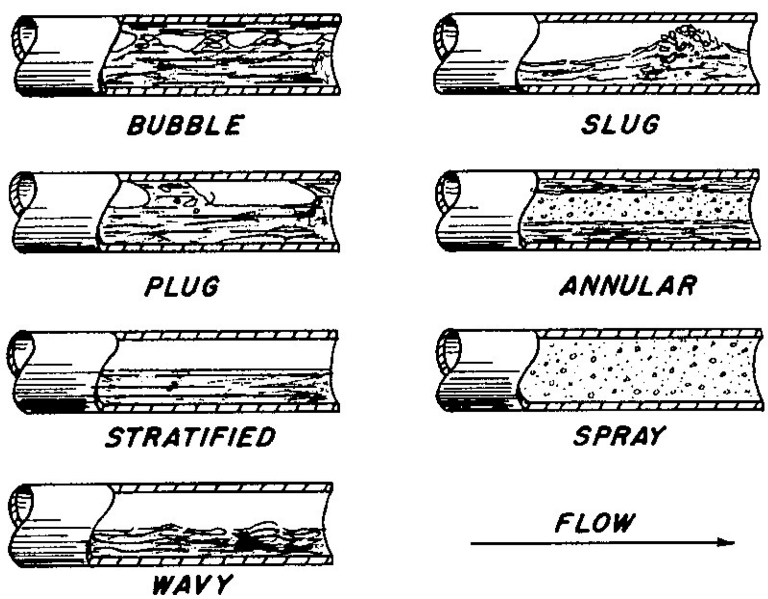


Figure 10.6 Various Two-Phase Horizontal Flow Regimes

The basic behavior of these two-phase systems depends on whether or not both the liquid and gas phases are present as a *continuous* phase; i.e., a continuous channel for flow exists for the phase. In bubble flow one has a series of bubbles of gas in the liquid. Gas is not a continuous phase. At the other end of the scale one sees spray or mist flow – small particles of liquid suspended in the gas. Liquid is now the discontinuous phase, gas the continuous phase.

In both cases the discontinuous phase may be considered to be merely altering the flow behavior of the continuous phase. One very simple approach is merely to modify the properties and/or friction factor of the continuous phase and then treat the system as single-phase flow. In bubble flow a liquid, single-phase calculation would be made. In spray or mist flow a gas correlation is used after this same correction.

This often is done by default when correlating data on existing systems. Most crude oil lines contain some gas and most gas lines contain at least entrained liquid. This is why we need separators and scrubbers.

In some of the regimes shown in Figure 10.6 both phases are continuous. This is a more complicated system because of the interactions between discrete phases. The gas moves more rapidly than the liquid. There is a shear force at the phase boundary. The character of flow depends on such variables as phase densities and viscosities, the velocity of each phase, gas-liquid ratio and properties like surface tension.

In a given length of line several flow regimes might occur because of varying forces and gas-liquid ratios. The latter changes as liquid condenses from gas or gas is formed from liquid, as dictated by phase behavior. Consideration of variables like these is a necessary part of developing a correlation.

Figure 10.7 shows four basic regimes that occur in vertical flow. A fifth regime, not shown, may be called froth flow. As noted later, this occurs at high liquid throughputs when the gas bubbles in the liquid are dispersed in an ever-increasing number of small bubbles as gas throughput increases.

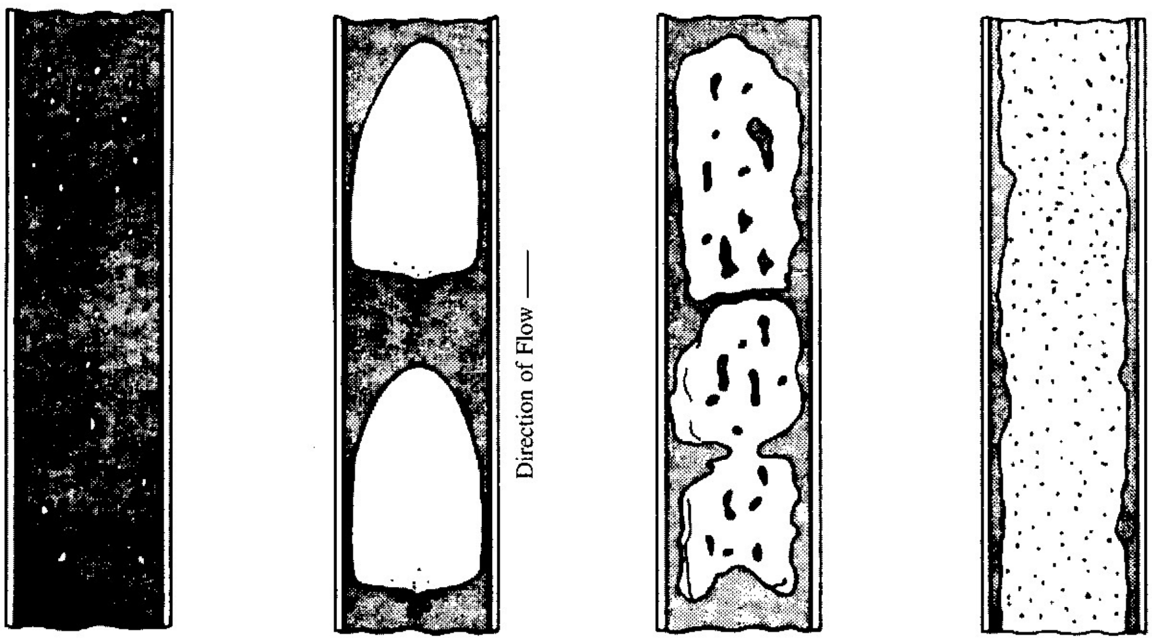


Figure 10.7 Vertical Two-Phase Flow Regimes

In vertical flow the force of gravity opposes the dynamic forces (instead of being at right angles to it). This results in *slippage*. Consequently, vertical flow exhibits some different characteristics than horizontal flow and may be more complicated.

Consider friction loss for a moment. The friction factor is a measure of all irreversible energy changes occurring in the pipe. In single-phase flow the friction factor primarily is a representation of the loss between fluid and the pipe wall and is correlated as a function of fluid properties, fluid velocity, pipe diameter and pipe roughness.

In two-phase flow there are two phases, with different properties, interacting with the pipe wall as well as with each other. A friction factor, therefore, depends on more, and complex, variables. The difference between correlations depends to a large extent on how these variables are treated in order to fit the available data.

Because of the complex forces involved in determination of an effective friction factor, several dimensionless groups in addition to Reynolds number have proven useful.

Froude Number This dimensionless group represents the ratio of inertial force to gravity force. It can be used to characterize flow whenever gravity force influences fluid flow.

$$Fr = \frac{v}{(gd)^{0.5}} \quad \text{or} \quad Fr = \frac{v^2}{gd}$$

The form at left is the basic equation. The right-hand expression is oftentimes more convenient to use and is often referred to as a Froude No. even though it is actually $(Fr)^2$.

Weber Number The dimensionless group is defined as the ratio of inertial force to surface force. It is equal to twice the ratio of kinetic energy to surface energy of a given volume of fluid. It may be written as

$$We = \frac{v}{(\sigma/\rho\ d)^{0.5}} \quad \text{or} \quad We = \frac{v^2\ \rho\ d}{\sigma}$$

Once again the right-hand form often is used because of its convenience. This dimensionless group is a logical correlation parameter where one phase is discontinuous because interfacial tension (σ) affects drop size.

Euler Number This dimensionless group is the ratio of applied pressure force to inertial force. It is a suitable correlating function in a flow system where pressure force is a controlling force, particularly if one of the fluids is compressible. It is written as

$$Eu = \frac{v}{(2\ \Delta P/\rho)^{0.5}} \quad \text{or} \quad Eu = \frac{\rho\ v^2}{2\ \Delta P}$$

In the above three dimensionless groups the nomenclature is as follows:

v = effective fluid velocity	ρ = fluid density
g = acceleration of gravity	σ = interfacial (surface) tension
d = diameter of pipe	p = pressure

Although these groups describe the mechanisms affecting two-phase flow, their application is not routine. Since the phases move at different velocities, what value of "v" does one use in a given group? Also, in a given situation what is the relative magnitude of the forces represented? Does one group control and therefore become suitable for correlation, or must several be used?

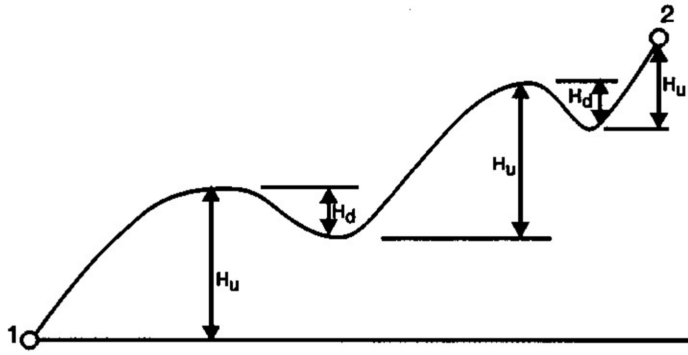
Different investigators have answered these questions in a different manner. In a two-phase system some composite or effective values are needed since each phase possesses different values for each term. Therefore, it must be remembered that all correlations are based on the same fundamental principles; they differ only in how questions like those above are answered in fitting available data.

In spite of the correlation accuracy that might be reported for a given correlation, we recommend assuming an accuracy of 15-20% in design of a new system. In planning, this leads to decisions which are economically sound and provide necessary operating flexibility. How good are the flow rate, pressure and property data to be used with a correlation in design? They are probably no more accurate than the correlation. So ... as a practical matter a proper correlation is as good as the data being inserted into it.

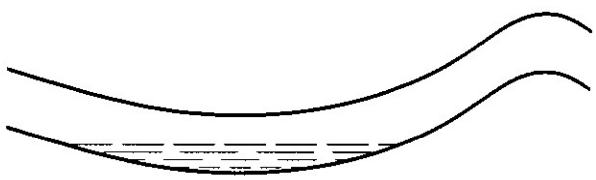
HORIZONTAL TWO-PHASE FLOW

No line buried in the ground or under water is truly horizontal. The word "horizontal" simply means that the length is a much larger number than any elevation change. In such lines two-phase flow occurs uphill and downhill as well as horizontally. Some aspects of vertical flow are occurring simultaneously with said horizontal flow.

The following figure is a view of a typical line where the magnitude of the uphill and downhill portions has been obtained by survey. Unlike single-phase flow, each change in elevation between (1) and (2) affects head considerations.



If the energy of the liquid entering a low spot is insufficient to carry it over the next "hill," liquid will collect in the low spot as shown next.



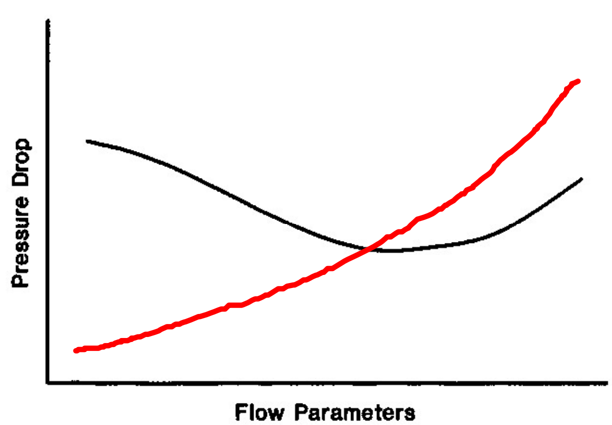
As this liquid builds up, the opening for gas decreases. At some point there is sufficient pressure force to start the liquid moving. A portion is gas lifted over the hill and the remainder slips back. The liquid flowing on contributes to liquid buildup in the next low spot. Then, in time, it surges. This process is repeated successively as liquid moves down the line.

This represents an inefficient use of pressure energy. It takes energy to overcome liquid inertia and produce the *momentum* needed. Furthermore, the lifting process is relatively inefficient. Both contribute to excess pressure loss. For this reason, it is desirable to maintain high enough flow rates to keep the liquid in motion.

In addition to excess pressure loss, liquid collection causes excess liquid holdup and results in larger liquid surges at the outlet of the line.

Both excess pressure drop and surges are noted in lines operating well below design capacity. As rate increases, both decrease. So ... the proper motto is, do not oversize your lines. Think small!

This is illustrated by the general figure below. Pressure loss goes through a minimum and then increases as flow rate increases. The reasons include the liquid segregation discussed above.



For the typical offshore installation where initial flow rates are low, "slug catchers" and other equipment must be designed for this worst case. The line must, of course, be sized for the maximum anticipated rate, but some consideration must be given for lower rates so that the system is satisfactory under all conditions expected.

Basic Correlations

All correlations are some form of a *momentum balance* and *continuity equation* containing terms for the head, friction and acceleration effects. These vary with the flow regime. Thus, it is common to write a different equation for each specific regime.

$$\underbrace{\left(\frac{dP}{dL}\right)}_{\text{total}} = \underbrace{\frac{f_{tp}\,\rho_{tp}\,v_{tp}^2}{2\,g_c\,d}}_{\text{friction}} + \underbrace{\frac{g\,\rho_{tp}\sin\Phi}{g_c}}_{\text{elevation}} + \underbrace{\left(\frac{\rho_{tp}\,v_{tp}}{g_c}\right)\left(\frac{d\,v_{tp}}{dL}\right)}_{\text{acceleration}} \tag{10.58}$$

Equation 10.58 includes several assumptions: 1) steady state, 2) no nuclear reactions, 3) no pumps or compressors in the system, and 4) adiabatic. With these limitations, this provides the basis of most fluid flow calculations.

The complexity of the solutions has increased with the development of computer models. The various correlations now available can be placed in three general classes.

Class 1

a. Liquid hold-up is not considered in density.
b. Liquid hold-up and wall friction losses are incorporated into a friction factor.
c. No distinction is made between flow regimes.

Class 2

a. Liquid hold-up is considered in density.
b. Liquid hold-up may be correlated separately.
c. Friction factors are based on composite properties.
d. No distinction is made between flow regimes.

Class 3

a. Density is adjusted for liquid hold-up.
b. Liquid hold-up is estimated using some concept of slip velocity (difference between gas and liquid flow rates).
c. The continuous fluid phase(s) determine(s) wall friction losses.
d. Flow regimes are considered.

Most of the correlations in the past twenty years are in Class 3 and are only applicable as computer solutions.

As noted in Reference 10.1, there is a group of correlations commonly used for horizontal flow. The usual, intelligent approach is to make the calculation by several methods to establish a likely performance range. No one method is superior to all others for general usage.

One of these methods (Flanigan's) is suitable for manual calculations and illustrates the relative role of some key variables.

Modified Flanigan Correlation[10.25]

This work was done on field systems and has proven useful even though it is relatively simple. The relationship between gas flow rate, diameter and pressure drop is represented by the Panhandle A equation (Table 10.1). Two corrections are made for two-phase flow.

1. The value of outlet pressure (P_2) is adjusted for the pressure loss due to uphill and downhill flow of two phases, including the effect of *holdups*.
2. The efficiency term (E) is correlated to reflect measured system performance based on gas velocity and liquid-gas ratio.

As thus modified the Panhandle A equation becomes

$$q_{sc} = K\,(T_{sc}/P_{sc})^{1.0788} \left[\frac{P_1^2 - (P_2 + \Delta P_2)^2}{T_m\, L\, z_m}\right]^{0.5394} (1/\gamma)^{0.4606}\,(d)^{2.6182}\,E_{tp} \qquad (10.59)$$

Where:			Metric	English
	q_{sc} =	gas rate at T_{sc}, P_{sc}	m^3/d	scf/d
	P =	absolute pressure	kPa	psia
	P_{sc} =	pressure, standard conditions	kPa	psia
	T_m =	mean absolute temperature of line	K	°R
	T_{sc} =	temperature, standard conditions	K	°R
	d =	inside diameter of pipe	m	in.
	L =	pipe length	m	mile
	γ =	gas relative density	–	–
	z_m =	mean compressibility factor	–	–
	E_{tp} =	two-phase efficiency		
	K =		1.198×10^7	435.87

The term ($P_2 + \Delta P_2$) is the outlet pressure of a truly horizontal line equivalent to that of the actual P_2.

The pressure correction for inclined flow is calculated from Equation 10.60

$$\Delta P_2 = A\,(\rho_L\, E_h\, \Sigma H_u - \rho_g\, \Sigma H_d) \qquad (10.60)$$

Where:			Metric	English
	ΔP_2 =	additive correction to Panhandle A P_2	kPa	psi
	A =	constant	0.009 81	0.0069
	ρ_L =	liquid density	kg/m^3	lb/ft^3
	ρ_g =	gas density	kg/m^3	lb/ft^3
	H =	head	m	ft
	E_h =	empirical head factor (Figure 10.8)	–	–
	H_u =	uphill heads	m	ft
	H_d =	downhill heads	m	ft

The head terms are found from the line profile as shown previously.

Figure 10.8 is a correlation to find the liquid head factor in Equation 10.60. *Superficial gas velocity* is the actual volumetric rate divided by cross-sectional area; it is assumed that gas occupies all of the pipe. At low velocity the head loss is high. Liquid collects in the low spots and is lifted periodically. This is inefficient. As velocity increases above 2 m/s [6.6 ft/sec] this head loss decreases rapidly. Above about 8 m/s [26.2 ft/sec] there is little effect of velocity on head pressure loss.

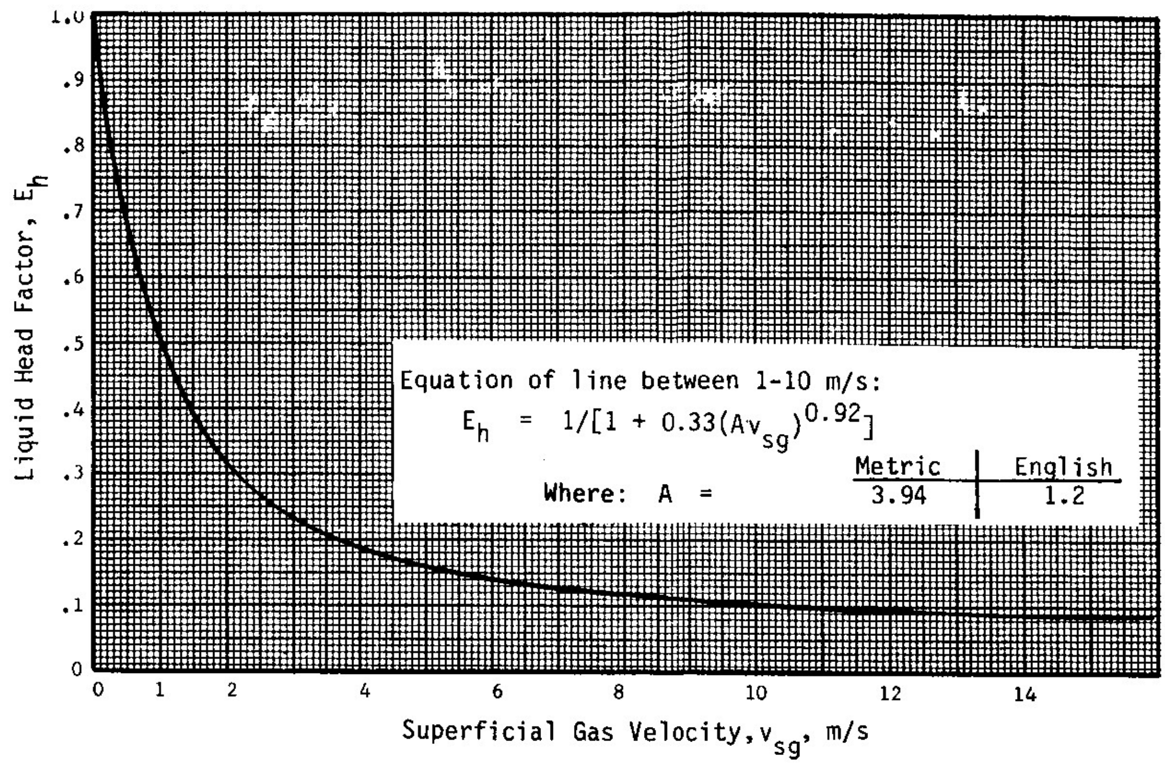


Figure 10.8 Flanigan's Liquid Head Factor Results.(10.25)

The velocity for use in Figure 10.8 may be calculated from Equation 10.61.

$$v_{sg} = (A)\,\frac{q_{sc}\, z_m\, T_m}{d^2\, P_m} \tag{10.61}$$

Where:		Metric	English
q	= gas flow rate (sc)	10^6 m^3/d	10^6 ft^3/d
d	= internal pipe diameter	m	in.
P_m	= mean pressure in section	kPa	psi
T_m	= mean temperature in section	K	°R
A	= constant for conversion	5.12	60.0
v_{sg}	= gas velocity	m/s	ft/sec
z_m	= compressibility at T_m and P_m		

The two-phase "efficiency" (E_{tp}) is found from Figure 10.9. The v_{sg} on the abscissa is the same one used for Figure 10.8. The "R" is the liquid-gas ratio in the pipeline segment.

At the upper right end of this curve the line contains a relatively small amount of liquid. The efficiency approaches that of a gas line. To the left, the amount of liquid increases and flow behavior approaches bubble and plug flow.

In many lines, the liquid/gas ratio, R, varies throughout the length. As the gas cools, R increases. This cooling follows normal temperature reduction patterns. On a long line, the value of R quickly becomes stable. If one only knows R at the inlet and outlet of such a line, using R at the outlet is the best of the two values since it probably exists for most of the line length.

Because of the variation of R, P and T with length, a long line is sometimes divided into segments for calculation purposes. An average or mean P and T is found for each section. R is found from an equilibrium

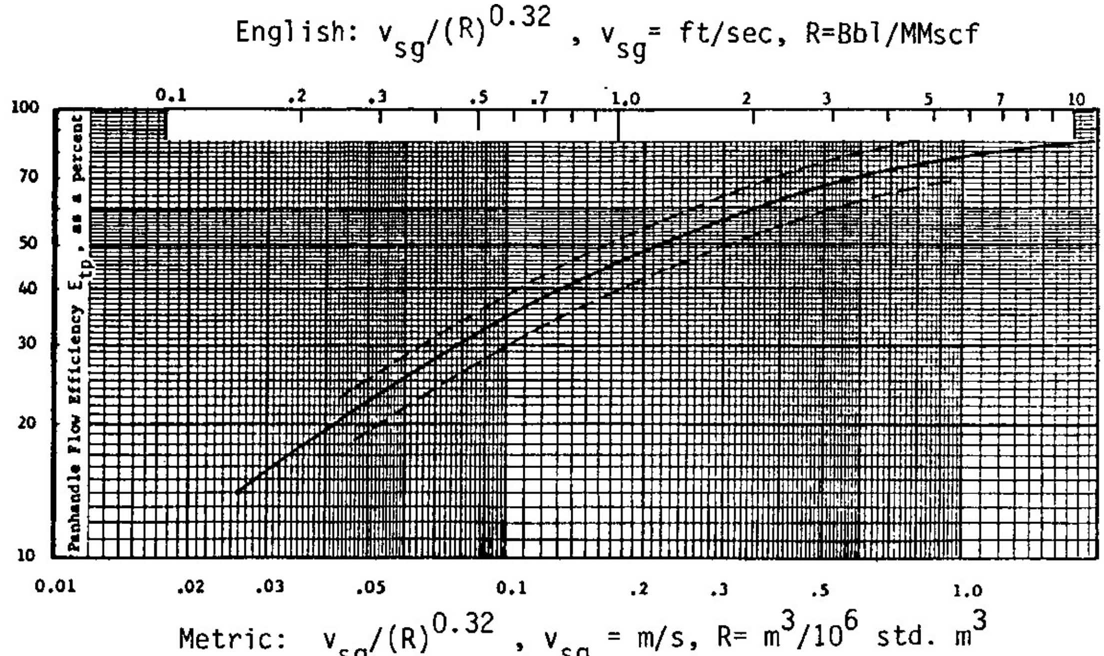


Figure 10.9 Flanigan's Holdup Correlation

calculation at these values. The ΔP of the segment is found after determining its regime and appropriate equation. If numerous segments in series are used, the calculation quickly becomes too time-consuming to be done manually.

Three different basic calculations are made using this method.

1. *Flow rate and diameter known, solve for pressure drop.* – This is a straight-forward solution. Solve Equation 10.59 for $(P_2 + \Delta P_2)$. P_2 is the desired outlet pressure – or if P_2 is fixed, solve equation for inlet pressure P_1.
2. *Flow rate and pressure drop known, solve for diameter.* – Assume a diameter to start a trial-and-error process. For this assumed value calculate ΔP_2 and E_{tp} and solve Equation 10.59 for "d." When value calculated checks the assumed value (within even pipe sizes), the correct diameter is known.

   What is a good first value of "d" to assume? Use Equation 10.61 for an assumed superficial velocity of 8 m/s [26 ft/sec] to solve for "d." This same assumption is used also in Figures 10.8 and 10.9 for the first trial.
3. *Pressure drop and diameter known, solve for flow rate.* – This also is trial-and-error. Assume a flow rate. This fixes a velocity for the first trial. When the assumed value agrees with the value calculated in Equation 10.59 with about 10%, you have a solution.

   One can guess a velocity as in (2) above and use Equation 10.61 to find a corresponding "q" as a first guess.

   This is a simple trial-and-error process because many of the equation values stay constant for all trials.

This modified Flanigan approach has proven useful for lines where the efficiency calculated is above 50%. Below this the combination of gas velocity and liquid loading is such that a more complex system tends to exist. This correlation certainly is valid only when one has the expectation that gas is present as a continuous system. Toward the lower end of the curve this is less likely.

Figure 10.10 shows results reported by Gould and Ramsey(10.26) for a 15-inch line containing a 0.7 relative density gas and a 0.83 relative density oil at various liquid-gas loadings. The pressure used was 6.9 MPa [1000 psig] and the temperature was 4°C [40°F]. The results should not be regarded as typical. They are shown primarily to illustrate the spread of results one can obtain by various methods. Reference 10.1 shows other test results.

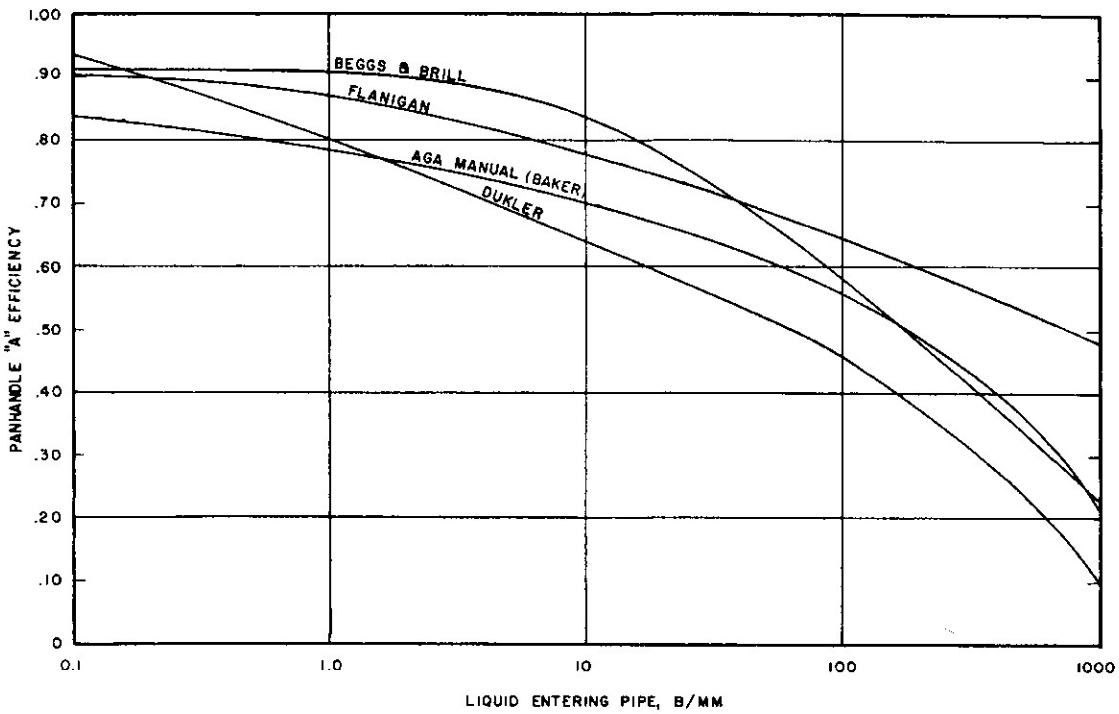


Figure 10.10 A Comparison of Calculation Methods for One Case

VERTICAL TWO-PHASE FLOW

The principles are the same as for horizontal or inclined flow except that gravitational forces are acting in an opposite direction to flow. The segregation patterns are different. Figure 10.11 illustrates the effect of the variables on vertical flow behavior.

The values of RN and N are two of the four dimensionless groups used to characterize system behavior.

$$RN = \text{gas velocity number} = v_{sg} (\rho_L/g\sigma)^{0.25}$$

$$N = \text{liquid velocity number} = v_{sL} (\rho_L/g\sigma)^{0.25}$$

Where:
- v_{sg} = superficial gas velocity
- v_{sL} = superficial liquid velocity
- ρ_L = liquid density
- σ = surface tension
- g = gravity acceleration

Since RN and N are dimensionless numbers, any consistent set of units may be used so long as the result is dimensionless. The gas velocities shown are found by dividing the volumetric flow rate of each phase by the total pipe area.

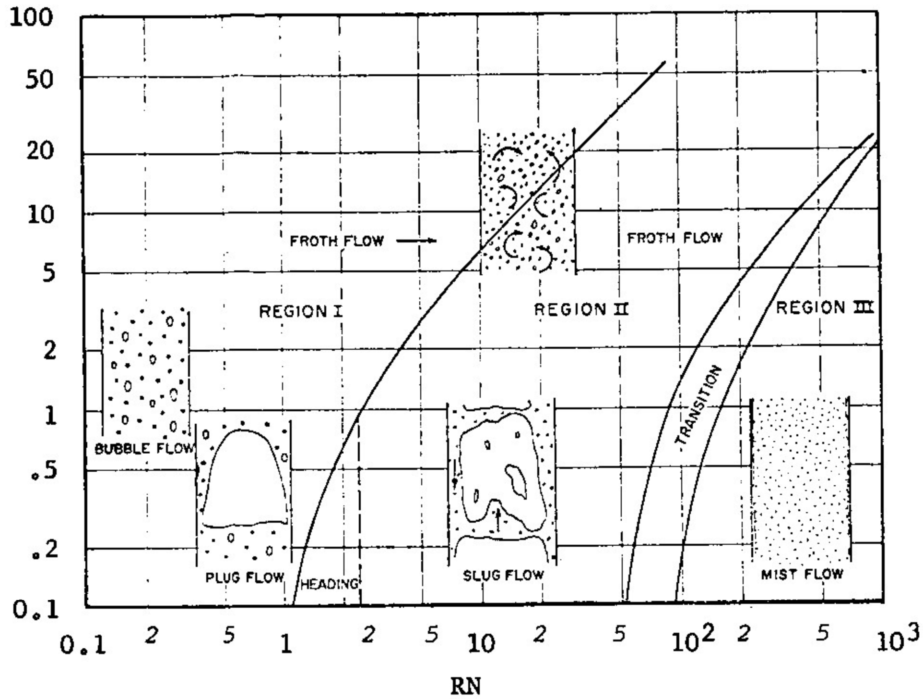


Figure 10.11 One Correlation of Vertical Flow Regimes

In Region I, at low gas numbers and high liquid numbers, one encounters a liquid with gas bubbles in it, so long as the gas-oil ratio is relatively low and the flowing pressure gradient primarily is the static head plus liquid friction loss. The relative magnitude of friction loss depends on flow rate.

For liquid rates less than 0.4 m/s [1.3 ft/sec], increased gas flow causes the bubbles to combine and form plugs. As gas flow increases further these plugs collapse and form slugs. In these regions wall friction is rather negligible.

If v_{sL} is still less than 0.4 m/s [1.3 ft/sec] but v_{sg} is about 15 m/s [49 ft/sec], or greater, the slug flow of Region II changes to mist flow of Region III. At this point the gas becomes the continuous phase with the liquid in droplet form and as a film along the wall. In Region III wall friction is a major factor in pressure loss. The liquid on the surface occurs in "ripples" which affect wall friction. The magnitude of these ripples (and thus wall friction) increases dramatically with gas flow rate. Liquid holdup in Region III is small.

Froth flow which occurs across the lines of Regions I and II occurs at high liquid velocities. Duns and Ros expect it to occur when v_{sL} is greater than 1.6 m/s [5.2 ft/sec]. At such rates no plug or slug flow was observed. No set flow pattern can be discerned. Only at high gas flows does separation of phases become discernible.

Regardless of the specific correlations used for calculation of a particular vertical system, Figure 10.11 and the implications of it provide a real insight into what is going on in the vertical pipe.

What are the primary factors? Liquid and gas velocities, pipe diameter, fluid viscosity, liquid surface tension and fluid density. Gas-oil ratio is a factor but may be represented (indirectly) by the superficial velocities and diameter. Stated simply, the type of flow is governed by fluid physical properties, their relative quantities and the degree of turbulence and other forces affecting fluid distribution in the pipe.

A group of correlations equivalent to those for horizontal flow are available for vertical flow. One of the earlier ones (Poettmann and Carpenter) is suitable for manual calculation.[10.27] This correlation works well for crude oil streams with a gas-oil ratio to about 180 m^3 gas/m^3 liquid [1000 scf/bbl] when the oil viscosity is less than 10 cp.

The friction factor plot shown in Figure 10.12 was prepared with the ordinate "f" as the composite friction factor; the abscissa as the numerator of Reynolds number, $dv\rho$.

$$d\,v\,\rho = \frac{A\,q_o\,m_o}{d} \qquad (10.62)$$

Where:		Metric	English
q_o =	oil production	m^3/d	bbl/day
m_o =	$\frac{\text{total mass of fluids}}{\text{volume of oil}}$	kg/m^3	lbm/bbl
d =	pipe diameter	m	ft
A =	unit conversion constant	9.88(E–06)	1.47(E–05)

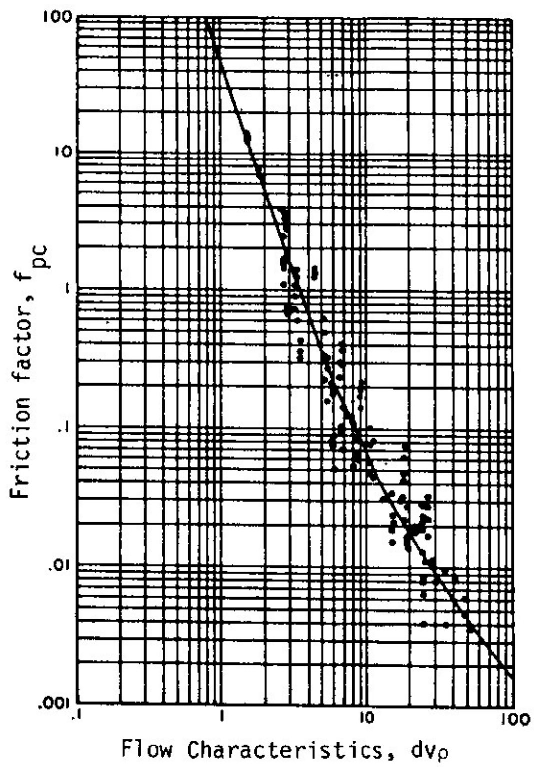


Figure 10.12 Poettman-Carpenter Friction Factor Correlation

The value of "m_o" may be found as:

$$m_o = A\left[\gamma_o + \left(\frac{q_w}{q_o}\right)(\gamma_w)\right] + (GOR)\,\rho_a\,\gamma_g \qquad (10.63)$$

Where:		Metric	English
γ_o =	relative density of oil	–	–
γ_w =	relative density of water	–	–
γ_g =	relative density of gas	–	–
q_o =	oil production	m^3/d	bbl/day
q_w =	water production	m^3/d	bbl/scf
GOR =	gas/oil ratio	std m^3/m^3	scf/bbl
A =	conversion constant	1000 kg/m^3	350 lb/bbl
q =	density of air at standard condition	1.21 kg/m^3	0.0764 lb/ft^3

With Equations 10.62 and 10.63, and Figure 10.12, one then can determine the Poettmann-Carpenter friction factor (f_{pc}). It is related to pressure loss by the equation

$$\Delta P/\Delta L = \rho_m (g/g_c) + \frac{f_{pc}\, q_o^2\, m_o^2}{A\, \rho_m\, d^5} \qquad (10.64)$$

Where:		Metric	English
ΔP	= total pressure loss	Pa	lbf/ft^2
ΔL	= length of pipe	m	ft
ρ_m	= mixture density	kg/m^3	lbm/ft^3
d	= pipe I.D.	m	ft
g/g_c	= force equivalent	9.81	1.0
A	= unit conversion constant	2.09(E+09)	7.41(E+10)

Although temperature does not occur directly in the above procedure, some average value must be known (or assumed) to calculate some of the variables.

Equation 10.64 is in oilfield terms but it is derived directly from the energy balance for fluid flow. The first term is the elevation term and the second term is the friction loss term. The equation could be written as

$$\Delta P = (g/g_c)(\rho_m)(\Delta L) + \frac{2\, f_{pc}\, (\Delta L)\, v_m^2\, \rho_m}{g_c\, d} \qquad (10.65)$$

The only difference between this equation and the single phase version is the use of mean density and velocity (along with a special friction factor plot). Mean density is the mass of total fluids flowing divided by the total volume of fluids flowing at a mean P and T. The constant in Equation 10.64 merely is the summation of all numbers to convert from velocity to a mass term and the numerical constants.

This calculation has been reduced to a series of charts. Figures 10.13 and 10.14 are for 2-inch and 2.5-inch tubing, respectively. The dashed lines are for a flowing density of 80 kg/m^3 [5 lb/ft^3], which show a reversal of position from the other data.

The use of this procedure is illustrated by Example 10.14 taken from Reference 10.28.

The same result could have been obtained from Figure 10.14. Multiply 302 times 343 to obtain 103 600 pounds of total fluid per day. Enter the left ordinate at about 103, proceed horizontally to the curved line for an average density of 40 lb/ft^3. Read vertically to lower abscissa to read about 0.285 psi/ft, checking the calculation.

Like all such calculations, one needs an average pressure and a temperature for the line segment being calculated to determine fluid properties. A useful correction for estimating temperature is Figure 10.15. The geothermal gradient in the area usually is known.

In using this correlation, calculate the total oil and water flow rate in m^3/day [bbl/day]. Proceed vertically to the appropriate geothermal gradient and read horizontally to the ordinate to find the *flowing temperature gradient* in °C/100 m [°F per 100 feet]. This number can be used to estimate average temperature for a given segment or over the entire wellbore.

Average pressure can be found by trial-and-error by assuming a value, calculating total ΔP, and then checking to see if the calculated mean pressure assumed checks the one calculated. If not, iterate until it does.

Sometimes in the early stages one does not know very much about the oil being handled. Figure 10.16 from Standing may prove useful for estimation purposes.

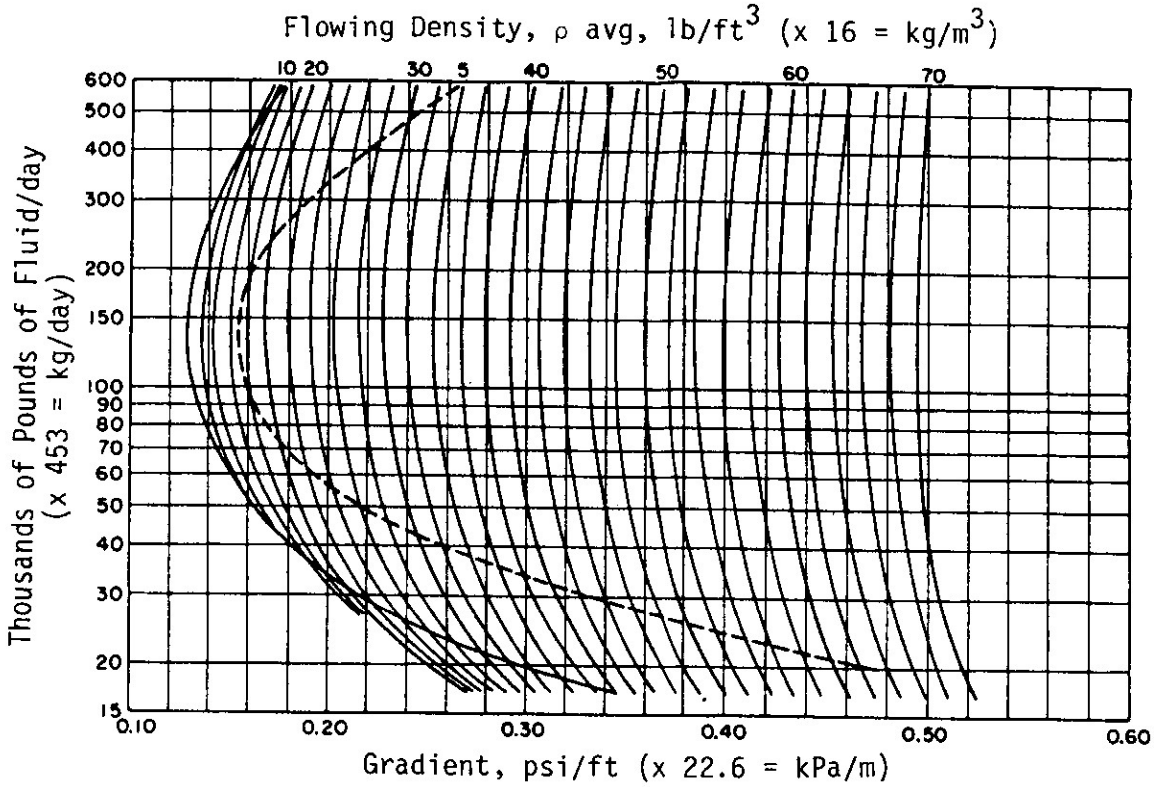


Figure 10.13 Pressure Gradients for 0.0507 m [1.995 in.] I.D. Tubing

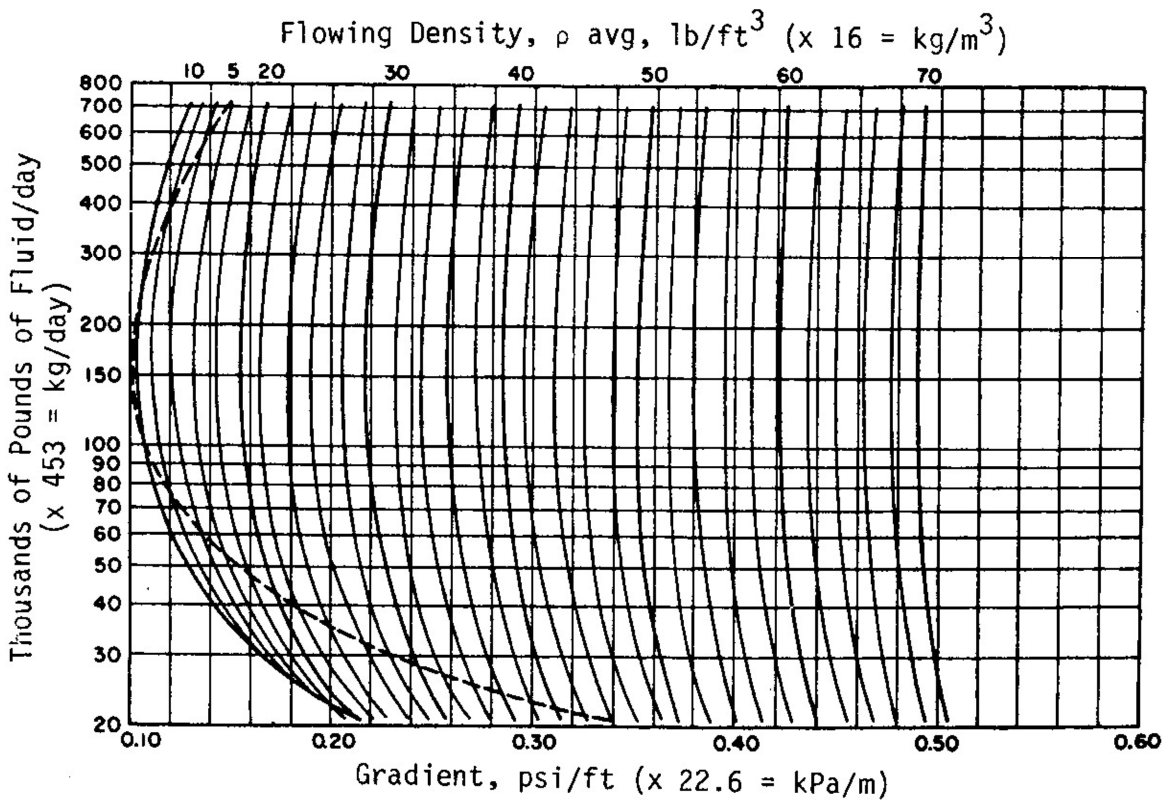


Figure 10.14 Pressure Gradients for 0.0620 m [2.441 in.] I.D. Tubing

Example 10.13: Correlate the field data of a flowing well and calculate pressure gradient, given:

Oil production = 48 m^3/d [302 STB/day]
Relative density of st oil = 42.3°API (*st* refers to "stock tank")
Gas-oil ratio = 167 m^3/m^3 [936 scf/stb]
Relative density of gas = 0.816
Tubing size = 0.062 m [2.441 in. I.D.]
Tubing pressure = 5.9 MPa [860 psia]
Flowing bottom-hole pressure = 17.4 MPa [2530 psia]
Depth to middle of perforations = 2088 m [6850 ft]
Average flowing temperature = 38°C [100°F]
No water production
Average density = 640 kg/m^3 [40 lbm/ft^3]

English: The relative density of the stock tank oil at 60°F is

$$\gamma_o = \frac{141.5}{131.5 + 42.3} = 0.814$$

Substituting in Equation 10.63, the weight rate of flow is

$$m_o = (350)(0.814) + 0.0764(0.816)936 = 343 \text{ lbm/st Bbl}$$

From Equation 10.62:

$$d\,v\,\rho = \frac{(1.47 \times 10^{-5})(302)(343)}{2.441/12} = 7.51$$

From Figure 10.12:

$$f_{pc} = 0.13 \text{ (approximately)}$$

From Equation 10.64:

$$\Delta P \Delta L = 40 + \frac{(0.13)(302)^2(343)^2}{(7.41 \times 10^{10})(40)(2.441/12)^5}$$

$$= 41.4 \text{ lbf/ft}^2\text{/ft} = 41.4/144 = \underline{0.29 \text{ psi/ft}}$$

Metric:

$$d\,v\,\rho = \frac{(9.88 \times 10^{-6})(48)(982)}{0.062} = 7.51 \quad , \quad f = 0.13$$

$$m = (1000)(0.814) + (168)(1.21)(0.816) = 982 \text{ kg/m}^3 \text{ oil}$$

$$\Delta P \Delta L = 640(9.81) + \frac{(0.13)(48)^2(982)^2}{(2.09 \times 10^9)(640)(9.16 \times 10^{-7})} = 6.514 \text{ kPa/m}$$

The Poettmann-Carpenter method is an example of flow where liquid is definitely a continuous phase and gas probably is discontinuous. This is really a form of *distributed* flow rather than intermittent flow.

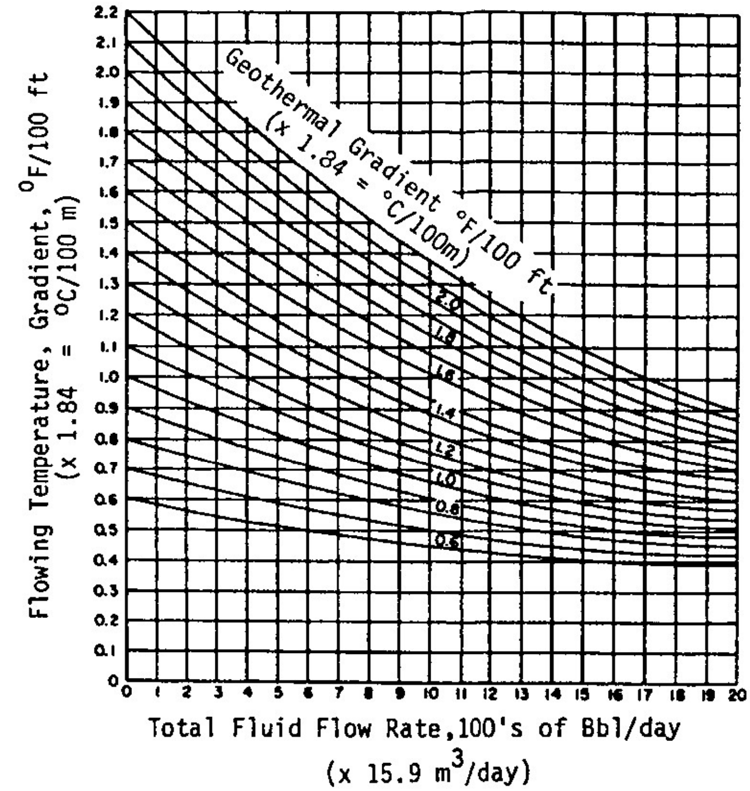


Figure 10.15 Flowing Temperature Correlation.

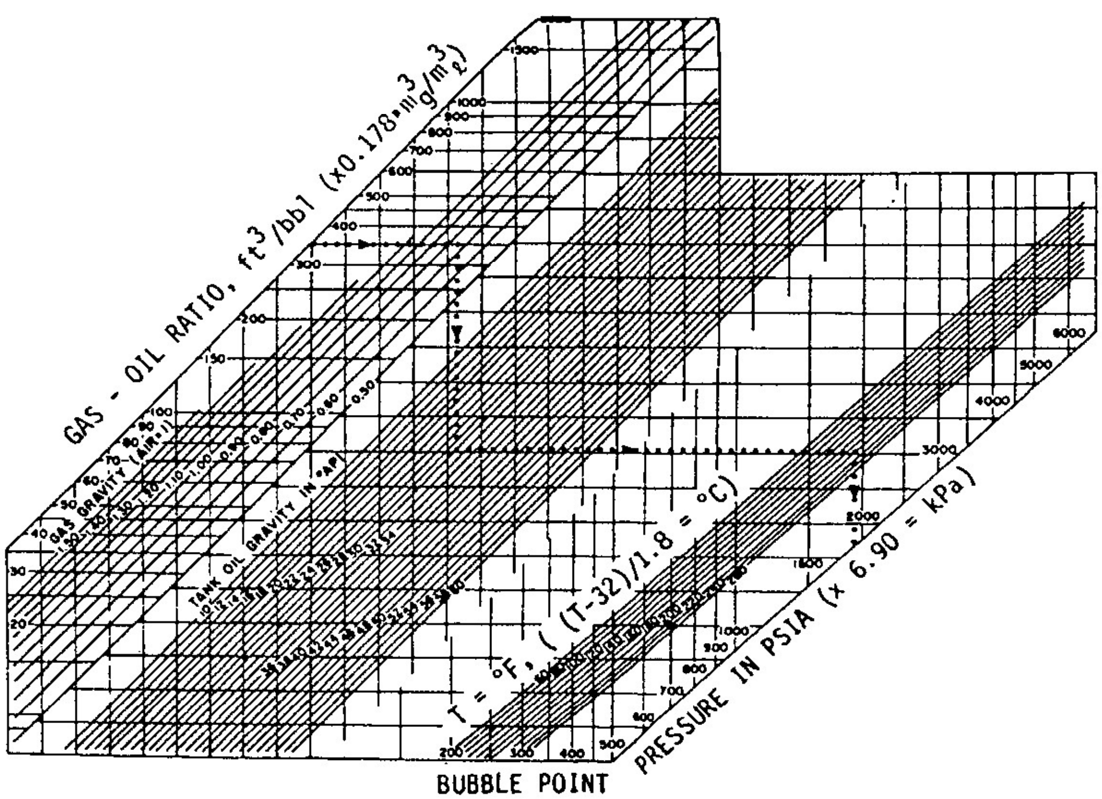


Figure 10.16 Standing's Solution Gas-Oil Ratio Correlation

THE CALCULATION STRATEGY

The two methods outlined herein are only two among many correlations available. The situation is much like PVT equations of state. All are empirical and use the same fundamental principles. They differ only in the manner in which they handle variables and the data used to obtain their numerical value. Names like Beggs and Brill, Martinelli, Dukler, Guzhov, Orkiszewski, etc., are used on different calculation models. These serve as an equation format which may be modified to fit data on a specific system.

No one model is inherently superior. Insofar as I am concerned, the strategy that should be followed is expressed by the opinions I added to Reference 10.1 in the process of preparing it for publication. A version of that follows.

Horizontal Flow

The relative amounts of the two phases and their velocities are of fundamental importance. The basic method of calculation depends on these, for they determine to a large degree the manner in which the phases interact (flow regimes).

Although a number of different flow regimes have been identified (directly or indirectly), the number probably exceeds true calculational ability. Having a large number of regimes enhances the researcher's chances of obtaining an empirical data fit (by working backward). But then, working the results forward (for real systems) poses problems. This is indicated indirectly by the fact that different investigators cannot agree on where to place the lines between different regimes.

It seems most logical to contend with only three basic flow regimes:

Distributed (mist, bubble, froth). – Where there is only one continuous phase, the other phase(s) being distributed throughout this phase as drops or bubbles.

Segregated (stratified, wave, annular). – Both phases present are continuous with the gas phase moving faster with a resultant shear force at the interface.

Intermittent (slug, plug). – Continuity of both phases is interrupted and a series of inertial and kinetic force changes occur throughout the line section.

The latter is, of course, most difficult because the overall performance of the line segment is the summation of a series of intermittent, somewhat random occurrences at various points within that segment.

Transition zones do occur but by the very definition of the word "transition" these are temporary, unstable regions that should not last very long for a properly operated line. For this reason, they should have little design application.

As a design factor, holdup is a very general term that tries, but cannot describe localized behavior within a segment. Yet, it is about all one currently can do. If it is assumed that over a finite period liquid plus gas out equals that coming in, the value of this number can be estimated from specified or calculated values. At least tentatively one can establish the likelihood of one of the three regimes above.

If the relative quantity of the continuous phase is above 80% and that phase is in *fully turbulent flow*, it is difficult to envision anything other than distributed flow as the predominant regime. Full turbulence would be determined by the usual criteria for gas and liquids, as the continuous phase.

What about liquid holdups from 20-80%? Intermittent flow is difficult to achieve unless the gas is in fully turbulent flow. The magnitude of the shear forces and the axial forces present will tend to produce some degree of intermittent flow as the "waves" formed begin to have an amplitude equal to pipe diameter.

It is probable that true segregated flow occurs only when the gas velocity (based on the fraction of the total area available to it) is in *partially turbulent flow* only. The degree of this most certainly depends on pipe diameter and fluid properties.

If one does not have distributed or segregated flow conditions available, the flow is intermittent by default. This is obvious, but the implications may not be. These are discussed in the next section.

> *Fluid Properties.* – It is routine to calculate density, viscosity and surface tension for a given phase. But, energy balances require composite properties for separate phases. The best one can do is calculate some kind of mean number that fits the correlation being used. This probably is not critical concern since the P and T used to obtain the value may contain more error than the calculation mode itself.
>
> *Line Size.* – This obviously is a factor. Pipe smoothness increases and friction drop becomes relatively less as pipe diameter increases. As the diameter gets larger the liquid slugs also get larger, by the very geometry involved. The waves in segregated flow must get higher to occupy the full cross-section.

In spite of this, it is interesting that data from small diameter pipes can be effectively applied to large diameters if the model used properly identifies the relative magnitude of the various forces involved. Diameter effects are represented by the frictional, kinetic, acceleration and gravity forces in the model.

> *Inclination.* – Uphill sections can exhibit all flow regimes. In the downhill section segregated flow normally will predominate if the volume of this section is comparable to the preceding uphill section. Gravity will separate the phases. If this is true, the pressure recovery probably can be represented reasonably by the gas static head involved.
>
> *Pressure.* – This effect shows up in both flow rate and fluid property calculations. However, scaling low pressure data to high pressures introduces a judgmental problem. For example, the density of gas changes far more rapidly with pressure than liquid. At 6.9 MPa [1000 psi] the typical natural gas is about 80-85% denser than at atmospheric pressure. Liquid density does not change very much. Thus, gas shear and impact forces change drastically, among other things.

Once again, though, the application of sound models based on lower pressure data to higher pressure systems does not appear to introduce serious error.

> *Temperature.* – Like pressure, this has an obvious effect. Calculation of a temperature profile is an approximation at best. Lines are coated and may be covered with concrete; water currents and surrounding soil characteristics vary. The method of temperature prediction is from actual data on similar systems (preferably under the same basic surrounding conditions).
>
> *PVT Behavior.* – A prediction is necessary for the real system. In almost all research studies, deliberately picked stable liquids eliminated this problem.

Of concern is changing liquid/gas ratios and fluid properties. Line segments must be chosen so that the change in these values is not so dramatic in that section as to compromise the calculations being made. The line segments need not be of equal length. Once the temperature approaches that of the surroundings, PVT behavior is relatively insensitive to subsequent changes in pressure, in most cases.

Vertical Flow

The same factors apply, although the character of flow is somewhat different because gravity is working opposite to flow forces instead of at right angles to them. There really are only two basic flow regimes – distributed and intermittent. One can obtain a liquid ring around the pipe but this is either a transition zone or a rather thin liquid ring with mist flow.

Distributed flow offers no particular calculation problem. Several methods are available which give useful results. Slug flow is a problem, both from a design viewpoint and operations. No one method is completely adequate for slug flow design.

APPLICATION OF THE TECHNOLOGY

It is no trick to design a two-phase line for a single flow rate with fixed gas/oil ratios and fluid properties. It could be sized so that flow is rather steady and liquid surging is handled easily with rather ordinary equipment. If flow is distributed, the design accuracy may be 5 to 10%. At worst, for any flow regime, it may be 15 to 20%. All of these are adequate for economic design.

The problem is created by the usual real situation that occurs. Early on in most cases, the line will be operating at less than its final design rate. Often it is not even certain at this point what the final rate will be. Also, reservoir characteristics dictate that the gas/oil ratio will change with time. So ... in the real world this is an exercise in uncertainty. The input quantities may be far more uncertain than the calculations employed. What does one do to reduce the risk to an acceptable level?

The first step is to produce design specifications that are a realistic picture of the future. These cannot be obtained in a committee meeting or independently by a contractor. They must involve input by geological, reservoir, production and even marketing specialists. Needed are forecasts of development patterns, performance expectations and the like. Required is a strategy for line development. Offshore, the line to shore will be installed early so it must reflect all of these factors. Early in its life it will be inefficient and possibly exhibit higher than normal instability. Predicting pressure loss is only a part of the problem. Equipment at one or both ends of the line must be able to handle the fluctuating pressures and liquid expected at all stages in line life. *Pigging* may be required during early life to compromise the problem.

There are numerous programs, correlations, and "experts" around to "help" the engineer calculate what is one of the more difficult yet common problems – two-phase flow calculations. Most experts will spend a great deal of time explaining why their favorite method is actually giving the correct answer even though it doesn't match the real world.

If this early planning is done intelligently and professionally, the stage is set for the next step – the engineering calculations. There are three basic strategies one can follow.

1. Pick your (or the company's) single favorite calculation method and presume it will be right.
2. Choose several of the "popular" calculation methods and use them to develop a range of results.
3. Compare your system's characteristics with various documented data and choose several methods that have been tested on comparable systems.

These are listed in order of usual preference. The first is closer to religion than engineering. There is no one best method for all applications. If (1) is used, more luck than skill is involved.

Table 10.7 gives some notes on some of the commonly available systems as a general guide. This table should not be taken as all inclusive or an endorsement; it is more a list of those systems "commercially available."

Both (2) and (3) may be satisfactory but (3) is the best approach. The system characteristics may indicate use of a given correlation in the suite of calculations, even though it is not widely used. The suite probably should use three calculation approaches, throughout the range of flow conditions expected for different diameters of lines. As in single-phase lines, more than one diameter of line will work. Unlike single-phase lines, though, over-sizing should be avoided. It is critical to have a high enough gas velocity to keep the liquid moving continuously. So ... think small. This tends to compromise the changing flow rates during line life.

TABLE 10.7
Two-Phase Pressure Drop Correlations

Correlation	Notes
Vertical Upward Flow	
Duns & Ros	Good in mist and bubble flow regions.
Angel-Welchon-Ross	Applicable for high flow areas and annulus flow. Recommended for high volume wells and low gas/oil ratios.
Hagedorn & Brown	Best available pressure drop correlation for vertical upward flow. Most accurate for angles of inclination greater than 70 degrees.
Orkiszewski	Results reliable for high gas/oil ratios. Most accurate for angles of inclination greater than 70 degrees.
Aziz	Generally slightly overpredicts pressure drop; other correlations tend to underpredict. This fact can be used to bracket the solution. Most accurate for angles of inclination greater than 70 degrees.
Beggs & Brill	Good for all angles of inclination. Predicts the most consistent results for wide ranges of conditions.
Gray	Specifically designed for condensate wells (high gas/oil ratios). Recommended ranges: velocity < 15 m/s [50 ft/sec]
Horizontal Flow	
Lockhart-Martinelli	Widely used in the chemical industry. Applicable for annular and annular mist flow regimes if flow pattern is known a priori. Do not use for large pipes. Generally overpredicts pressure drop.
Eaton	Do not use for diameters < 50 mm [2 in.]. Do not use for very high or very low liquid holdup. Underpredicts holdup for $H_L < 0.1$. Works well for $0.1 < H_L < 0.35$.
Dukler	Good for horizontal flow. Tends to underpredict pressure drop and holdup. Recommended by API for wet gas lines.
Beggs & Brill	Use the no-slip option for low holdup. Underpredicts holdup. Most consistent and well-behaved correlation.
Inclined Flow	
Mukherjee-Brill	Recommended for hilly terrain pipelines. New correlation based heavily on in situ flow pattern. Only available model that calculates flow patterns for all flow configurations and uses this information to determine modeling technique.

The correlations available can be divided into pre- and post-computer periods. In the pre-period, the complexity of the correlation was limited by calculational ability. The "fitting" process, therefore, dealt with the overall factors affecting flow only. In the post-period it was convenient to use more variables and introduce more empirical constants. There are advantages in this from a correlation development viewpoint, but it also allows one to out-compute real knowledge about the system being designed. The bottom line of all this is that one should not choose or discard a calculation method because of its age or complexity.

The circumstances affecting a calculation strategy affect the choices. If one works for a large company with research capability, it is likely that test have been run on actual systems. It is equally likely that basic models have been modified to fit this information. An expanded or modified version of existing correlations has been developed. This is good because it adds to knowledge. The correlation may be better than the basic one from which it comes, but it still is not a panacea for all two-phase flow problems. The problem with such in-house correlations is that they sometimes are anointed with "sacred" powers they cannot possess.

PHYSICAL PROPERTIES

Fluid properties effect two-phase flow calculations much more than normally considered. The overall simulation accuracy depends on accurate fluid property calculations. The selection of the most appropriate correlation for any application is one of the more difficult choices made. Unfortunately most engineers avoid this choice and choose the "default" – which should be renamed the "not-my-fault" – option. The proper correlation to use is quite simply the one that matches your system.

Fluid property correlations can be divided into two major categories: 1) non-compositional and 2) compositional models. Non-compositional modeling is an empirical method of estimating those properties: vapor-liquid split, viscosity, etc. Non-compositional models are often employed because the information required – gas specific gravity, liquid specific gravity, GOR – are readily available. These methods are often less accurate at predicting certain critical physical properties. Compositional models require gas and/or liquid analysis which are often not available. Use of these models will not necessarily improve the model. Several things are necessary for these models to improve the accuracy of the fluid flow calculations – proper sampling techniques, proper analysis, proper correlation choice, etc.

Non-Compositional Correlations

Non-compositional fluid models predict bulk fluid properties from the gas and/or liquid specific gravities, gas oil ratio, water oil ratio, etc. The fluid property correlations are often divided into five subcategories: 1) blackoil, 2) gas condensate, 3) single-phase gas, 4) single-phase liquid, and 5) pure component.

Blackoil Model

"Blackoil" refers to a multi-phase fluid model commonly used in the oil and gas industry. This model predicts fluid properties from the specific gravities of the gas and oil, and the volume of gas produced per volume of oil (GOR). Empirical correlations determine the phase split and physical properties are then calculated for each phase.

The blackoil model assumes the liquid at stock tank conditions remains in the liquid phase at all pressures and temperatures. The gas can exist as free gas or as dissolved gas. This type of model is often used in reservoir simulations where the complexity of the fluid flow limits the allowable complexity of the physical property modeling. This type of model should only be used for relatively stable fluids – API less than 45 and GOR less than 5000 scf/Bbl.

In the blackoil model solution gas/oil ratio (SGOR), formation volume factor (FVF) for oil and water, and solution gas/water ratios (SGWR) are required. SGOR and SGWR are the fluid properties which determine the "in-situ" phase split. The in-situ properties, as calculated by the blackoil model, are an approximation of the true vapor-liquid equilibrium (VLE) and physical properties at a particular temperature and pressure. The FVF is used to calculate in-situ liquid densities. Several solution gas/oil ratio correlations have been developed.

Oil formation volume factor (FVF) is the in-situ volume of the oil phase occupied by one standard volume of oil. It is normally expressed as bbl/stb. This factor is used to determine the in-situ liquid density. The total volume of the oil is actually the volume of oil plus the volume of dissolved gas both corrected for pressure.

Solution gas water ratio (SGWR) is the in-situ standard volume of gas dissolved per unit volume of stack tank water. It is normally expressed as scf/stb. A correlation developed by Culberson and McKetta is often used for this calculation. This correlation assumes the dissolved gas is pure methane. The accuracy of this method is normally within 5 percent. The water formation volume factor (FVF) is the in-situ volume of the water phase occupied by one standard volume of water. It is normally expressed as bbl/stb. Water FVF is calculated from water densities. The effect of dissolved water is often neglected in two-phase flow programs. The effect on flow calculations is minimal but the effect on gas flow from a down stream separator can be significant.

Physical properties must be calculated for each phase. The density calculated greatly effects the prediction of flow by affecting the mixture density, estimated flow regime, etc. as discussed below. Viscosity is the most important, but often neglected parameter in two phase flow. Various correlations are available from different programs. In addition, some options are available for viscosity which allow the oil-water mixture is treated as one fluid, two fluids, or an emulsion. Surface tension is also used in several correlations to predict the flow regime. Once used to predict the flow regime the effect of surface tension is minimal on two-phase flow calculations. However if the wrong flow regime has been predicted, the whole calculation is suspect. In addition, most programs give the option of entering tabular physical property data to replace calculated properties..

Gas Condensate

Most programs that offer a "Gas-Condensate" option using the procedure of API 14B. This model assumes no liquid is present below the lower dew point or above the upper dew point. In-situ condensate flow rate is calculated by multiplying the standard volumetric flow rate by a pseudo-formation volume factor (pseudo-FVF). The pseudo-FVF is a function of condensate specific gravity, dew point pressure, pressure, and temperature. This model assumes the condensate density is constant and equal to the density at stock tank conditions. Dewpoint pressure, mass phase split and surface tensions of condensate and water are calculated from empirical relationships. Dewpoint pressure is a function of condensate specific gravity and temperature.

Single Phase Gas

Single phase gas properties are usually calculated by the gas physical properties listed in the blackoil model. Gas density is calculated from Equation 3.12.

Single Phase Liquid

This model uses the liquid gravity to calculate physical properties. The oil and water correlations listed above are often used with two modifications: the SGOR is zero and the liquid is assumed to be incompressible. Care should be used when entering specific gravity as many programs assume a specific gravity larger than 1.0 (9 °API, for example) is water not oil.

Pure Components

Most flow programs offer pure component options. These are usually limited to water (steam) and CO_2. Tabular or specific data correlations are used. The accuracy of most of these correlations is high for the range of normal operations.

Compositional Fluid Models

Compositional fluid modeling is a method for describing a stream based on its pure components. Equilibrium phase splits and phase properties are determined by blending the properties of the stream constituents. Typically equations of state are used to predict properties and VLE. The accuracy of a compositional fluid model depends on the accuracies of the component properties, the mixing rules, the equation of state, and the accuracy of the sample. For petroleum fractions, the programs must develop pseudo components to use in the correlation. These methods are reviewed in Volume 3 of "Gas Conditioning and Processing.". Compositional models predict the following fluid properties: 1) equilibrium K-values and VLE; 2) gas and liquid enthalpies; 3) gas and liquid densities; 4) gas and liquid viscosities; 5) surface tension; 6) gas and liquid thermal conductivities.

The Calculation Suite

There is no way one can designate in general which suite to use. The problem is very much like that in well logging or PVT behavior. The choice is dictated by what one knows about the system under study.

The newer methods tend to be more general and will be applied on a broader basis than the older methods. It is recommended, though, that one of the older methods be included in each suite. For horizontal flow where the liquid loading is within the correlation and gas is the primary use, the modified Flanigan approach may be one suitable method. If mist flow is indicated, Duns and Ros is a candidate for both horizontal and vertical flow. If liquid is the only continuous phase (bubble flow), a regular liquid flow equation with correction of physical properties for gas bubbles may be used. For vertical bubble flow, Poettmann-Carpenter may be used within the limits specified; outside these limits a modified version like that of Hagedorn-Brown may be employed.

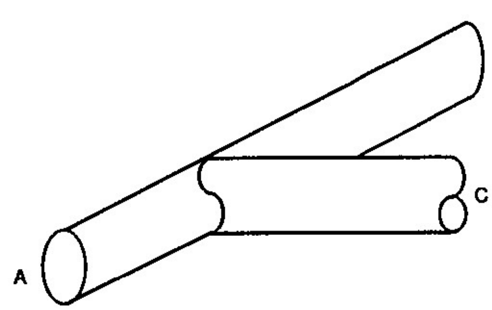
What do you do with the results of the calculations? Regard them as the *range* of pressure performance expected. Design the total system to operate *anywhere* within the pressure range and liquid handling need indicated.

A different suite may be used for early life calculations than at later life full-flow conditions. When operating at or near design capacity, pressure and liquid fluctuations should be reasonably predictable.

FLOW SPLITTING

The flow in a line may be split between two or more lines at some point. This occurs in distribution systems, at surface facilities, river crossings and the like. In a two-phase system the liquid split will not be the same as that of the gas, necessarily.

There are three forces involved in determining the amount of liquid entering each branch of the tee shown in the sketch – *inertial, gravitational* and *centripetal.* If the flow is A to B the inertial forces tend to keep the liquid flowing in a straight line. The gravitational forces tend to pull the liquid to the bottom. Thus, the angle of the side connection should make a difference. Everything else being equal, a tee in a vertically upward position should get less liquid than one in the downward position. Where the side branch is horizontal, gravity should have little effect on the split.



As the gas enters a side branch, C, a circular motion is imparted to it. When a mass traverses a circular path with constant speed, the velocity changes in *direction* but not in *magnitude*; velocity is a vector quantity. If one examines what must go on according to the laws of physics, an unbalanced set of forces occur which cause acceleration toward the center of the circle. It takes energy to produce acceleration so that a localized pressure change (drop) occurs (somewhat akin to that in an orifice meter).

Fluid always flows in a direction toward the lowest pressure available. In a horizontal tee, liquid will only enter the side tee if the centripetal forces produce a driving force in that direction that over comes the inertial forces for straight-ahead flow.

Newton's First Law of Motion says that a mass in motion will move in a straight line unless acted on by unbalanced forces causing it to deviate. In this case, the forces are gravitational and centripetal, the equations for which are:

Gravitational: $F = mg/g_c$ Centripetal: $F = mv^2/r$

Where: r = radius of circular motion

Centripetal force, a major consideration in flow-splitting, depends on velocity, the radius of the pipe and the mass involved. Velocity, in turn, depends on total flow rate. Theory suggests that the amount of liquid entering a sidebranch would depend on the amount of gas entering that branch. This has been confirmed experimentally.(10.29-31) Figure 10.17 shows the result of a test on a gas system in the Netherlands by Oranje and co-workers.

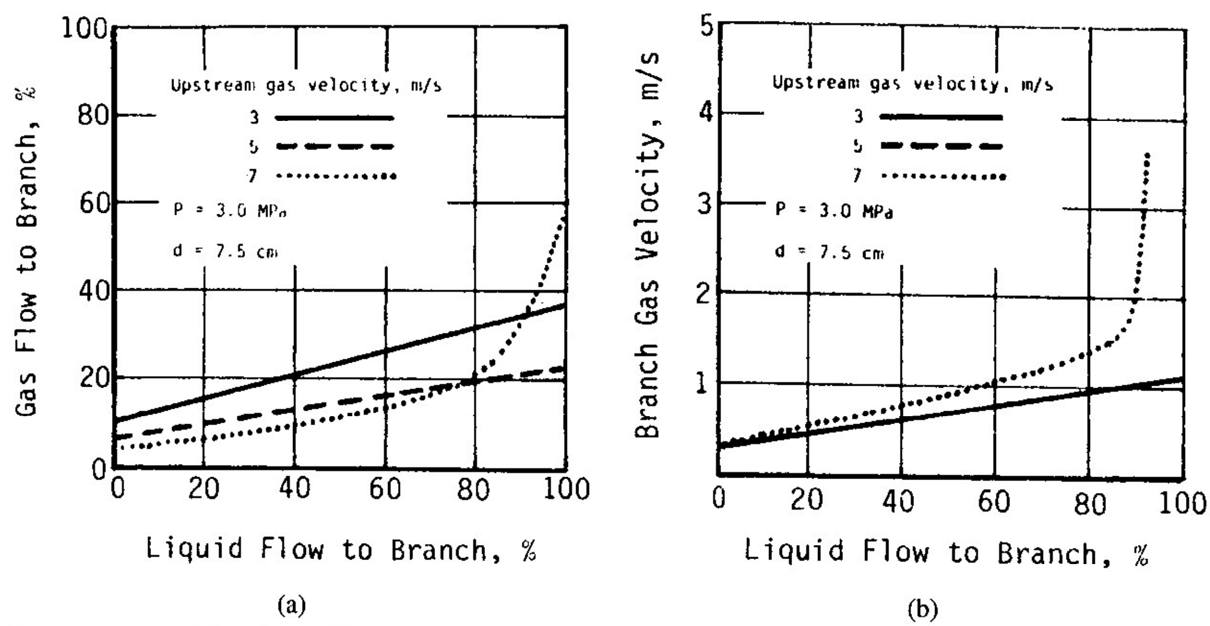


Figure 10.17 Behavior of Two-Phase Grid System

Notice the effect of upstream velocity on behavior. At 5-7 m/s [16-25 ft/sec] velocity the curves are linear; at 3 m/s [10 ft/sec] a break occurs at about 85% of the liquid flowing to the branch. This shows the effect of velocity on behavior.

Hong*(10.31)* expanded this kind of study. Although the work was done at low pressures in 9.5 mm [3/8 in.] transparent pipe, the results check reasonably well with Oranje's work in 76 mm [3 in.] pipe at higher pressure.

Once again, as gas velocity increases centripetal force increases. For a given velocity, an increase in the liquid amount increases inertial force without affecting centripetal force. Thus, more of the liquid moves in a straight line. As viscosity increases, all other factors remaining constant, more of the liquid enters the side branch. Why? Increased liquid viscosity causes increased slippage between gas and liquid, decreasing liquid velocity (and thus its inertial forces).

When the gas/liquid flows into "C" and out "A" and "B" the liquid split is proportional to the gas split when 15-85% of the gas enters "A" or "B."

OFFSHORE RISERS

The flow in a riser may differ from that in a wellbore where it is at the end of a relatively long, essentially horizontal, flow line. Holdup and surging from that line is transmitted to the relatively short riser. The riser may have to handle far more liquid than a well because the line can feed it liquid surges that far exceed those possible by gas-lift or reservoir mechanisms.

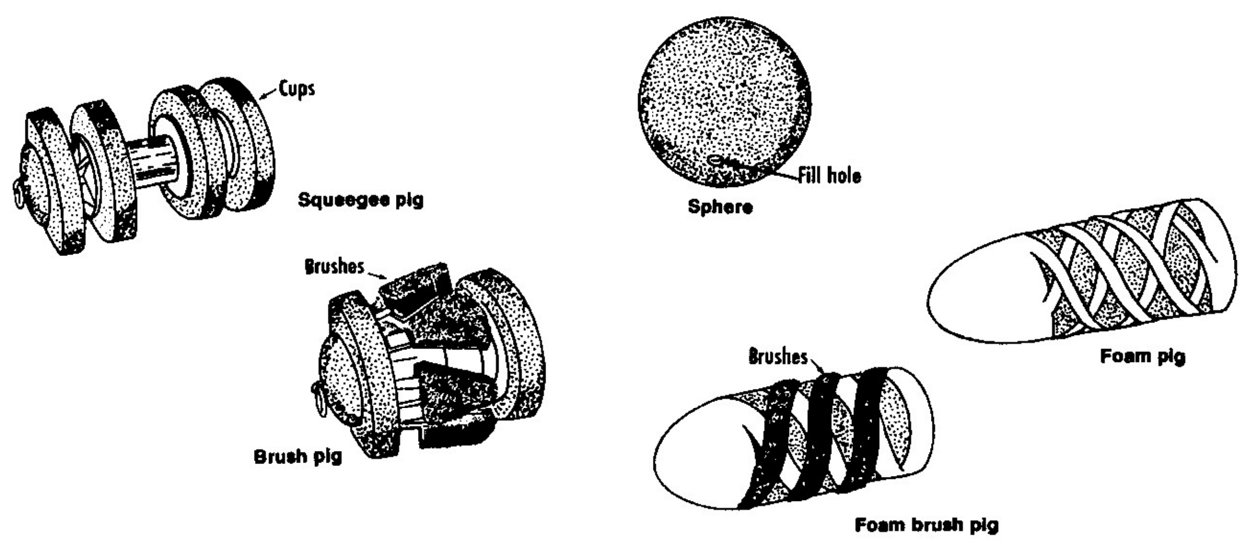
The pressure at the bottom of the riser can vary if the holdup in the riser is not about the same as in the line feeding it. If the riser holdup is too large and the gas velocity too small to provide continuous liquid lift, too much of the liquid reverses and flows downward. Liquid accumulates, causing an unstable pressure situation. This is relieved by large liquid slugs periodically leaving the riser at high velocity. The changes in liquid amount and the corresponding pressure changes can be dramatic. Onshore a large "slug catcher" installation can be provided; offshore this is uneconomical on the platform itself. This is one of the practical reasons why the line immediately ahead of the riser should be horizontal or have a slightly upward slope of 2 to 5 degrees. From some unpublished tests, the length of this section probably should be several times the riser height. The upward incline eliminates a possible "sump" effect and serves to decrease pressure/holdup instabilities.

Many studies of this problem have been made, but most have not been published. Data that have been published show how important gas velocity is in lifting liquid.[10.32-33] Gas velocities above 2.5-3.0 m/s [8-10 ft/sec] are required to minimize liquid slug length and degree of surging. It is feasible to hold liquid slug size to 10% of the riser volume.

Holding a back-pressure (choking) at the top of the riser can dampen severe slugging. But, this is not a substitute for maintenance of velocity. If flow rate varies widely, the use of multiple risers may be indicated.

PIPELINE PIGGING

Pipeline pigs come in many forms for different applications: (1) removal of liquids collecting in lines during operations or used for hydrostatic testing, (2) to scrape wax, scale and other solids from the pipe walls, (3) to separate different liquids in product pipelines, and (4) to apply internal pipe coatings for corrosion protection.



Guidelines for selecting and using pigs are readily available.[10.34-37] There are three general classifications: scraper or brush type, foam type and spheres. The choice depends on service and the need to pass through valves and turn corners.

The design and operation of launchers and receivers is summarized in Reference 10.35.

HYDRAULIC DIAMETER

Most fluid flow equations contain a diameter term, expressed as "D" or "d." When the duct for fluid flow is not circular, one must use and equivalent diameter in these equations. It is found as shown below if the duct is running full:

Annulus:	inner diameter $= d_i$ outer diameter $= d_o$	$d_e = d_o = d_i$
Square:	side $= a$	$d_e = a$
Rectangle:	sides a and b	$d_e = \dfrac{2\,ab}{a + b}$

The substitution of d_e for d in an equation theoretically is not correct when the ratio of the inner to the outer diameter exceeds 0.3 but appears suitable for use in tubing and casing.

In annular flow the wall friction might vary with the two conduits involved. If a roughness correction is being made, some average is necessary. One approach is to use the relative diameters to prorate roughness effects.