# PIPELINE ENGINEERING

## FLUID FLOW

### Mechanical Energy Balance

$$g\Delta z + \int v dp + \Delta\left(\frac{V^2}{2}\right) = W_o - \sum F \qquad (1\text{-}1)$$

potential energy change | expansion work | Kinetic energy change | Work added/ subtracted by compressors or pumps/expanders | Sum of friction losses

Note that the balance is per unit mass. In differential form

$$gdz + vdp + VdV = \delta W_o - \delta F \qquad (1\text{-}2)$$

Rewrite as follows

$$dp = -\rho\left(g \cdot dz - V \cdot dV - \delta F + \delta W_o\right) \qquad (1\text{-}3)$$

Divide by $dL$ ($L$ is the length of pipe)

$$\left.\frac{dp}{dL}\right|_{Tot} = -\rho g \cdot \frac{dz}{dL} + \rho V \cdot \frac{dV}{dL} + \rho\frac{\delta F}{\delta L} - \rho\frac{\delta W_o}{\delta L} \qquad (1\text{-}4)$$

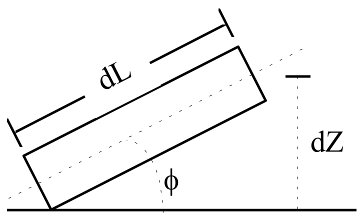
or:

$$\left.\frac{\mathrm{dp}}{\mathrm{dL}}\right)_{\mathrm{Tot}} = \left.\frac{\mathrm{dp}}{\mathrm{dL}}\right)_{\mathrm{elev}} + \left.\frac{\mathrm{dp}}{\mathrm{dL}}\right)_{\mathrm{accel}} + \left.\frac{\mathrm{dp}}{\mathrm{dL}}\right)_{\mathrm{frict}} \qquad (1\text{-}5)$$

($\frac{\delta W_o}{\delta L}$ is usually ignored, as the equation applies to a section of pipe)

The above equation is an alternative way of writing the mechanical energy balance. It is not a different equation.

The differential form of the potential energy change is




$$g\frac{dZ}{dL} = g\sin\phi \qquad (1\text{-}6)$$

Friction losses: We use the Fanning or Darcy-Weisbach equation (Often called Darcy equation)

$$\delta F = \frac{2V^2 f}{D} dL \qquad (1\text{-}7)$$

an equation that applies for single phase fluids, only (two phase fluids are treated separately). The friction factor, in turn, is obtained from the Moody Diagram below.

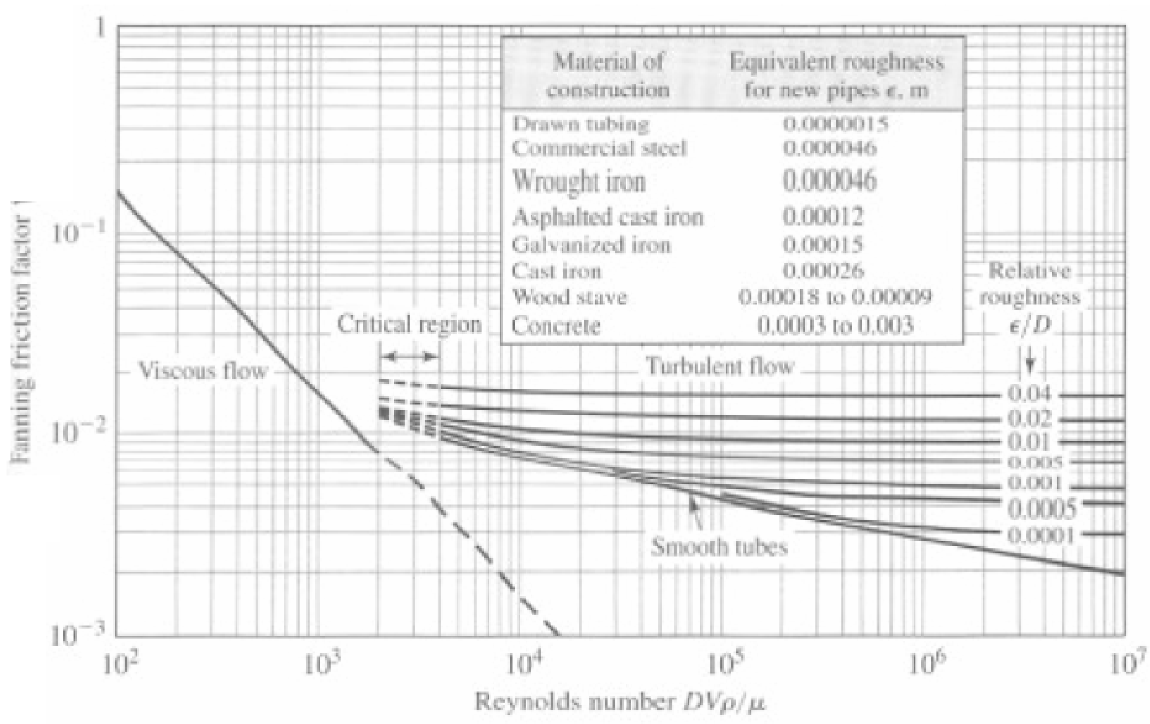



**Figure 1-1:** *Moody Diagram*

<u>Friction factor equations.</u> (Much needed in the era of computers and excel)

Laminar Flow

$$f = \frac{16}{\text{Re}} \qquad (1\text{-}8)$$

Turbulent Flow $$f = \frac{0.046}{\mathrm{Re}^a} \tag{1-9}$$

smooth pipes: a=0.
Iron or steel pipes a=0.16

Turbulent Flow $$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right) \quad \text{(Colebrook eqn)} \tag{1-10}$$

Equivalent length of valves and fittings: Pressure drop for valves and fittings is accounted for as equivalent length of pipe. Typical values can be obtained from the following Table.

**Table 1-1:** *Equivalent lengths for various fittings.*

| *Fitting* | $\frac{L_e}{D}$ |
|---|---|
| $45^{O}$ elbows | 15 |
| $90^{O}$ elbows, std radius | 32 |
| $90^{O}$ elbows, medium radius | 26 |
| $90^{O}$ elbows, long sweep | 20 |
| $90^{O}$ square elbows | 60 |
| $180^{O}$ close return bends | 75 |
| $180^{O}$ medium radius return bends | 50 |
| Tee (used as elbow, entering run) | 60 |
| Tee (used as elbow, entering branch) | 90 |
| Gate Valve (open ) | 7 |
| Globe Valve (open ) | 300 |
| Angle Valve (open) | 170 |

## Pressure Drop Calculations

**Piping is known. Need pressure drop. (Pump or compressor is not present.)**

*Incompressible Flow*

a) Isothermal (ρ is constant)

$$\left.\frac{dp}{dL}\right|_{Tot} = -\rho\left(g\frac{dZ}{dL} + V\frac{dV}{dL} + \frac{dF}{dL}\right) \tag{1-11}$$

for a fixed $\phi$ $\Rightarrow$ $V$ constant $\Rightarrow$ $dV = 0$

$$\delta F = 2V^2 \cdot f \cdot \left(\frac{\delta L}{D}\right) \tag{1-12}$$

$$\Delta p = -\rho\left[g \cdot \Delta Z + 2V^2 \cdot f \cdot \frac{L}{D} + \sum F\right] \tag{1-13}$$

b) Nonisothermal

It will not have a big error if you use $\rho(T_{average})$, $v(T_{average})$

**Exercise 1-1:**
Consider the flow of liquid water (@ 20°C) through a 200 m, 3" pipe, with an elevation change of 5 m. What is the pressure drop?

Can the Bernoulli equation assuming incompressible flow be used for gases? The next figure illustrates it.

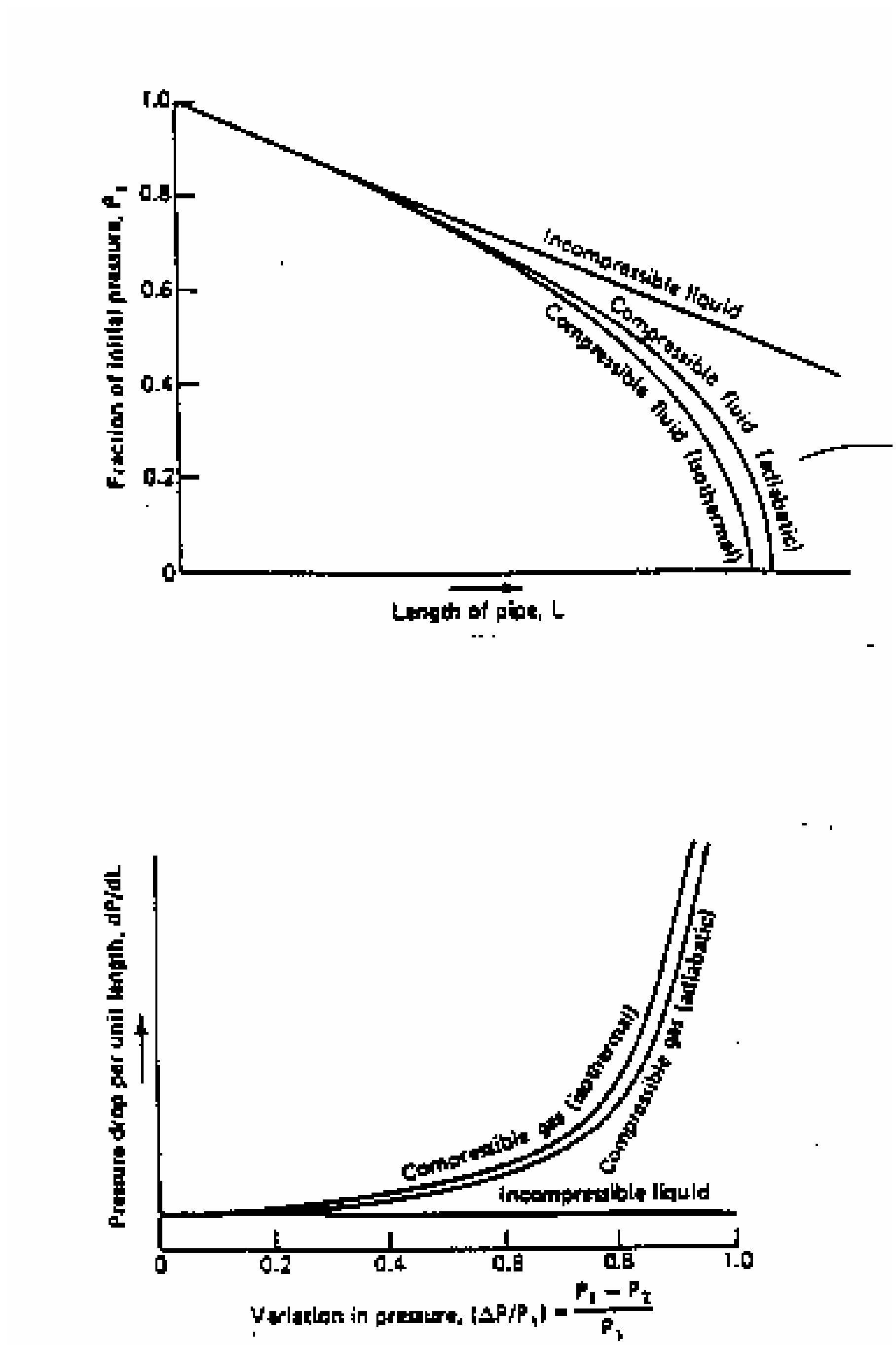



**Figure 1-2:** Error in Bernoulli equation

In conclusion, if $\frac{p_{out} - p_{in}}{p_{in}} \leq 0.2 - 0.3$ using the assumption of incompressibility is OK.

*Compressible Flow (Gases)*

a) Relatively small change in T (known)

For small pressure drop (something you can check after you are done) can use Bernoulli and fanning equation as flows

$$gdz + vdp + d\left(\frac{V^2}{2}\right) = -dF \qquad (1\text{-}14)$$

Then

$$\frac{g}{v^2}dz + \frac{1}{v}dp + \frac{V}{v^2}dV = -\frac{dF}{v^2} \qquad (1\text{-}15)$$

but $V = v\frac{G}{A}$ , where

$V$ = Velocity (m/sec)
$v$ = Specific volume ($m^3$/Kg)
$G$ = Mass flow (Kg/sec)
$A$ = Cross sectional area ($m^2$)

Then,

$$\frac{g}{v^2}dz + \frac{1}{v}dp + \left(\frac{G}{A}\right)\frac{dV}{v} = -\frac{dF}{v^2} = -2f\left(\frac{G}{A}\right)^2\frac{dL}{D} \qquad (1\text{-}16)$$

Now put in integral form

$$g\int\frac{dz}{v^2} + \int\frac{dp}{v} + \left(\frac{G}{A}\right)^2\int\frac{dV}{V} = -2\cdot\left(\frac{G}{A}\right)^2\cdot\frac{1}{D}\cdot\int f\,dL \qquad (1\text{-}17)$$

Assume

$$T_{av} = \frac{T_{in} + T_{out}}{2} \qquad (1\text{-}18)$$

$$p_{av} = \frac{2}{3}\left[p_{in} + p_{out} - \frac{p_{in}p_{out}}{p_{in} + p_{out}}\right] \text{ which comes from } p_{av} = \frac{\int_{in}^{out} p\ p\ dp}{\int_{in}^{out} p\ dp} \qquad (1\text{-}19)$$

$$f_{av} = \frac{f(T_{in},P_{in}) + f(T_{out},P_{out})}{2} \qquad (1\text{-}20)$$

The integral form will now be

$$\rho_{av}^2 \cdot g \cdot \Delta z + \int_{in}^{out} \frac{dp}{v} + \left(\frac{G}{A}\right)^2 \ln\left(\frac{V_{out}}{V_{in}}\right) = -2 \cdot \left(\frac{G}{A}\right)^2 \cdot f_{av} \cdot \frac{L}{D} \qquad (1\text{-}21)$$

Now use $p \cdot v = \frac{Z \cdot R \cdot T}{M}$, where $M$: Molecular weight. Then $\rho_{av} = \frac{p_{av} M}{Z_{av} R\, T_{av}}$, which leads to:

$$\int \frac{dp}{v} = \frac{M}{Z_{av} R T_{av}} \int p \cdot dp = \frac{M}{2 \cdot Z_{av} R T_{av}} \left(p_{out}^2 - p_{in}^2\right) = \frac{\rho_{av}}{2\, p_{av}} \left(p_{out}^2 - p_{in}^2\right) \qquad (1\text{-}22)$$

Therefore;

$$\rho_{av}^2 g \cdot \Delta z + \frac{\rho_{av}}{2\, p_{av}} \left(p_{out}^2 - p_{in}^2\right) + \left(\frac{G}{A}\right)^2 \ln\left(\frac{V_{out}}{V_{in}}\right) = -2 \left(\frac{G}{A}\right)^2 f_{av} \frac{L}{D} \qquad (1\text{-}23)$$

but,

$$\frac{V_{out}}{V_{in}} = \left(\frac{Z_{out} \cdot T_{out}}{Z_{in} \cdot T_{in}}\right) \cdot \frac{p_{in}}{p_{out}} \qquad (1\text{-}24)$$

Then

$$\rho_{av}^2 g \cdot \Delta z + \frac{\rho_{av}}{2\, p_{av}} \left(p_{out}^2 - p_{in}^2\right) + \left(\frac{G}{A}\right)^2 \ln\left(\cdot \frac{Z_{out} \cdot T_{out}\, p_{in}}{Z_{in} \cdot T_{in}\, p_{out}}\right) = -2 \left(\frac{G}{A}\right)^2 f_{av} \frac{L}{D} \qquad (1\text{-}25)$$

To calculate $Z_{av}$ Kay's rule is used. This rule states that the reduced pressure and temperature of the gas is obtained using the average pressure and temperature (as above calculated) and a pseudo critical pressure and temperature.

$$p_r = \frac{p_{av}}{p_C^{`}} \qquad (1\text{-}26)$$

$$T_r = \frac{T_{av}}{T_C^{`}} \qquad (1\text{-}27)$$

In turn the critical pressure and temperatures are obtained as molar averages of the respective components critical values.

$$p_C^{`} = \sum_i y_i\, p_{C,i} \qquad (1\text{-}28)$$

$$T_C^{`} = \sum_i y_i T_{C,i} \qquad (1\text{-}29)$$

With these values the $Z$ factor comes from the following chart:

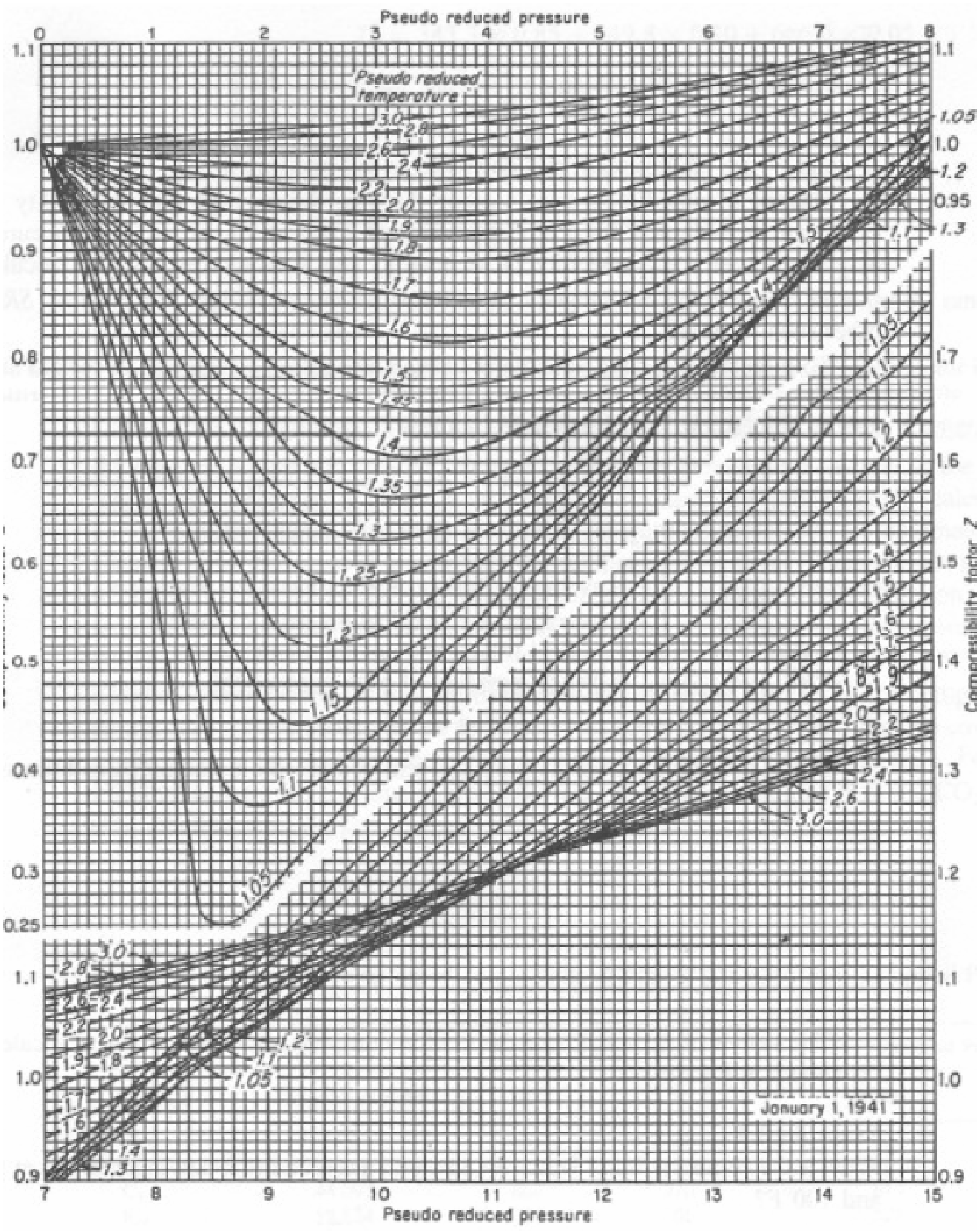



**Figure 1-3:** *Natural Gas Compressibility Chart*

Equation (1-25) can be further simplified. First neglect the acceleration term because it is usually small compared to the others, to obtain:

$$\rho_{av}^2 g\cdot\Delta z+\frac{\rho_{av}}{2p_{av}}\left(p_{out}^2-p_{in}^2\right)+2\left(\frac{G}{A}\right)^2 f_{av}\frac{L}{D}=0 \tag{1-30}$$

Form this equation we can get $G$, as follows:

$$G^2=\frac{\pi^2D^5}{32f_{av}L}\left[\frac{M\left(p_{in}^2-p_{out}^2\right)}{2Z_{av}RT_{av}}-\frac{\frac{M^2p_{av}^2}{Z_{av}RT_{av}}g\cdot\Delta z}{Z_{av}RT_{av}}\right] \tag{1-31}$$

But the volumetric flow at standard conditions is given by $p_sQ=\frac{G}{M}Z_sRT_s$ where the subscript $s$ stands for standard conditions. Therefore:

$$Q^2=\frac{\pi^2R}{32}\frac{Z_s^2\,T_s^2}{Mp_s^2}\left[\frac{\left(p_{in}^2-p_{out}^2\right)-2\frac{M\;\;p_{av}^2}{Z_{av}RT_{av}}g\cdot\Delta z}{2Z_{av}T_{av}L}\right]\frac{D^5}{f_{av}} \tag{1-32}$$

Now, if $\Delta z$=0, we get

$$Q^2=\frac{\pi^2R}{32}\frac{Z_s^2\,T_s^2}{Mp_s^2}\left[\frac{\left(p_{in}^2-p_{out}^2\right)}{2Z_{av}T_{av}L}\right]\frac{D^5}{f_{av}} \tag{1-33}$$

which can be rearranged as follows:

$$p_{in}^2-p_{out}^2=K\,Q^2 \tag{1-34}$$

where $K=\frac{64}{\pi^2R}\frac{Mp_s^2Z_{av}T_{av}}{Z_s^2\,T_s^2}\frac{f_{av}}{D^5}L$ and is known to be $W\times L$, a product of a resistant factor $W$ times the length $L$. With this, we have $W=\frac{64}{\pi^2R}\frac{Mp_s^2Z_{av}T_{av}}{Z_s^2\,T_s^2}\frac{f_{av}}{D^5}$.

To calculate pressure drop we recognize that average pressures are a function of $p_{out}$, which is unknown. Then we propose the following algorithm:

a) Assume $p_{out}^{(1)}$ and calculate $p_{av}^{(1)}$

b) Calculate $K^{(i)}=\frac{64}{\pi^2R}\frac{Mp_s^2Z_{av}^{(i)}T_{av}^{(i)}}{Z_s^2\,T_s^2}\frac{f_{av}^{(i)}}{D^5}$

b) Use formula to get a new value $p_{out}^{(i+1)}=\sqrt{p_{in}^2-K^{(i)}\,Q^2}$

d) Continue until $\dfrac{p_{out}^{(i+1)} - p_{out}^{(i)}}{p_{out}^{(i)}} \le \varepsilon$

Depending on the choice of friction term expression, several formulas have been reported for equation (1-34). They are summarized in the following table.

**Table 1-2:** *Different forms of compressible flow equations*

| Equation | Formula[a] |
|---|---|
| Fritzsche[b] | $Q_b = 1.720\left(\frac{T_b}{P_b}\right)\left[\frac{\left(P_1^2-P_2^2\right)D^5}{T_f L}\right]^{0.538}\left(\frac{1}{G}\right)^{0.462}$ |
| Fully Turbulent | $Q_b = 0.4696\left(\frac{T_b}{P_b}\right)\left(\frac{P_1^2-P_2^2}{G T_f Z_{avg} L}\right)^{0.500}\cdot \log\left(3.7\frac{D}{K_e}\right)\cdot D^{2.5}$ |
| Panhandle B | $Q_b = 2.431\left(\frac{T_b}{P_b}\right)^{1.02}\left(\frac{P_1^2-P_2^2}{G^{0.961} L T_f Z_{avg}}\right)^{0.510}\cdot D^{2.53}$ |
| Colebrook-White | $Q_b = 0.4696\left(\frac{T_b}{P_b}\right)\left(\frac{P_1^2-P_2^2}{G T_f Z_{avg} L}\right)^{0.500}\cdot\left[-4\log\left(\frac{K_e}{3.7D}+\frac{1.4126\sqrt{f}}{R_e}\right)\right]\cdot D^{2.5}$ |
| IGT Distribution | $Q_b = 0.6643\left(\frac{T_b}{P_b}\right)\left(\frac{P_1^2-P_2^2}{T_f L}\right)^{5/9}\left(\frac{D^{8/3}}{G^{4/9}\mu^{1/9}}\right)$ |
| Mueller | $Q_b = 0.4937\left(\frac{T_b}{P_b}\right)\left(\frac{P_1^2-P_2^2}{T_f L}\right)^{0.575}\left(\frac{D^{2.725}}{G^{0.425}\mu^{0.150}}\right)$ |

**Table 1-2 (continued):** *Different forms of compressible flow equations*

| | | | |
|---|---|---|---|
| Panhandle A[b] | $Q_b = 2.450\left(\frac{T_b}{P_b}\right)^{1.0788}\left(\frac{P_1^2-P_2^2}{T_f L Z_{avg}}\right)^{0.539}\left(\frac{D^{2.618}}{G^{0.461}}\right)$ | | |
| Pole | $Q_b = C\left(\frac{h_w D^5}{GL}\right)^{0.500}$ | **Pipe Diameter, (in.)** | **C[c]** |
| | | 3/4 to 1 | 1.732 |
| | | 1 1/4 to 1 1/2 | 1.905 |
| | | 2 | 2.078 |
| | | 3 | 2.252 |
| | | 4 | 2.338 |
| Spitzglass (High Pressure)[d] | $Q_b = 3.415\left[\frac{(P_1^2-P_2^2)D^5}{GL(1+3.6/D+0.03D)}\right]^{0.500}$ | | |
| Spitzglass (Low Pressure)[d] | $Q_b = 3.550\left[\frac{h_w D^5}{GL(1+3.6/D+0.03D)}\right]^{0.500}$ | | |
| Weymouth | $Q_b = 1.3124\left(\frac{T_b}{P_b}\right)\left[\frac{(P_1^2-P_2^2)D^{16/3}}{GT_f L}\right]^{0.500}$ | | |

[a] The units of the quantities in all of these equations are: $D$ = in.; $h_w$ = in. wc; $L$ = ft; $P_1$, $P_2$, $P_b$ = psia; $Q_b$ = MCF/hr; $\mu$ = $lb_m$/ft sec; and $T_f$, $T_b$ = °R.
[b] The constants 1.720 and 2.450 include: $\mu = 7.0 \times 10^{-6}$ $lb_m$/ft. sec.
[c] Includes: $P_b$ = 14.73 psia; $T_b$ = 492°R; and $T_f$ = 500°R.
[d] The constants 3.415 and 3.550 include: $P_b$ = 14.7 psia; $T_b$ = 520°R; and $T_f$ = 520°R.

## Exercise 1-2:

Natural gas (84,000 std $m^3$/hr at 49 atm and 38°C) is sent from a gas refinery to a city, through a 16" pipeline. The distance is 170 Km. The gas reaches the other end at ground temperature, (5 °C). The gas to have the following molar fractions: Methane: 98%, ethane: 1.2%, propane: 0.75%, and water: 0.05%. We also assume Re~5 $10^6$ and ε/D =0.01.

As a first approximation, we recommend using the Panhandle A equation: $p_{in}^2 - p_{out}^2 = K\,Q^{1.855}$ W= $2.552 \times 10^{-4} T_{in} \times s^{0.855}/D^{4.586}$ (Wilson G.G., R. T. Ellington and J. Farwalther, 1991, *Institute of Gas Technology Education Program,* Gas distribution Home Study Course) , where s is the gas gravity (=$M_{gas}/M_{air}$)=0.65 for natural gas), $T_{in}$ is in °R and D in inches)

What is the pressure drop?

## Heat Transfer Effects

To account for temperature changes due to heat transfer, we use total energy balance

$$gdz + d(vp) + VdV + du = \delta q + \delta w_o \tag{1-35}$$

where the following is identified:

- Potential energy change: $g\,dz$
- Rate of work done on the fluid element by pressure forces: $d(vp)$
- Kinetic energy change: $VdV$
- Internal energy changes: $du$
- Heat transfer: $\delta q$. This is given per unit mass flowing (Kcal/h)/(m$^3$/h)
- Work added: $\delta w_o$. This term is due to pumps and compressors. Since we will treat these separately, this term is usually set to zero for pipes.

But the heat $\delta q$ is given by interactions with the ambient surroundings:

$$\delta q = U(T_o - T)\frac{\pi D}{G}\,dL \tag{1-36}$$

where $U$ is the heat transfer coefficient, $T_o$ is the outside pipe temperature, $\pi DdL = dA$ (see next figure) and $G$ is the flowrate.

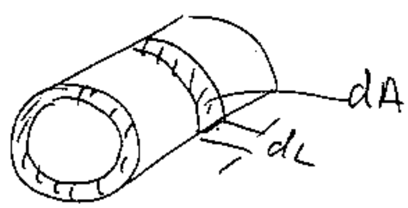



**Figure 1-4:** *Area element*

Then, (ignoring $\delta w_o$ because there are no pumps) to get:

$$gdz + dh + d\left(\frac{V^2}{2}\right) = \frac{U\left(T_o - T\right)DdL}{G} \tag{1-37}$$

Integrate and solve for $h_{out}$ (use $T_{av}$ in the heat transfer equation)

$$h_{out} = h_{in} + \frac{U\left(T_o - T_{av}\right)\pi DL}{G} - \left[\frac{V_{out}^2 - V_{in}^2}{2}\right] - g\left(z_2 - z_1\right) \tag{1-38}$$

But

$$V_{out} = v_{out}\cdot\frac{G}{A} = Z_{av}\cdot\frac{RT_{av}}{p_{out}M}\frac{G}{A} \tag{1-39}$$

Finally, to obtain the outlet temperature, one would need to obtain it form the enthalpy and pressure in the outlet

$$T_{out} = T_{out}\left(p_{out}, h_{out}\right) \tag{1-40}$$

The procedure suggested is then:

a) Assume $T_{out}$, $p_{out}$

b) Use mechanical energy balance to obtain $p_{out}^{(1)}$

c) Use total energy balance to obtain $h_{out}^{(1)}$

d) get temperature $T_{out}^{(1)}$

e) Go to b) and continue until convergence

# SCENARIO II

One has a turbine or Compressor/pump and needs $W_o$. We use total energy with $\delta q = 0$ and $dz = 0$

$$dh = \delta w_o - \frac{dV^2}{2} \tag{1-41}$$

Integrating, one obtains:

$$w_o = \frac{W}{G} = \Delta h + \Delta\left(\frac{V^2}{2}\right) \tag{1-42}$$

In this expression, we have $w_o$ given in Joules/Kg, $W$ in Joules/sec and $h$ in Joules/Kg. Thus, the work of the compressor/pump is given by:

$$W = G\left[\Delta h + \Delta\left(\frac{V^2}{2}\right)\right] \tag{1-43}$$

For compressors, $W$ is positive, while for turbines, it is negative. However, $\Delta h$ is known for liquids because enthalpy does not vary much with pressure. In addition, there isn't much temperature change in pumps). However, for gases, $\Delta h$ is much harder to obtain. Therefore we go back to the Mechanical Energy equation for pumps/compressors. Indeed, the Bernoulli equation gives

$$W = G\left[\int v dp + \Delta\left(\frac{V^2}{2}\right)\right] \approx G\int v dp \tag{1-44}$$

where the acceleration term has been neglected. For pumps, the density is constant, so one obtains:

$$W = \frac{G}{\rho} \Delta p \qquad (1\text{-}45)$$

For compressors, one needs to obtain an expression of volume knowing that the evolution is isentropic (or nearly isentropic). Thus, $pv^n$ = constant ($n=Cp/Cv$ for ideal gases $n>Cp/Cv$ for real gases). Substituting $v = p_s^{1/n} \cdot v_s \cdot \left(\frac{1}{\rho}\right)^{1/n}$ integrate to get

$$W = G\left[\frac{n}{n+1}\right] p_{in} v_{in} \left[\left(\frac{p_{out}}{p_{in}}\right)^{\frac{n-1}{n}} - 1\right] \qquad (1\text{-}46)$$

The above expression does not include the compressibility factor. A better expression, which includes the efficiency, is

$$W = G\left[\frac{n}{n+1}\right] \frac{Z_{in} + Z_{out}}{2} \frac{1}{\eta_a} RT_{in} \left[\left(\frac{p_{out}}{p_{in}}\right)^{\frac{n-1}{n}} - 1\right] \qquad (1\text{-}47)$$

The efficiency factor is usually between 60 to 80% and normally given by the manufacturer. One expression for such factor is:

$$\eta_a = \frac{T_{in}\left[\left(\frac{p_{out}}{p_{in}}\right)^{\frac{n-1}{n}} - 1\right]}{T_{out} - T_{in}} \qquad (1\text{-}48)$$

Finally, the outlet temperature is obtained from

$$p_{in} v_{in}^n = p_{out} v_{out}^n \qquad (1\text{-}49)$$

Using the gas law to obtain $v_{in} / v_{out}$ in terms of temperatures and pressures, substituting and rearranging, on e obtains:

$$\frac{T_{out}}{T_{in}} = \left[\frac{p_{out}}{p_{in}}\right]^{\frac{n-1}{n}} = [CR]^{\frac{n-1}{n}} \qquad (1\text{-}50)$$

where $CR$ is the compression ratio. Normally, manufacturers recommend not exceeding 300 $^o$F at the outlet.

**Exercise 1-3:**
The natural gas of exercise 1-2, is available originally at 2 atm. Calculate the compression work needed to reach delivery pressure (49 atm) using one compressor. Calculate the outlet temperature and determine the duty needed to cool the gas down to the corresponding inlet conditions. Is it acceptable to use one compressor?

We now discuss the compression ratio. This is limited in compressors to the range 1.2 to 6. Extra compressors should be added if the CR >6, and after-coolers need to be added to control the temperature. If more than one compressor is to be used, the practice is to use the same CR for all.

**Exercise 1-4:**
Consider two compressors.

- Write the power expression for each one assuming the gas is cooled down to its inlet temperature after compression.
- Add both expressions to obtain the total work as a function of the intermediate pressure (the rest should not be a variable)
- Take first derivative and obtain the desired result that $CR_1=CR_2$

**Exercise 1-5:**
Obtain the set of compressors needed to compress the gas of exercise 1-2 properly, that is, limiting the temperature and using the right CR.

# Two Phase Flow

Two phase flow has several regimes, which are depicted in the next figure:

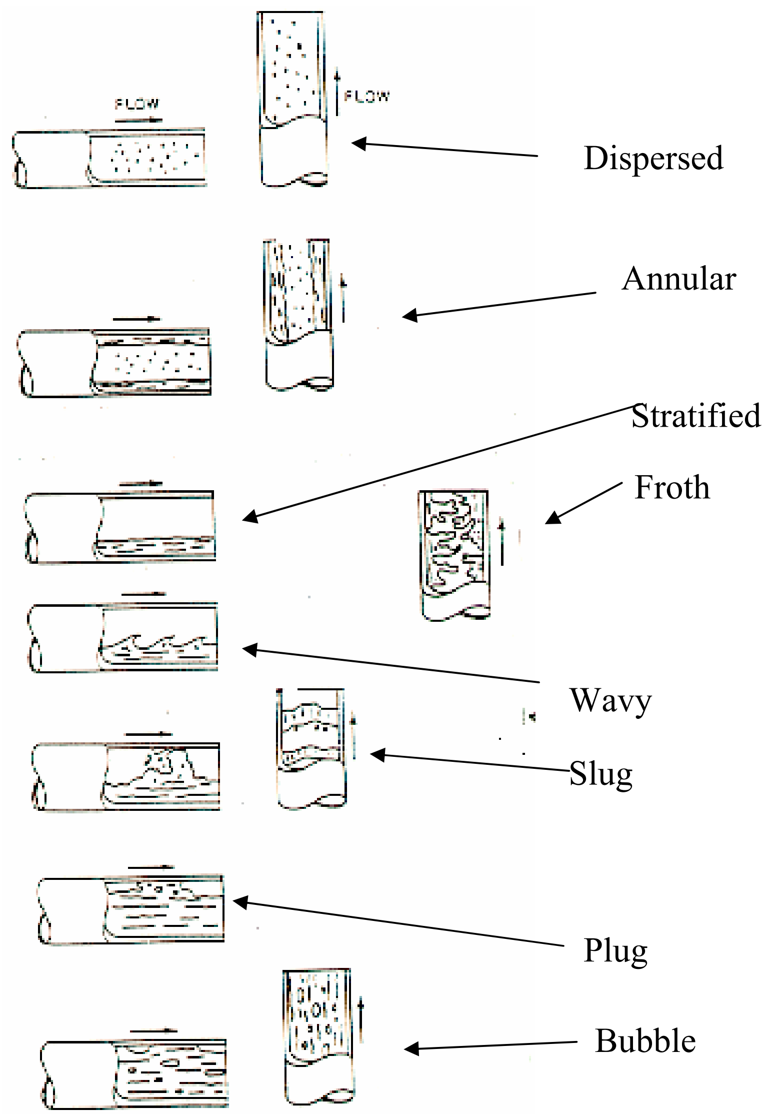



**Figure 1-5:** *Two phase flow regimes*

The two extreme cases are:

- Bubble: Vapor and Liquid in Equilibrium (Benzene 40%, Toluene 60%)
- Dispersed: Liquid and gas (air and benzene).

The latter is common in gas pipelines; the former is common in crude pipelines, especially light crudes.

One important thing to recognize is that except for the extreme cases, the phases travel at different velocities. Typical velocities are shown in the next table:

**Table 1-3:** *Typical velocities of two phase flows*

| REGIME | LIQUID VEL(ft/sec) | VAPOR VEL.(ft/sec) |
|---|---|---|
| Dispersed | Close to vapor | > 200 |
| Annular | <0.5 | > 20 |
| Stratified | <0.5 | 0.5-10 |
| Slug | 15 (But less than vapor vel.) | 3-50 |
| Plug | 2 | < 4 |
| Bubble | 5-15 | 0.5-2 |

To predict the flow patterns, one needs to use the Baker Plot (next Figure) for horizontal pipes (there is a similar one for vertical pipes).

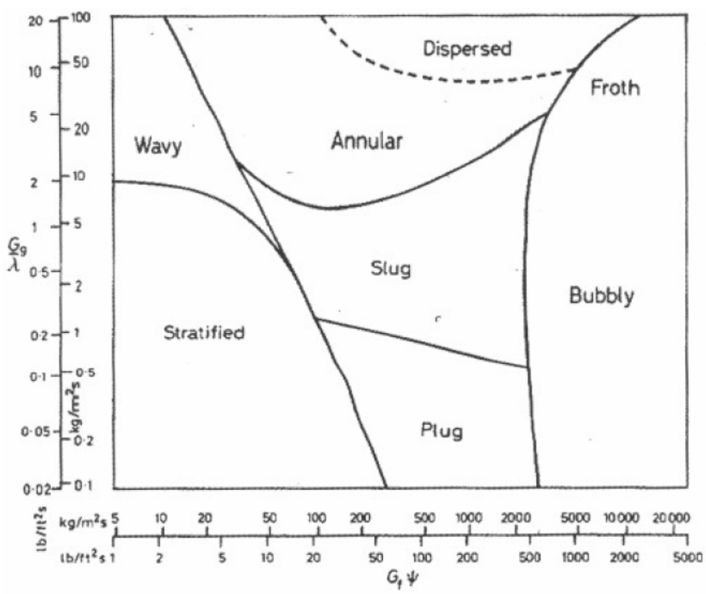



**Figure 1-6:** *Two phase flow regimes transitions*

In this diagram, we have $G_g = \left(\frac{W_g}{A}\right)$, $G_l = \left(\frac{W_L}{W_g}\right)\lambda$, ($W$ in lb/h, A in in$^2$) which are the superficial velocity of the vapor and the liquid, respectively. In turn, the parameters are given by $\lambda = 0.463\sqrt{\rho_L \rho_g}$ (with densities given in lb/ft$^3$) $\psi = \frac{1147}{\sigma}\frac{\mu_L^{1/3}}{\rho_L^{2/3}}$ (with the surface tension given in dyn/cm and the viscosity in cp)

We note that:

1) $\lambda$ and $\psi$ depend on the fluid property only
2) $G_l$ depends on the ratio of flows (Known beforehand. Not a design parameter)

3) $G_g$ depends on the vapor/gas superficial velocity. It can be modified changing the diameter
4) Transition boundaries are not at all that sharp.

From this diagram, we notice that following change of regimes in a pipe. As the pressure drop is large, then the density of the vapor is lower.

1) $\lambda\psi \square \sqrt{\rho_g} \quad \Rightarrow \quad G_l\lambda\psi \square \sqrt{\rho_g} \Rightarrow$ Abscissa decreases

2) $\frac{1}{\lambda} \square \frac{1}{\sqrt{\rho_g}} \quad \Rightarrow \quad \frac{G_g}{\lambda} \square \frac{1}{\sqrt{\rho_g}} \Rightarrow$ Ordinate increases

Thus trajectories are always "up" and "to the left". Thus a bubbly flow may become, plug, slug or annular, an annular may become wavy or dispersed, depending on the starting position in the plot, and so on.

# PRESSURE DROP

Lockart and Martinelli (1949) developed one of the first correlations. It is based on multiplying the pressure drop obtained by considering the vapor phase occupying the whole pipe, by a factor

$$\Delta p_{TwoPhase} = \phi^2 \Delta p_{VaporPhase} \tag{1-51}$$

In turn, the correction factor is given by $\phi = aX^b$, where $X = \frac{\Delta p_{LiquidPhase}}{\Delta p_{VaporPhase}}$. The following table gives some typical values of the corresponding constants:

**Table 1-4**: Constants for Lockart and Martinelli's correlation

| | *a* | *b* |
|---|---|---|
| Bubble | 14.2 | 0.75 |
| Slug | 1190 | 0.82 |
| Stratified (horizontal) | 15400 | 1 |
| Plug | 27.3 | 0.86 |
| Annular | 4.8 -0.3125 D(in) | 0.343-0.021 D(in) |

We notice that there are several more modern correlations, which will be explored later. In turn, the pressure drop due to gravity, is given by

$$\left.\frac{dp}{dL}\right)_{gravity} = \left[\varepsilon_g \rho_g + (1-\varepsilon_g)\rho_l\right] g \sin\theta \tag{1-52}$$

where $\varepsilon_g$ is the (void) fraction of gas. We omit the pressure drop due to acceleration.

# Hydrate Formation

Hydrates are crystalline structures between water and hydrocarbons. One typical example is given in the figure below:

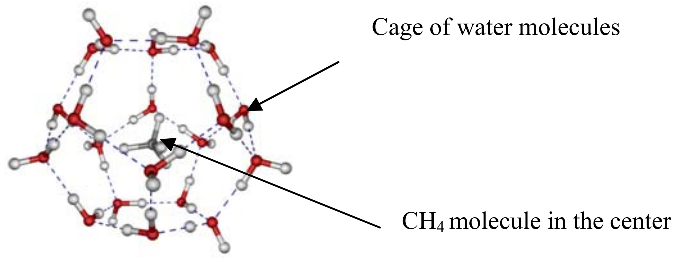



**Figure 1-7:** *Methane Hydrate*

The next figure shows the Pressure-Temperature diagram of water-hydrocarbon systems. Curve 1-1 represents the curve for vapor pressure of the hydrocarbon.

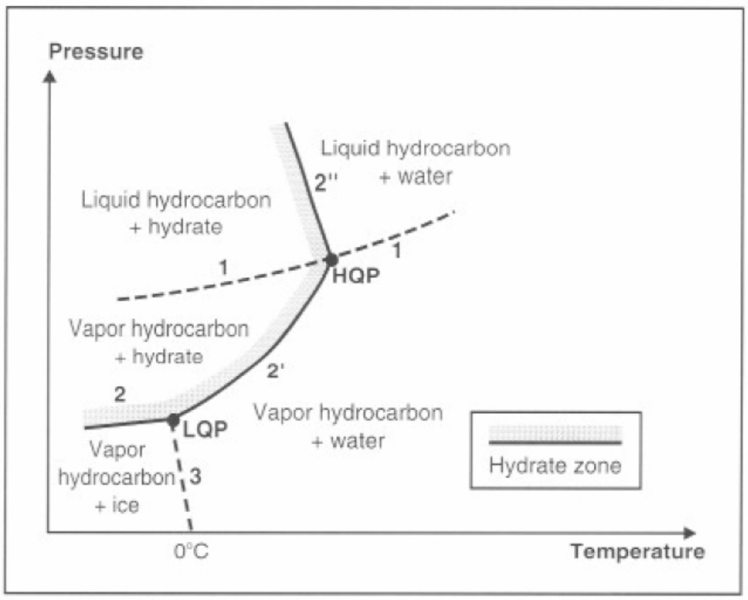



**Figure 1-8:** *Generic Hydrate P-T diagram*

The next figure shows some specific cases of hydrocarbons:

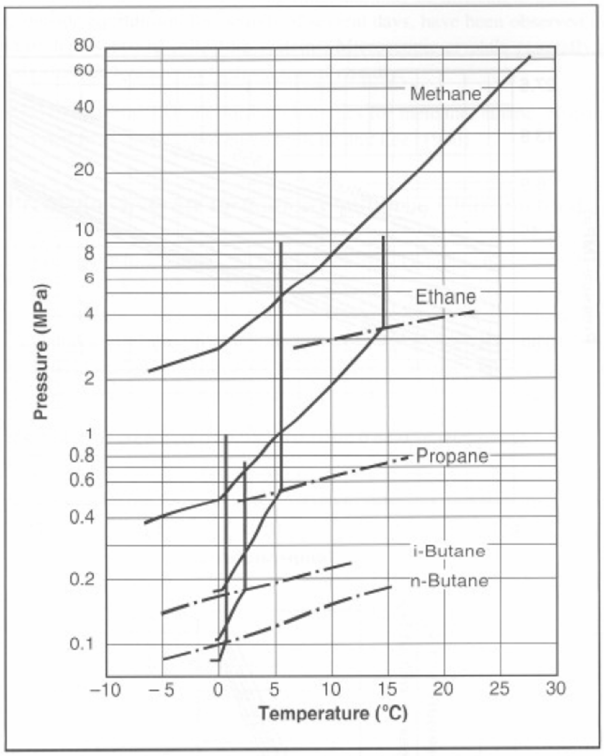



**Figure 1-9:** *Hydrate P-T diagram for various hydrocarbons*

Clearly, in high pressure pipelines, favorable thermodynamic conditions for hydrate formation can be encountered. It is therefore important to keep in mind that these conditions need to be avoided. These hydrates can be prevented from forming through heating, pressure change (not a choice in pipelines) and the introduction of inhibitors. These inhibitors are salts, alcohols, glycols, ammonia and MEA. The most widely used is methanol. The next figure shows the depression of hydrate formation temperature observed for various hydrates.

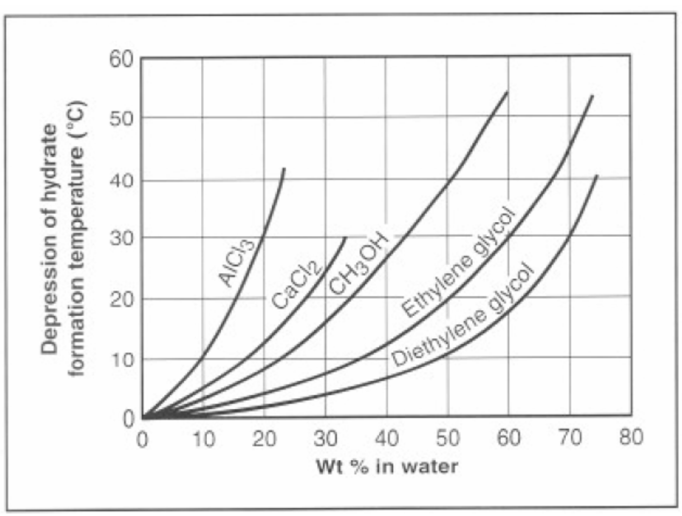



**Figure 1-10:** *Hydrate temperature formation depression*

# Pipeline Costs

Historical pipeline and compressors installed cost data were obtained from the Oil & Gas Journal special report on Pipeline Economics, September 3, 2001. Pipeline per mile cost distribution for different pipe diameters and compressor installed cost for different horsepower requirement are plotted in the following figure. All cost figures are updated to 2005 dollars using Marshal & Swift cost indexes.

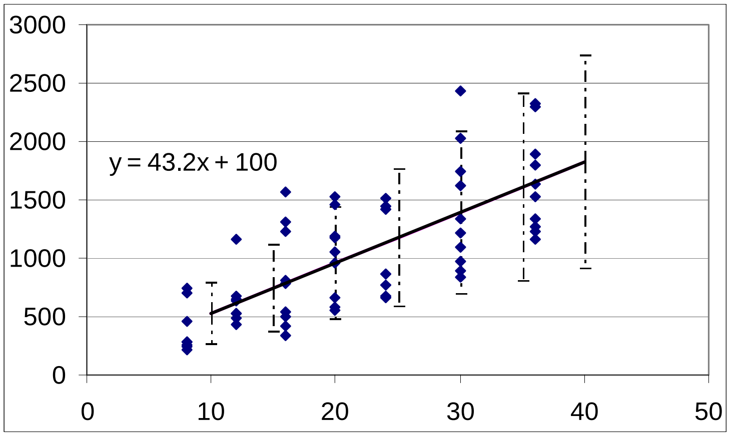



**Figure 1-11:** *Pipe average cost (k$/mile) vs. ID*

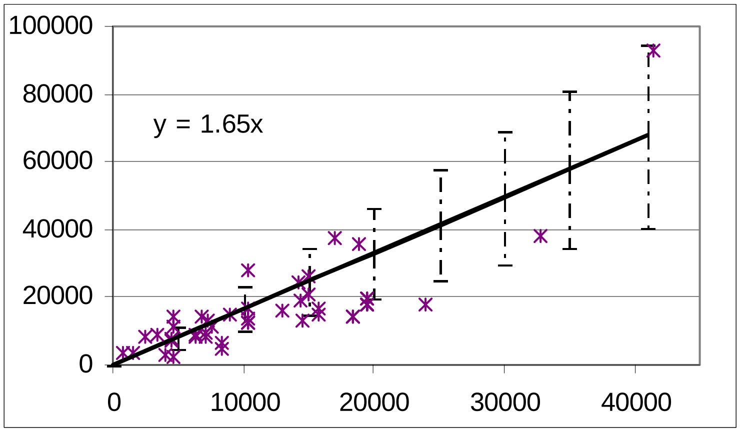



**Figure 1-12:** *Compressor cost (k$) vs. horsepower*

Fixed Capital Investment were calculated by adding the installed cost of a pipe length (assumed 5000 miles) and the cost of all required recompression stations. The Fixed Capital Investments obtained are then divided by the pipe length to obtain a *per mile* cost profile for different flow rates. The curve in the next figure shows that this cost profile takes a logarithmic shape.

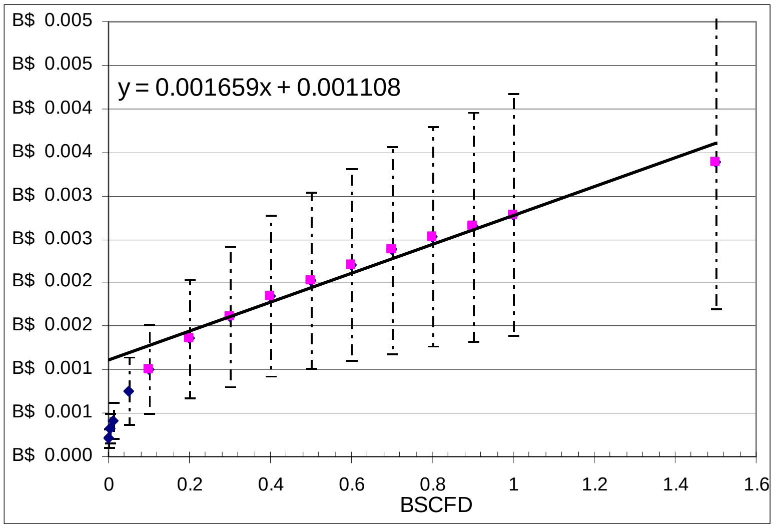



**Figure 1-13:** *Pipeline fixed cost (b$/mile) vs. capacity (BSCFD)*

A linear correlation gives the following form:

$$FCI(B\$/mile) = 0.001659 * Capacity(BSCFD) + 0.001108$$

Operating costs for pipelines were estimated as follows; an average of 5 operators is assumed to be the requirement for each compression station, with an hourly wage of \$21. Direct supervisory and clerical labor is assumed to be 20% of operating labor. Compressor fuel requirement is estimated at 8,000 Btu / BHP-HR, and fuel cost at \$2.5 per million Btu. Maintenance cost is assumed to be 7% of the FCI for compressors and 3% for pipes while insurance is 1% for compressors and 0.4% for pipes. Operating cost per pipeline mile versus capacity is plotted in the next figure:

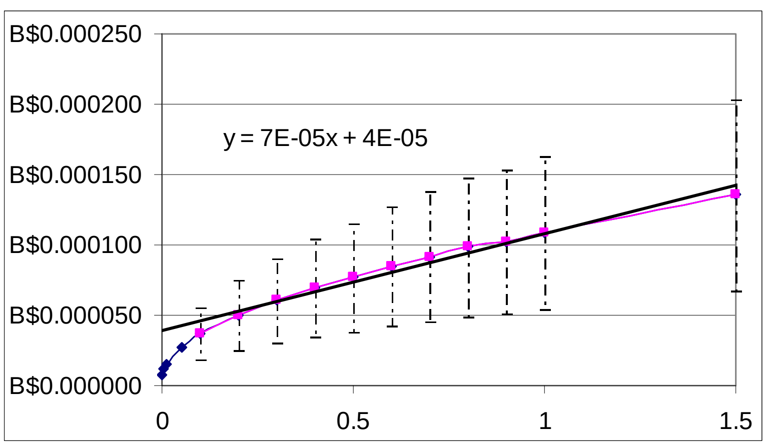



**Figure 1-14:** *Pipeline per mile annual operating cost vs. capacity*

This estimate should be reasonable with about 40% accuracy. Similar linear approximation to that of the FCI is assumed. Linear regression was used to estimate the operating cost dependence of the capacity, ignoring capacities less than 100 MMSCFD. This gave a general correlation of the following form:

$$Oper.Cost(B\$/mile/year) = 0.00007 * Capacity(BSCFD) + 0.00004$$

**Exercise 1-9:**
Consider the pipeline of Exercise 1-8:
- Vary the pipeline maximum pressure (1200 psia) to some lower and higher value. Adjust the diameter accordingly and calculate the number of recompression stations.
- Calculate the cost. Can you say that 1200 psia is the right pressure?

# Pipeline Looping

Pipeline looping is the practice of designing pipelines with segments run in parallel. This practice increases the pipeline flow capacity without altering the final pressure. If temperature is close to ambient temperature, the location of a loop does not change the final delivery pressure. However, when temperature changes substantially, then the location of a loop has an influence. Thus, in these cases, for example, it is recommended to loop in the upstream region, where the gas is hotter. This allows the gas to cool down faster and therefore increase the delivery pressure.

Consider the following example: A 100 Km length (20" OD) pipeline is used to send 289 MMSFD at an inlet pressure of 1,200 psia and a temperature of 45 $^oC$. The pipe roughness is 750 μ inches, and a soil temperature of 10 $^oC$.

Three alternatives were studied for this pipeline. a) No looping, b) Looping the first 25 Km, and c) Looping the last 25 Km. The results of a simulation are shown in the next figure:

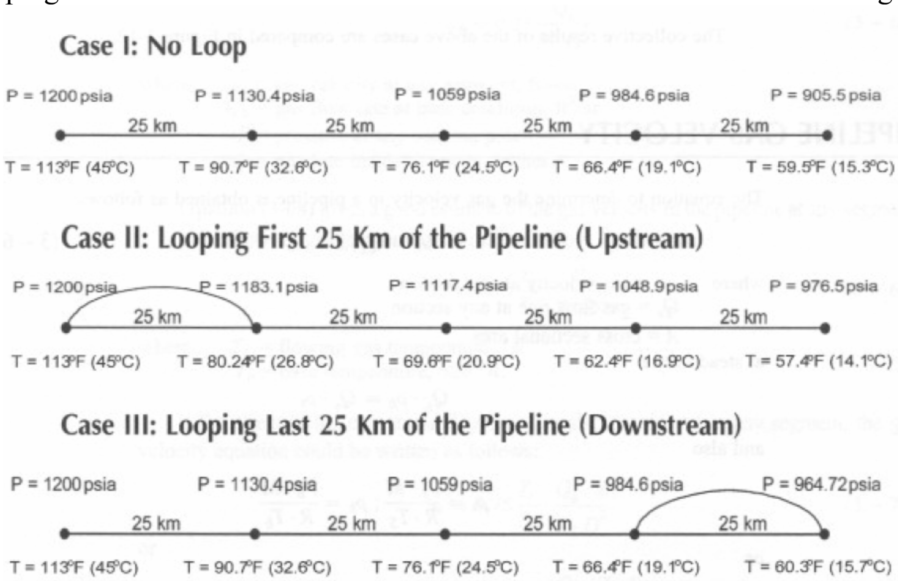



**Figure 1-15:** *Results from Looping*

**Exercise 1-10:**
Verify the results of figure 1-15 using the simulator.

# Retrograde condensation

One very common phenomenon in pipelines is retrograde condensation. Consider the P-T plot of the next figure. It corresponds to a gas with the following composition: Methane: 93.47 %, Ethane: 3.52%, Propane: 0.585%, n-butane: 0.16%, i-butane: 0.11%, pentane: 0.055%, i-pentane 0.05%, hexane: 0.09%, heptane: 0.04%, octane: 0.03%, nonane: 0.01%, $CO_2$: 0.0545, $N_2$: 1.34%. Assume a 15", 200 Km pipeline starts at 60 atm and 15 $^oC$. If the external temperature is 5 $^oC$ (U=1 BTU/hr-ft$^{2o}$F), then it is clear that there will be liquid formation in this pipeline, even if the operation is isothermal.

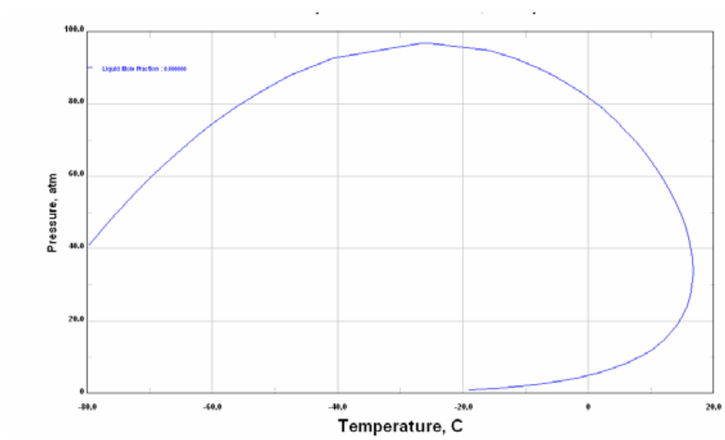



**Figure 1-16:** *P-T diagram of example gas and retrograde condensation*

Interestingly, if the pressure at the other end is low enough, then the liquid might vaporize again. This means that the pressure drop regime inside the pipe might change and one has to be careful in performing the simulations.

**Exercise 1-11:**
Generate the answers for the above example using the simulator. Change the pipe diameter and the length to verify the statements.

# Pipeline Optimization Process

## J-Curve Analysis

Conventional Pipeline design methods, which rely mostly on hand calculations or at best on simple spreadsheet suggest that the compressor size and the pipe diameter be varied and the cost of service ($/(m$^3$*Km) for the first year be plotted as a function of flowrate. Typical assumptions are that there is no volume buildup in the pipe, the time value of money is neglected and that the facilities are designed to sustain the flows.

For example, Figure 1-17 shows one such exercise performed for three different diameters and parametric at different maximum operating pressures (MOP) and compression ratios.
Efficient operating ranges that are flat are preferred

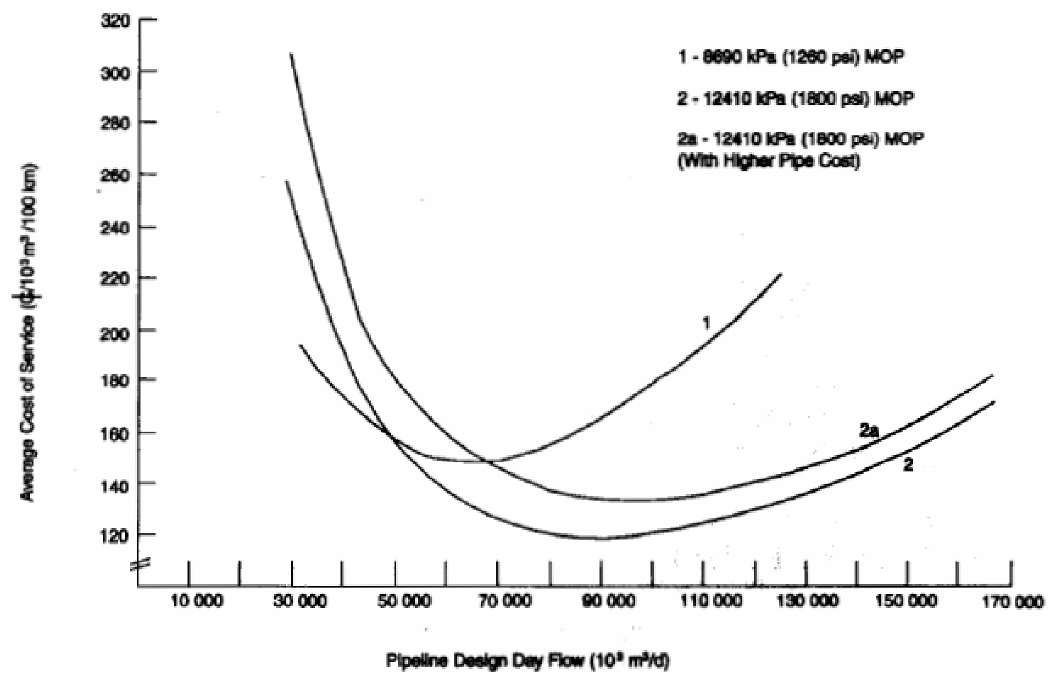



**Figure 1-17:** *J-Curves for various diameters*

**Exercise 1-12:**
Explain why *J*-curves go through a minimum.

## Optimization Parameters

J-Curves are a simplistic first approach but one that can provide a first approximation to the right diameter and compressors. Thus one needs to establish

- **Route:** In most cases this is defined by a variety of other factors and given to the designer.
- **Pipeline Initial Capacity:** Most pipelines are constructed taking into account the fact that demand at the receiving end(s) will increase through time. Thus, one is faced with the decision of designing for future capacity and underutilize the pipeline for some time or design for current or more short term capacity and use loops to expand later.
- **Expansions:** If capacity expansions are considered, then they need to take place through looping. Not only the new loop has to be designed, but its timing and capacity be selected.
- **Maximum operating pressure:** This choice has already been considered in constructing the J-Curves. However, in more complex situations, one is faced with multiple delivery points with different delivery pressures, etc.
- **Pipe Size:** This choice has already been considered in the J-Curve selection but needs to be revisited anyway in view of the influence of the other factors.
- **Load Factor:** This factor is the ratio between the average daily volume delivered divided by the peak volume. If this ratio is too small, then storage facilities for inventory holding (salt caverns, underground caverns, abandoned reservoirs, etc if available, or large LNG or high pressure, CNG, storage tanks) are more convenient than more powerful compressors and larger diameters. The issue to resolve is when these are substituted by inventory holding sites. In addition, the question remains where these holding sites should be located.
- **Compressor Station Spacing:** While an earlier exercise suggests that when multiple compressors are used it is best to keep the compression ratio equal, this hypothesis needs verification. Reciprocating compressors are chosen when the power requirement is smaller than 5,500 Kw. Compression ratios recommended for centrifugal compressors are generally in the range of 1.25 to 1.35 and smaller than 1.5 for reciprocating compressors.

### Exercise 1-13:

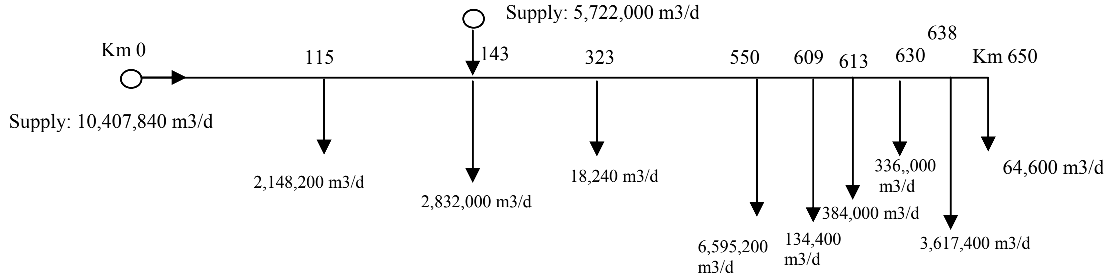
Consider the pipeline of Exercise 1-8:

- Assume two compression stations will be used.
- Determine (by inspection and using a simulator what is the best compression ratio for each compressor. Aim at minimizing total work only.

- **Heating and Cooling:** Clearly, cooling leads to significant savings because pressure drop is reduced. However, money needs to be spent to install and run the coolers. Thus the trade-off needs to be resolved.

**Exercise 1-14:**

Consider the shown in the following figure




- The piping is in the ground and is not insulated. Assume a ground temperature of 25$^o$C and a ground conductivity of 0.7 W/(m $^o$C). The gas elevation profiles are provided in the following table:

| Km | Elevation (m) |
|---|---|
| 0 | 42 |
| 115 | 7 |
| 143 | 14.93 |
| 323 | 60 |
| 550 | 10 |
| 609 | 120 |
| 613 | 122 |
| 630 | 235 |
| 638 | 470 |
| 650 | 890 |

- The gas ((1.9% methane, 5% Ethane, 2% propane, 1% n-butane and 0.1% n-pentane) is supplied at the two points indicated in the diagram at 1,367 kPa and 35$^o$C in the first station (Km 0), and 1520 kPa and 30$^o$C in the second (Km 143).
- Determine using simulations a) Piping diameter, b) Compressors at the supply station, c) cooling required. Do not use a pressure above 5,600 Kpa. Use cost data provided above.
- Will new compressors be needed/beneficial?

# OPTIMAL DESIGN

A gas-gathering and transmission system consists of sources of gas, arcs composed of pipeline segments, compressor stations, and delivery sites. The design or expansion of a gas pipeline transmission system involves capital expenditures as well as the continuing cost of operation and maintenance. Many factors have to be considered, including

1. The maximum number of compressor stations that would ever be required during a specified time horizon
2. The optimal locations of these compressor stations
3. The initial construction dates of the stations
4. The optimal solution for the expansion for the compressor stations
5. The optimal diameter sizes of the main pipes for each arc of the network
6. The minimum recommended thickness of the main pipes
7. The optimal diameter sizes, thicknesses, and lengths of any required parallel pipe loops on each arc of the network
8. The timing of constructions of the parallel pipe loops
9. The operating pressures of the compressors and the gas in the pipelines

In this example we describe the solution of a simplified problem so that the various factors involved are clear. Suppose that a gas pipeline is to be designed so that it transports a prespecified quantity of gas per time from point *A* to other points. Both the initial state (pressure, temperature, composition) at *A* and final states of the gas are known. We need to determine.

1. The number of compressor stations
2. The lengths of pipeline segments between compressor stations
3. The diameters of the pipeline segments
4. The suction and discharge pressures at each station.

The criterion for the design will be the minimum total cost of operation per year including capital, operation, and maintenance costs. Note that the problem considered here does *not* fix the number of compressor stations, the pipeline lengths, the diameters of pipe between stations, the location of branching points, nor limit the configuration (branches) of the system so that the design problem has to be formulated as a nonlinear integer programming problem. Figure E13.4a illustrates a simplified pipeline that we use in defining and solving the problem.

Before presenting the details of the design problem, we need to distinguish between two related problem, one being of a higher degree of difficulty than the other. If the capital costs of the compressors are a linear function of horsepower as shown in line *A* in Figure E13.4b, the transmission line problem can be solved as a nonlinear programming problem by one of the methods discussed in Chapter 8. On the other

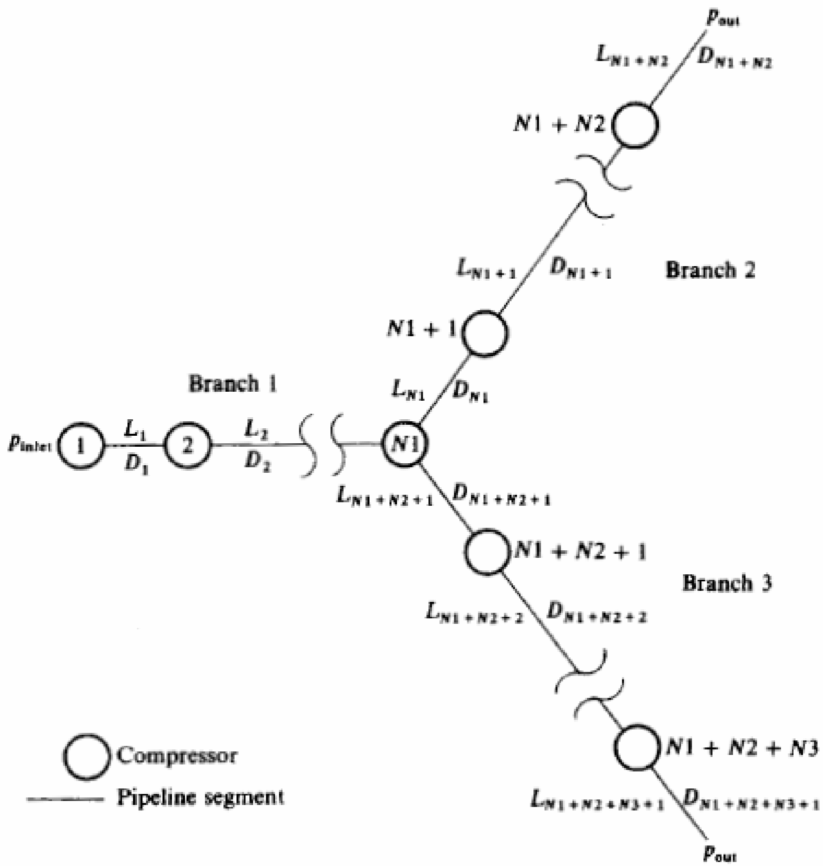



**FIGURE E13.4a**
Pipeline configuration with three branches.

hand, if the capital costs are a linear function of horsepower with a fixed capital outlay for zero horsepower as indicated by line *B* in Figure E13.4b, a condition that more properly reflects the real world, then the design problem becomes more difficult to solve and must be solved by a branch-and-bound algorithm combined with a nonlinear programming algorithm as discussed later on. The reason why the branch-and-bound method is avoided for the case involving line *A* is best examined after the mathematical formulation of the objective function (cost function) has been completed. We split the discussion of the transmission line problem into five parts: (1) the pipeline configuration, (2) the variables, (3) the objective function and costs, (4) the inequality constraints, and (5) the equality constraints.

**The pipeline configuration**. Figure E13.4a shows the configuration of the pipeline we are using in this example and the notation employed for the numbering system for the compressor stations and the pipeline segments. Each compressor station is represented by a node and each pipeline segment by an arc. N1, N2, and N3 represent the maximum number of possible stations in each of the three branches. Pressure increases at a compressor and decreases along the pipeline segment. The transmission

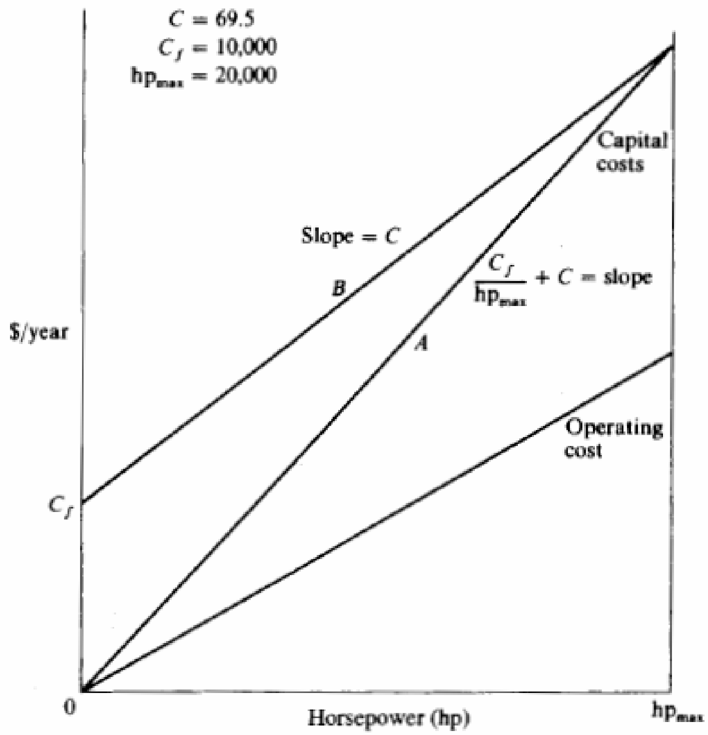



**FIGURE E13.4b**
Capital and operating costs of compressors.

system is presumed to be horizontal. Although a simple example has been selected to illustrate a transmission system, a much more complicated network can be accommodated that includes various branches and loops at the cost of additional computation time. For a given pipeline configuration each node and each arc are labeled separately. In total there are

- $n$ total compressors $[n = \sum(N_i)]$
- $n - 1$ suction pressures (the initial entering pressure is known)
- $n$ discharge pressures
- $n + 1$ pipeline segment lengths and diameters (note there are two segments issuing at the branch)

**The variables.** Each pipeline segment has associated with it five variables: (1) the flow rate $Q$; (2) the inlet pressure $p_d$ (discharge pressure from the upstream compressor); (3) the outlet pressure $p_s$ (suction pressure of the downstream compressor), (4) the pipe diameter $D$, and (5) the pipeline segment length $L$. Inasmuch as the mass flow rate is fixed, and each compressor is assumed to have gas consumed for operation of one-half of one percent of the gas transmitted, only the last four variables need to be determined for each segment.

**The objective function.** Because the problem is posed as a minimum cost problem, the objective function is the sum of the yearly operating and maintenance costs of the compressors plus the sum of the discounted (over 10 years) capital costs of the pipeline segments and compressors. Each compressor is assumed to be adiabatic with an inlet temperature equal to that of the surroundings. A long pipeline segment is assumed so that by the time gas reaches the next compressor it returns to the ambient temperature. The annualized capital costs for each pipeline segment depend on pipe diameter and length, but are assumed to be \$870/(in.)(mile)(year). The rate of work of one compressor is

$$W = (0.08531)Q\frac{k}{k-1}T_1\left[\left(\frac{p_d}{p_s}\right)^{z(k-1)/k} - 1\right] \qquad (a)$$

where $k = C_p/C_v$ for gas at suction conditions (assumed to be 1.26)
$z$ = compressibility factor of gas at suction conditions ($z$ ranges from 0.88 to 0.92)
$p_s$ = suction pressure, psi
$p_d$ = discharge pressure, psi
$T_1$ = suction temperature, °R (assumed 520°R)
$Q$ = flow rate into the compressor, MMCFD (million cubic feet per day)
$W$ = rate of work, horsepower.

Operation and maintenance charges per year can be related directly to horsepower and are estimated to be between 8.00 and 14.0 \$/(hp)(year), hence the total operating costs are assumed to be a linear function of compressor horsepower.

Figure E13.4b shows two different forms for the annualized capital cost of the compressors. Line $A$ indicates the cost is a linear function of horsepower [\$70.00/(hp)(year)] with the line passing through the origin, whereas line $B$ assumes a linear function of horsepower with a fixed initial capital outlay [\$70.00/(hp)(year) + \$10,000] to take into account installation costs, foundation, and so on. For line $A$, the objective function in dollars per year for the example problem is

$$f = \sum_{i=1}^{n}(C_0 + C_c)Q_i(0.08531)T_1\left(\frac{k}{k-1}\right)\left[\left(\frac{p_{d_i}}{p_{s_i}}\right)^{z(k-1)/k} - 1\right] + \sum_{j=1}^{m} C_s L_j D_j \qquad (b)$$

where $n$ = number of compressors in the system
$m$ = number of pipeline segments in the system ($= n + 1$)
$C_0$ = yearly operating cost \$/(hp)(year)
$C_c$ = compressor capital cost \$/(hp)(year)
$C_s$ = pipe capital cost \$/(in)(mile)(year)
$L_j$ = length of pipeline segment $j$, mile
$D_j$ = diameter of pipeline segment $j$, in.

You can now see why for line $A$ a branch-and-bound technique is not required to solve the design problem. Because of the way the objective function is formulated, if the ratio $(p_d/p_s) = 1$, the term involving compressor $i$ vanishes from the first summation in the objective function. This outcome is equivalent to the deletion of compressor $i$ in the execution of a branch-and-bound strategy. (Of course the pipeline segments joined at node $i$ may be of different diameters.) But when line $B$ represents the compressor costs, the fixed incremental cost for each compressor in the system at zero horsepower ($C_f$) is *not* multiplied by the term in the square brackets of Equation ($b$). Instead, $C_f$ is added in the sum of the costs whether or not compressor $i$ is in the system, and a nonlinear programming technique cannot be used alone. Hence, if line $B$ applies, a different solution procedure is required.

**The inequality constraints.** The operation of each compressor is constrained so that the discharge pressure is greater than or equal to the suction pressure

$$\frac{p_{d_i}}{p_{s_i}} \leq 1, \quad i = 1, 2, \ldots, n \tag{c}$$

and the compression ratio does not exceed some prespecified maximum limit $K$

$$\frac{p_{d_i}}{p_{s_i}} \geq K_i, \; i = 1, 2, \ldots, n \tag{d}$$

In addition, upper and lower bounds are placed on each of the four variables

$$p_{d_i}^{\min} \leq p_{d_i} \leq p_{d_i}^{\max} \tag{e}$$

$$p_{s_i}^{\min} \leq p_{s_i} \leq p_{s_i}^{\max} \tag{f}$$

$$L_i^{\min} \leq L_i \leq L_i^{\max} \tag{g}$$

$$D_i^{\min} \leq D_i \leq D_i^{\max} \tag{h}$$

**The equality constraints.** Two classes of equality constraints exist for the transmission system. First, the length of the system is fixed. With two branches, there are two constraints

$$\sum_{j=1}^{N1-1} L_j + \sum_{j=N1}^{N1+N2} L_j = L_1^*$$

$$\sum_{j=1}^{N1-1} L_j + \sum_{j=N1+N2+1}^{1N+N2+N3+1} = L_2^* \tag{i}$$

where $L_k^*$ represents the length of a branch. Second, the flow equation, the Weymouth relation (GPSA handbook, 1972), must hold in each pipeline segment

$$Q_j = 871 D_j^{8/3}\left[\frac{p_d^2 - p_s^2}{L_j}\right]^{1/2} \tag{j}$$

where $Q_j$ = a fixed number
$p_d$ = the discharge pressure at the entrance of the segment
$p_s$ = the suction pressure at the exit of the segment

To avoid problems in taking square roots, Equation ($j$) is squared to yield

$$(871)^2 D_j^{16/3}(p_d^2 - p_s^2) - L_j Q_j^2 = 0 \tag{k}$$