FACTORS AFFECTING THE SIZING OF GAS RANGES

By

W. Blejde[1]

ABSTRACT

Existing fluid flow techniques may be readily applied to the design of ranges for seam gas drainage. The Reynolds number characterises the flow regime within the pipe and it can be used in conjunction with data on absolute pipe surface roughness to determine the system friction factor.

The application of the solution to the isothermal compressible flow equation utilising Moody friction factors and the concept of equivalent length of pipe for fittings and valves provides a practical method for sizing gas ranges.

The physical capacity of a range can be decreased by the existence of significant quantities of water in the range and also by the ingress of air into the system at poor pipe joints etc. Both problems can be eliminated by adequate attention to design detail.

NOMENCLATURE

A	Area (m^2)
C_1	Constant
C_D	Discharge coefficient
d	Orifice diameter (m)
D	Pipe diameter (m)
e	Absolute pipe roughness (m)
f	Friction factor
G	Mass velocity (Kg/sec)
gc	Dimensional constant
h	Head (m)
i	Hydraulic gradient (h/l)
L,l	Pipe length (m)
M	Molecular weight
P_1, P_2	Absolute pressure (Kpa)
R_H	Hydraulic radius (m)
$\bar{R}$	Gas constant
R	Shear stress (Kg/m sec^2)
T	Absolute temperature (^oK)
u	Pipe flow velocity (m/sec)
V	Specific volume (m^3/Kg)
ρ	Density (Kg/m^3)
γ	Specific heat ratio (Cp/Cv)
μ	Viscosity

1 Supervising Process Engineer, Australian Iron & Steel Pty Ltd

REYNOLDS NUMBER

The initial experimental work concerned with the determination of pressure drop due to flow through a pipe was carried out by Reynolds (1881), and expanded by Stanton and Pannell (1914). The results from these workers is usually conveniently summarised in the form of Figure 1.

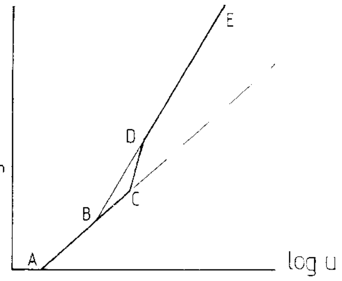


Fig.1 - The effect of fluid velocity on the hydraulic gradient

At low velocities (region A-B) the hydraulic gradient is directly proportional to the fluid velocity and the slope of the log (hf/l) vs log u relationship is unity. As the velocity within a pipe is increased, an area of instability is reached (B-D) in which the experimental points are poorly defined. A further increase in fluid velocity (region D-E) results in the establishment of another flow regime in which the pressure drop increases more rapidly and the slope of the relationship becomes 1.8. This is known as the turbulent flow regime.

The extrapolation of D-E back onto A-C defines the point B which is known as the critical velocity and denotes the velocity at which the flow characteristic changes from a streamline character. The region BD denotes the transition zone between streamline and turbulent flow.

Experimental work carried out by Reynolds has shown that the critical velocity is inversely proportional to pipe diameter and directly proportional to fluid viscosity.

The Aus.I.M.M. Illawarra Branch Symposium,
"Seam Gas Drainage with particular reference to the Working Seam", May 1982

Reynolds was able to characterise flow by defining a dimensionless group which is the ratio of the fluid momentum (or inertial) force to the viscous force.

This dimensionless number is known as Reynolds number and is defined by the following equation:

$$\text{Reynolds Number } (N_{Re}) = \frac{u\rho D}{\mu} \qquad 1$$

It was found that for $N_{Re} < 2000$ the flow is usually streamline whereas for $N_{Re} > 4000$ where inertial forces have a much larger effect than viscous forces, the flow is turbulent.

Friction Factor - Definition

If the flow along a pipe of diameter 'D', length 'dl' is considered, it can be readily seen that 'R' the shear stress at the wall of the pipe will present a total frictional force 'F_f' defined by the following equation number two.

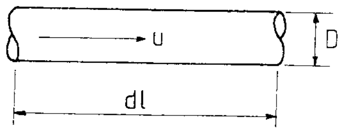


Figure 2

$$F_f = R\pi\ Ddl \qquad 2$$

This frictional force will cause a pressure drop dP_f. A simple force balance yields that:

$$R\pi Ddl = -dP_f \pi \frac{D^2}{4}$$

$$\therefore -dP_f = 4\frac{(R)}{(\rho u^2)}\frac{dl}{D}\rho u^2 \qquad 3$$

If the fluid is considered incompressible, the velocity of the fluid 'u' is not a function of 'P' or 'l' and equation number three can be integrated directly to give the following equations:

$$\Delta P_f = 4\frac{(R)}{(\rho u^2)}\frac{l\rho}{D}u^2 \qquad 4a$$

$$\Delta h_f = 4\frac{(R)}{(\rho u^2)}\frac{lu^2}{gD} \qquad 4b$$

The concept of fluid head 'h' is often used interchangeably with pressure in engineering circles hence expressions for pressure drop and head loss have been derived.

Friction Factor-Experimental Data

Stanton and Pannell (1914) determined the relationship between pressure drop for a number of fluids as a function of Reynolds number.

They presented their results in the form of a graph of $R/\rho u^2$ against N_{Re} and it was found that for a given pipe material (ie, constant surface roughness), the results for all fluids, pipe diameters and velocities fall on a single curve. This curve exhibited the same break points observed by Reynolds.

The Work by Stanton and Pannell was extended by Moody (1944) who correlated the quantity 'f' (which is equivalent to $8\ R/\rho u^2$) in terms of Reynolds number and the relative pipe roughness 'e/D' where 'e' is a linear quantity representing the surface roughness of the pipe. These results are summarised in Figure 3.

Figure 3 may be divided into four separate regions:

i. $N_{Re} < 2000$

This represents the streamline flow region in which all of the data is represented by a single curve and where friction factor is independent of pipe roughness.

This region is characterised by the equation:

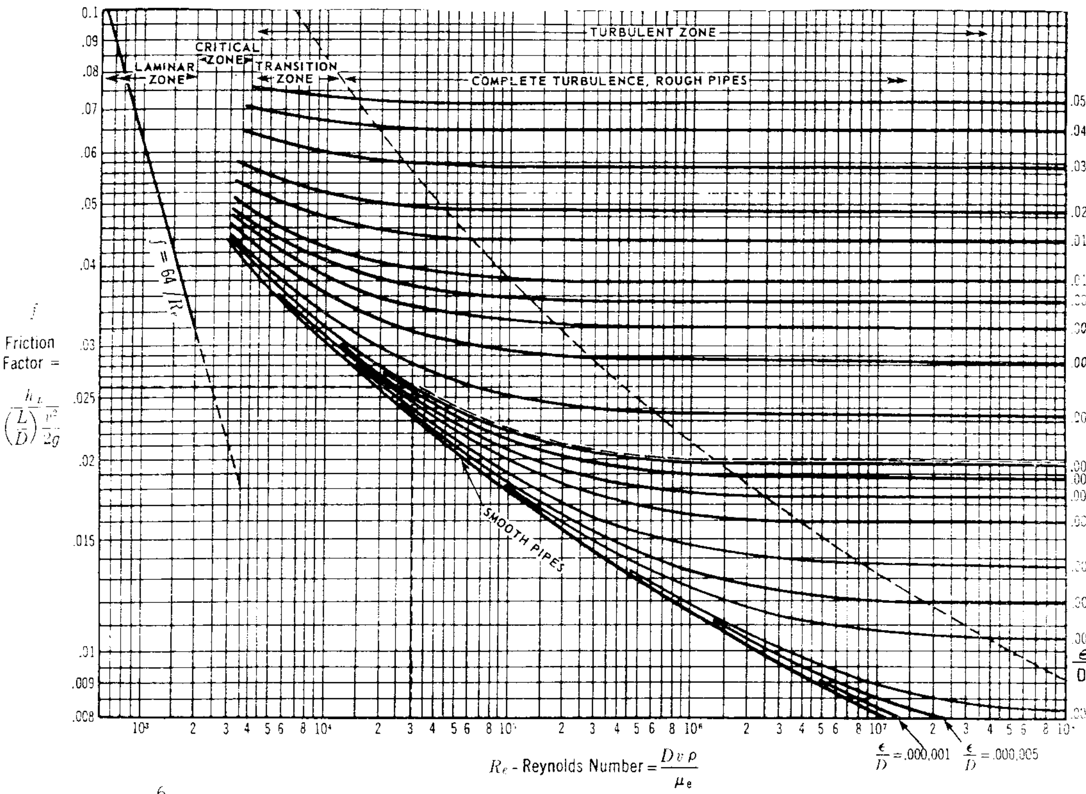
$$f = \frac{8R}{\rho u^2} = \frac{64}{N_{Re}} \qquad 5$$

ii. $2000 < N_{Re} < 4000$

This region is the unstable transition flow regime between streamline and turbulent flow.

iii. $N_{Re} > 4000$

This represents the established turbulent flow regime where the friction factor is a function of Reynolds number and relative surface roughness.



iv. $N_{Re} > 10^6$

This is the highly turbulent flow regime where the friction factor is independent of Reynolds number and is a function of relative surface roughness only.

Fig.3 - The Effect of Reynolds Number & Relative Roughness on Friction Factor

Technical data books make reference to a number of friction factors including the Fanning friction factor and the Darcy friction factor which are equal to $2(R/\rho u^2)$, $R/\rho u^2$ and even $8R/\rho u^2$ (Moody friction factor).

From Figure 3 it is evident that the estimation of surface roughness is critical only at very high Reynolds numbers. The values of absolute roughness have been determined for a number of materials and the values are presented in Table 1.

	e (ft)	(m)
Drawn Tubing	.000005	1.52×10^{-6}
Commerc.Steel	.00015	4.572×10^{-5}
Galv.Iron	.0005	1.52×10^{-4}
Cast Iron	.00085	2.591×10^{-4}
Concrete	.001-.01	3.05×10^{-4}

Table 1 - Values of Absolute Roughness for Selected Pipe Surface Materials

When the value of the absolute roughness of a surface is not known, it may be deduced from a pressure drop/flow measurement for the same material in another installation.

The effect of pipe surface deterioration through corrosion will be very significant and may increase absolute roughness by up to ten times the normally accepted value.

Non Circular Ducts

The equations considered so far have been developed for flow through a circular cross section in a conventional pipe. In the case of ducts of non circular cross section, a quantity known as the hydraulic diameter DH replaces the pipe diameter D. The hydraulic diameter is defined as four times the cross sectional area divided by the wetted perimeter.

The concept is applicable to turbulent flow only and for stream line flow exact expressions relating pressure drop to velocity for ducts of different shapes have to be obtained.

Pipe Fittings

Many experiments have been carried out to determine the pressure loss due to flow as a function of velocity and for practical purposes it has been shown that for turbulent flow, the pressure loss is proportional to the square of the velocity.

The great diversity of valve and fitting types has made it virtually impossible to obtain test data for every size of valve and fitting in use today, however several general methods have been devised to characterise devices. The two most common are the resistance coefficient and the equivalent length methods.

i. Resistance Coefficient

The decrease in static head due to velocity is given by the following equation:

$$\Delta h = \frac{u^2}{2g} \qquad 6a$$

Equation six defines a quantity known as the velocity head. Flow through a valve or fitting in a pipeline causes a reduction in static head which may be expressed in terms of velocity head through use of a resistance coefficient 'K'.

$$\Delta h_L = K \frac{u^2}{2g} \qquad 6b$$

ii. Equivalent Length of Pipe is usually expressed as L/D and is defined as the equivalent length of pipe which will cause the same pressure drop as the valve or pipe fitting under the same flow condition.

The pressure drop in a given system is determined by summing the equivalent length of all of the pipe fittings and valves to the total pipe length and substituting this new length quantity into equation 4a. Typical data for some common fittings is presented in Table 2.

	Le	K
45^o elbow	15D	.3
90^o (Standard radius	35D	.7
90^o (square elbow)	60D	1.2
Gate Valve (fully open)	7D	0.15
Gate Valve (¾ open)	40D	1.0
Gate Valve (½ open)	200D	4.0
Gate Valve (¼ open)	800D	16.0

TABLE 2 - Values of equivalent length and resistance coefficient for selected pipe fittings.

More comprehensive data for pipe fittings including sudden contractions, enlargements, globe valves etc, may be obtained from Crane (1965).

Compressible Flow

If the pressure drop along a pipe ($P_1 - P_2$) is less than ten percent of the initial pressure P_1, the assumption of incompressibility can be made and equations 4a and 4b may be applied directly, however if the pressure changes by a significant amount, allowance must be made for density changes which occur as the fluid flows through the pipe. For short insulated lines the flow will be practically adiabatic whereas for longer uninsulated lines the flow will approach isothermal conditions.

The case of isothermal flow has been defined and the most convenient solution to the energy balance equation has been developed by Dodge (1944) and takes the following form:

$$P_1^2 - P_2^2 = \frac{fLG^2\bar{R}T}{4g_c R_H M A^2}\left(.0479 + \frac{0.221 R_H}{fL} \log_{10}\frac{P_1}{P_2}\right) \qquad 7$$

In pipelines of appreciable length, the second term in brackets is negligible.

The kinetic energy of the fluid increases as the pressure head in the pipe falls at the expense of internal energy and hence the temperature tends to fall. The maintenance of isothermal conditions is thus dependent on the transfer of an adequate amount of heat from the surroundings.

It can be readily shown that the required amount of heat is $\Delta \frac{1}{2} u^2$. Thus if the change in kinetic energy is small, the required amount of heat is small and the condition remains essentially isothermal.

The general solution for the adiabatic case is summarised by equations 8a, 8b.

$$[illegible]\ u^2 [illegible] D = (\gamma-1)/2\gamma - P_1/V_1 (A/G)^2 (1-(V_1/V_2)^2) - \frac{\gamma+1}{\gamma} \ln V_2/V_1 \quad 8a$$

$$[illegible]\ V^2 + (\gamma/(\gamma-1)) PV = \mathrm{CONSTANT} \quad 8b$$

Graphical techniques have been developed by Lapple (1943) for both adiabatic and isothermal solutions. His work indicates that for a given line and pressure drop, adiabatic conditions never result in a discharge rate more than 20% higher than that for isothermal conditions.

The greatest discrepency will occur with very short pipes for which isothermal flow cannot be realised practically with a gas, however for long pipes (Le = 1000D) the discharge rate for a given pressure drop is practically the same regardless of whether the flow is isothermal or adiabatic.

Thus for practical design work, equation 7 is adequate for pressure drop calculations for long pipes. If significant elevation changes occur, say greater than one thousand metres, a correction factor for change in potential energy also has to be applied. In the design of a new system it is common to decide on a pipeline velocity, usually around 20m/sec and then check the actual system pressure drop.

One common fluid present in seam gas drainage systems which will have a significant effect on system pressure drop is the existence of water in the liquid phase. The water tends to accumulate in the system low points and consequently reduce the pipe hydraulic diameter and thus increase system pressure drop unless adequate provisions are included for drainage from the range. Water flowing along the bottom of a gas range in the direction opposite to that of the gas stream will produce a greater pressure drop by virtue of the greater shear forces between the gas and liquid surfaces.

Another factor adversely affecting system capacity is the possibility of inleakage of air at poorly sealed pipework joints, inadequate hoses etc. Careful attention to detail should permit elimination of this problem which can reduce the effectiveness of seam gas drainage by virtue of the loss of vacuum pump capacity and increase in system pressure drop due to higher gas mass rates.

The rate of air inleakage into a vacuum system can be readily calculated from compressible flow theory.

Leakage Rate into a Vacuum System

The maximum velocity of a compressible fluid flowing in a pipe or discharging from an orifice is limited by the velocity of propagation of a pressure wave which travels at the speed of sound. Once sonic velocity is reached, any increase in pressure driving force cannot further increase the gas velocity, rather the extra pressure energy is dissipated in shock waves and the jetting of the fluid.

The maximum velocity of a compressible fluid is defined by the following equation:

$$u = \sqrt{\gamma RT} \quad 9$$

The maximum velocity is reached when the ratio of the downstream pressure to upstream pressure reaches the critical pressure ratio defined by:

$$\frac{P_2}{P_1} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma-1}} = 0.53 \text{ for air} \quad 10$$

It can be readily shown that maximum discharge rate for air through an orifice is:

$$G = .475\ A\sqrt{P_1} \quad 11a$$

For pressure ratios greater than the critical pressure ratio, the rate of discharge through an orifice is;

$$G = 14.59\ A\rho_2 \left(1-\left(1-\frac{\Delta P}{101.3}\right)^{0.29}\right)^{\frac{1}{2}} \quad 11b$$

Metering Devices

The most common class of metering device in use for gaseous flows is the head flow meter in which the pressure drop across a suitable restriction is used to measure flow.

The most common devices of this type are the sharp edged orifice plate and the venturi meter. The characterisic equation of these devices is summarised by:

$$M = \frac{C_1\ C_D\ d^2}{\sqrt{1 - (d/D)^4}} \sqrt{\Delta h\rho} \quad 12$$

The orifice plate has associated with it a much higher permanent pressure loss (typically .7h) than is the case with the venturi meter (permanent pressure loss is typically .15h), however this disadvantage is offset by the higher capital cost of the venturi installation.

The design and construction of head flow meters is detailed in BS1042 (1964).

CONCLUSION

1. Existing fluid flow techniques may be readily applied to the design of ranges for seam gas drainage.

2. The Reynolds number permits characterisation of the flow regime. It can be used in conjunction with the relative pipe surface roughness to determine the system friction fact by application of the Moody diagram.

3. The concept of equivalent length of pipe permits the determination of the effect of various pipe fittings and valves on total system resistance.

4. The application of the solution to the isothermal compressible flow equation utilising Moody friction factors and the concept of equivalent length of pipe provides a practical method for sizing gas ranges.

5. The physical capacity of a range can be decreased by the existence of significant quantities of water in the range or the existence of air in-leakage. Both problems can be eliminated with attention to design detail.

LIST OF FIGURES

LIST OF TABLES

ACKNOWLEDGEMENT

The author wishes to thank Australian Iron & Steel Pty Ltd for their consent to take part in this conference and to Dr A Hargraves for his kind invitation and great assistance.

REFERENCES

British Standards Institution, 1964. Methods for the Measurement of Fluid Flow in Pipes, BS1042, Part 1.

Crane Co, 1965 Technical Paper No.410 Flow of Fluids Through Valves, Fittings and Pipe.

Dodge D, 1944 Chemical Engineering Thermodynamics McGraw-Hill, New York.

Lapple C E, 1943 Isothermal and Adiabatic Flow of Compressible Fluids. Trans Am Inst Chem Eng 39: 385.

Moody L F, 1944. Friction Factors for Pipe Flow. Trans Am Soc Mech Eng 66: 671.

Reynolds O, 1881 Papers on Mechanical & Physical Subjects 2, 51.

Stanton T & Pannell J, 1914, Similarity of Motion in Relation to the Surface Friction of Fluids. Phil Trans Roy Soc 214: 199.

DISCUSSION

R. LAMA (Kembla Coal & Coke Pty. Ltd.): It is easy to calculate the size of the pipes if it is assumed that the gas is going to be pure and going to flow with a particular density and at a particular vacuum and all that. But invariably it is found that after certain use, particularly the pipelines that are inbye of the main pipelines, get filled up with water and they get filled up also with mud. Is there any experience available without cutting the pipes and opening them out and clearing them? Some ideas can be formulated about when to clear those pipes, and can this time be picked up either from the changes in the suction or from the horse power requirements, if the pipes have been clogged and something needs to be done?

W. BLEJDE (Australian Iron & Steel): For a given metered flow in a gas drainage system, the required pressure across the pump and across the range can be readily calculated.

A more practical technique is to actually calibrate the system by taking note of the flow readings and the corresponding system suction pressures needed to achieve these flows. If a blockage occurs in the system, the system resistance will increase very rapidly and this will be accompanied by a greater suction pressure needed to deliver the same gas rate. So it is advisable that once a plant is in operation to get that experience and calibrate it.

L. LUNARZEWSKI (Visiting Polish Methane Drainage Specialist to the BHP Steel Division Collieries): Was the effect of main fan suction, and various quantities of water considered in the calculations?

W. BLEJDE: The effect of water will be two-fold. Water present in the bottom of the range acts as a restriction and will reduce the flow area for the gas path and this effect is directly calculable by replacing the pipe diameter by the quantity known as the hydraulic diameter (which is defined as four times the flow area over the wetted perimeter) in fact permits the calculation of the equivalent pipe diameter of the restricted pipe and it can be applied directly to the pressure drop equations. The other effect is due to the water droplets in small pipes leading into the main gas range. The effect of these would be to increase the pressure drop by taking energy away from the fluid to impart kinetic energy to the droplets. The water droplets would be knocked out by the inertial separator effect of the main gas range in which the velocities are considerably lower and hence there would be very few droplets entrained in the main range.

L. LUNARZEWSKI: Yes but quantity of water varies, it depends on time and even on weather and other factors. How is this accounted for in the calculations?

W. BLEJDE: Adequate drainage should be provided and this can be done by selecting all the low points and installing adequate piping and seal pipes to ensure that water does not build up in the range. This may be difficult because of the need to establish a barometric leg which may not be practicable in all parts of the mine. Through sufficient attention to design detail, the range can be kept essentially free of water.