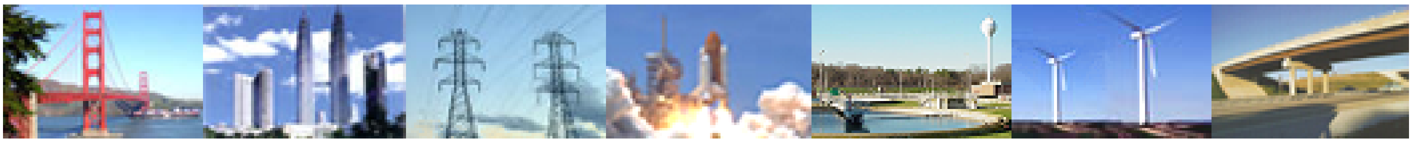

PDHonline Course M270 (12 PDH)

Selecting the Optimum Pipe Size

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Selecting the Optimum Pipe Size

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Randall W. Whitesides, P.E.

Introduction

Pipe, What is It?

Without a doubt, one of the most efficient and natural simple *machines* has to be the pipe. By definition it is a hollow cylinder of metal, wood, or other material, used for the conveyance of water, gas, steam, petroleum, and so forth. The pipe, as a conduit and means to transfer mass from point to point, was not invented, it evolved; the standard circular cross sectional geometry is exhibited even in blood vessels.

Pipe is a ubiquitous product in the industrial, commercial, and residential industries. It is fabricated from a wide variety of materials - steel, copper, cast iron, concrete, and various plastics such as ABS, PVC, CPVC, polyethylene, and polybutylene, among others.

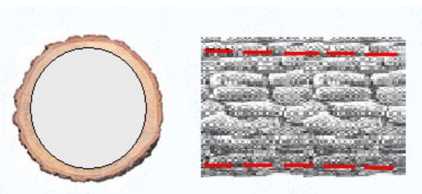
Pipes are identified by *nominal* or *trade* names that are proximately related to the actual diametral dimensions. It is common to identify pipes by inches using NPS or *Nominal Pipe Size*. Fortunately pipe size designation has been standardized. It is fabricated to nominal size with the outside diameter of a given size remaining constant while changing wall thickness is reflected in varying inside diameter. The outside diameter of sizes up to 12 inch NPS are fractionally larger than the stated nominal size. The outside diameter of sizes 14 inch NPS and larger are equal to the stated nominal size. Pipe wall thicknesses are specified by schedule number with 5 being the thinnest and 160 be the thickest. An older designation scheme for pipe wall thickness which still enjoys popular usage indicates nominal weight. This labeling system is depicted in Table 1 on page 2. A more detailed dimensional table for NPS pipe is provided in Table 13 on page 77.

When was It Used?

Clay pipes have been found in excavations dated as early as 4000 BCE. They were used in Mesopotamia, the Indus Valley civilization, the Minoan civilization, and of course the Roman Empire (which also used lead pipes).[1] People have used pipes for thousands of years. Perhaps the first use was by ancient agriculturalists who diverted water from streams and rivers into their fields. Archeological evidence suggests that the Chinese used reed pipe for transporting water to desired locations as early as 2000 BCE.[2]

TABLE 1 Cross Reference of Pipe Wall Thickness Designations

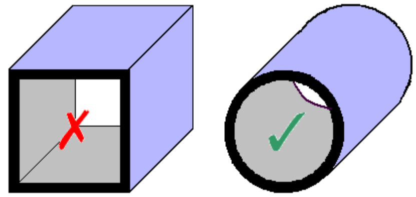
Schedule No.	*Weight Abbreviation*	*Description*
5	—	—
10	LG	Light gauge
—	LW	Light weight
20	—	—
30	—	—
40	ST	Standard weight
60	—	—
80	XS	Extra strong
100	—	—
120	—	—
140	—	—
160	XX	Double extra strong



Early American settlers knew nothing of lead or iron pipe - they knew only to build with wood, the country's bounty. Water pipes were made of bored-out logs, felled from hemlock or elm trees. The trees were cut into seven-to-nine-foot lengths, with trunks being 9-10 inches in diameter. The interior was drilled or bored with steel augers. One end was rammed to form a conical shape, and logs were jammed together in series, using a bituminous-like pitch or tar to seal the joints. Sometimes the logs would be split and hollowed out, put together, and connected with iron hoops. A gravity water system would be set-up, starting from a spring or stream on high ground, allowing water to flow downhill to the house or farm. In the early 1700s, New York, as well as Boston, had constructed a wooden pipe system under the roads, and sold water at street pumps or hydrants. Wooden pipes were common until the early 1800s when the increased pressure required to pump water into rapidly expanding city streets began to split the pipes. A change was made to iron. In 1804, Philadelphia earned the distinction as the first city in the world to adopt cast iron pipe for its water mains. It was also the first city in America to build large scale waterworks as it drew upon the ample supply of the Schuykill River.[3]

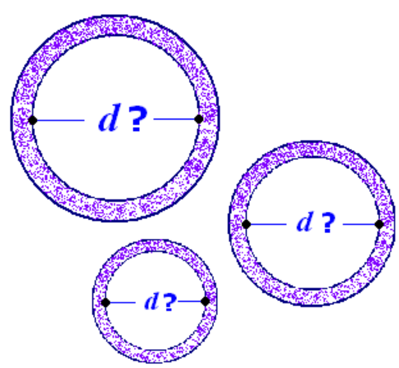
Early pipe size selection was simple. The original pipe fabricators/layers of old were for the most part, not concerned with determining optimum pipe sizes since mere timber availability dictated the diameters installed.

Why is It Round?



Aside from the use of the round tree by our predecessors due to its natural availability, pipe with a circular cross section is preferred over other geometric cross sections for a host of reasons. A quick mathematical examination reveals that a circular cross section results in the least surface area per unit of volume. Surface area is important because it relates directly to the amount of material required to fabricate the conduit, as well as the amount of protective coating or insulation required to cover it. The circular cross section is hydrodynamically more efficient than non-circular sections in that it presents less contact surface area for fluid friction for a given volumetric flow rate. Additionally, a curvilinear profile promotes a smooth flow path devoid of abrupt inside corners or pockets that promote localized fluid stagnation. Structurally, the curved shell produced by the circular pipe cross section allows a multiplicity of alternative stress paths and gives the optimum form for transmission of many different load types. This holds true for both internal loads like pressure and external loads such as those produced by back fill material or internal vacuum conditions. The single curvature allows for a simple fabrication process and is very efficient in resisting loads. Pressures are resisted very well by the in-plane behavior of a shell. Because this pressure difference is across a curved surface, a hoop of circumferential membrane stress is developed in the pipe wall.

The Optimum Pipe Size



Optimum pipe size denotes the best pipe size. From a simplistic standpoint, the best pipe size is obviously the smallest size that will accommodate the application at hand. From a realistic standpoint, optimum pipe size can have many meanings with proper consideration of the application. Optimum can mean economically efficient over the life of a system. Optimum can mean, as in the case of sulfuric acid, the pipe size which would limit the fluid velocity to a value which prevents pipe wall erosion in elbows which ultimately results in structural failure. Optimum can mean, as in the case of fluids with suspended solids, that pipe size which would produce a predetermined fluid velocity which is known to sustain the solids in suspension.

Too often, optimum pipe size is confused to be limited to mean most economic pipe size. Moreover, in addition to meaning satisfactory and *maybe* the most economical, optimum pipe size means that diameter which acts or produces the required effect with a minimum of waste and expense. This size can be far from the most economic due to specific process restraints.

Quick Reference Chart of Useful Pipe Size Equations

To gain immediate benefit from this course, this section provides a summary chart of formulas for determining approximate pipe sizes. The chart is presented at this point, before embarking on a detailed treatment of incompressible versus compressible fluid flow, single versus two phase flow patterns and the like. It offers quick access to formulas to estimate probable or *target* pipe sizes. This listing contains both rational and empirical equations that have evolved through industry's experience. They do not necessarily take into consideration possible mitigating parameters such as erosion, solids suspensions, or slug flow ramifications. These and other important considerations are covered later in the course. The mathematical expressions which follow are combinations of both simplified distillations of rational parent equations as well as *rules-of-thumb*. A list of the nomenclature used in the formulas and equations is provided immediately following the quick reference table.

In the interest of brevity, equation nomenclature and units of measure are not provided separately with each equation. The student is therefore encouraged to frequently review (or better still, print) the nomenclature listing of Table 3.

Units of measure for a specific equation variable are generally constant between equations. The exception is the flow variable *"Q"* which can be either cubic feet per second (ft^3/sec) or gallons per minute (gal/min; gpm) depending on the need for unit consistency within the equation. Students are encouraged to conduct *mental* dimensional analysis when viewing equations and worked examples.

TABLE 2 Quick Reference Summary Chart & Collection of Pipe Size Formulae

(Important: refer to Table 3 Equation Nomenclature and Units)

EXPLANATION	EQUATION	RANGE & LIMITATIONS
Pressurized Flow of Liquids		
Darcy-Weisbach frictional head loss	$d = 12\left(\frac{0.0311\, f\, L\, Q^2}{h_f}\right)^{0.2}$	Q in gal/min.
Liquid general flow equation	$d = 0.64\sqrt{\frac{Q}{V}}$	Q in gal/min.
Nominal pipe size for non-viscous flow	$d = \left(\frac{Q}{1.2}\right)^{1/3} - 2$	$Q > 100$ gal/min
Nominal pipe size for non-viscous flow	$d = 0.25\sqrt{Q}$	$Q \leq 100$ gal/min
Pump suction size to limit frictional head loss	$d = \sqrt{0.0744\, Q}$	Q in gal/min
Pressurized Flow of Gases		
Gas general flow equation	$d = 0.29\sqrt{\frac{Q\, T}{P\, V}}$	Q in standard ft^3/min; T in °R; P in lb/in^2 absolute.
Gas general flow equation	$d = \left[\frac{Q}{8306\sqrt{\frac{P_1^2 - P_2^2}{G\,T\,L\,Z\,f}}}\right]^{0.4}$	Isothermal fully turbulent flow; L in miles; Q in standard ft^3/day.
Minimum diameter to limit erosional gas flow	$d = 0.001\sqrt{Q}\left(\frac{Z\,T\,G}{P}\right)^{0.25}$	Q in standard ft^3/day; T in °R; P in lb/in^2 absolute.
Weymouth gas flow equation	$d = \left[\frac{W^2\, G\, L}{786.8\left(P_1^2 - P_2^2\right)}\right]^{0.1876}$	Isothermal fully turbulent flow; L in miles.
Two-Phase Flow		
Relief valve discharge flashing flow	$d = \left[\frac{W\sqrt{Z\,T/M_w}}{408.245\, M\, P}\right]^{0.5}$	Isothermal flow of an ideal gas for selected Mach Number M.

EXPLANATION	EQUATION	RANGE & LIMITATIONS
Suitable pipe size for gas/liquid flow	$d = 0.2256\sqrt{\dfrac{W_g/\rho_g + W_l/\rho_l}{V}}$	With known two-phase mixture velocity V.
Steam and Vapor		
Steam & vapor general flow equation	$d = 1.75\sqrt{\dfrac{W v_g x}{V}}$	
Gravity Flow		
Manning formula for maximum flow	$d = \left[1525\dfrac{Qn}{\sqrt{S}}\right]^{0.375}$	Based on $y/d = 0.983$; Q in ft^3/sec.
Manning formula for full pipe flow	$d = \left[1639\dfrac{Qn}{\sqrt{S}}\right]^{0.375}$	Based on $y = d$; Q in ft^3/sec.
Minimum opening size of self-venting side entry overflow	$d = 0.92Q^{0.4}$	Q in gal/min; $2 \le d \le 18$; $y/d = 0.75$
Minimum pipe size for self-venting vertical pipe flow	$d = \left(\dfrac{Q}{27.8\,r^{5/3}}\right)^{3/8}$	Where r is ratio of annual flow area to pipe cross sectional area; Q in gal/min.

TABLE 3 Equation Nomenclature and Units

Symbol	***Definition***	***U.S. customary units***
A	Cross sectional flow area	ft^2
A	Exponential correlation constant	Dimensionless
b	Settling velocity constant	Dimensionless
C	Total cost	U.S. $
C	Total life cycle cost present value	U.S. $
C_i	Initial or first cost	U.S. $
C_o	Annual operating cost	U.S. $
c	Mean volume fraction	Decimal percent
D	Inside pipe diameter	ft
D_P	Solid particle diameter	in
d	Inside pipe diameter	in
FU	Fixture Unit	gal/min
f	Colebrook friction factor	Dimensionless

TABLE 3 Equation Nomenclature and Units (continued)

Symbol	*Definition*	*U.S. customary units*
G	Specific gravity	Dimensionless
g	Gravitational constant	ft/sec^2
HP	Energy input to overcome friction	Horsepower
i	Annual cost escalation rate	Decimal percent
F_r	Froude number	Dimensionless
h	Critical static head	in
h_f	Frictional head loss	ft of fluid
K	Settling velocity constant	Dimensionless
L	Length	ft
M	Mach number	Dimensionless
M_w	Molecular weight	lb_m /lb-mol
m	Hindered settling exponent	Dimensionless
N_e	Breakeven time	Years
N_{Re}	Reynolds number	Dimensionless
n	Useful life of project alternative	Years
n	Manning friction factor	sec–ft $^{-1/3}$
n	Settling velocity constant	Dimensionless
P	Pressure	lb_f/in^2
P	Wetted perimeter	ft
P_g	Gas-phase pressure	lb_f/in^2
P_l	Liquid-phase pressure	lb_f/in^2
Q	Volumetric flow rate	ft^3/sec
Q	Volumetric flow rate	gal/min
R	Hydraulic radius	ft
r	Ratio of annular flow areas	Dimensionless
S	Slope (hydraulic gradient)	Dimensionless
T	Saturated steam temperature	° F
T	Gas temperature	° R
t_o	System annual operation time	hr
V	Mean fluid velocity	ft/sec
V	Mean fluid velocity	ft/min
V_S	Single particle settling velocity	ft/sec
V_{SH}	Hindered settling velocity	ft/sec
v	Specific volume	ft^3/lb_m
v_g	Saturated steam specific volume	ft^3/lb_m
W	Weight flow rate	lb_m /hr
W_g	Gas two phase weight flow rate	lb_m /hr
W_l	Liquid two phase weight flow rate	lb_m /hr
x	Steam quality	Decimal fraction

TABLE 3 Equation Nomenclature and Units (continued)

Symbol	*Definition*	*U.S. customary units*
Y_g	Gas compressibility factor	Dimensionless
y	Liquid flow depth	ft
	Greek symbols	
α	Piping system alternative choice	Dimensionless
β	Piping system alternative choice	Dimensionless
Δ	Differential change	Dimensionless
ε	Pipe absolute roughness	ft
η_m	Electric motor efficiency	Decimal percent
η_p	Pump hydraulic efficiency	Decimal percent
μ	Absolute viscosity	cP
υ	Kinematic viscosity	ft^2/sec
ρ	Fluid density	lb_m/ft^3
ρ_F	Mixed phase fluid density	lb_m/ft^3
ρ_P	Solid particle density	lb_m/ft^3
χ	Mole fraction	Decimal fraction

When units other than those listed are used, it will be expressly noted at the time of equation presentation.

Pipe Flow Basics

Flow Regimes

During the 1880s an Irish Engineer and physicist named Osbourne Reynolds conducted experiments by visually monitoring flow patterns in glass tubes with dye injected fluids. His observations resulted in the now famous dimensionless quantity which bears his name, the Reynolds number, N_{Re}. The Reynolds number relates the physical and geometric properties of fluid density, velocity, viscosity, and pipe diameter. His studies showed that in essence three flow regimes exist:

Laminar (smooth) for $N_{Re} < 2{,}000$;
Transitional for $2{,}000 < N_{Re} < 4{,}000$; and
Turbulent for $N_{Re} > 4{,}000$

When colored filaments are injected into a flowing fluid stream, their appearance assumes the configurations shown in the diagram below for each of the three regimes.

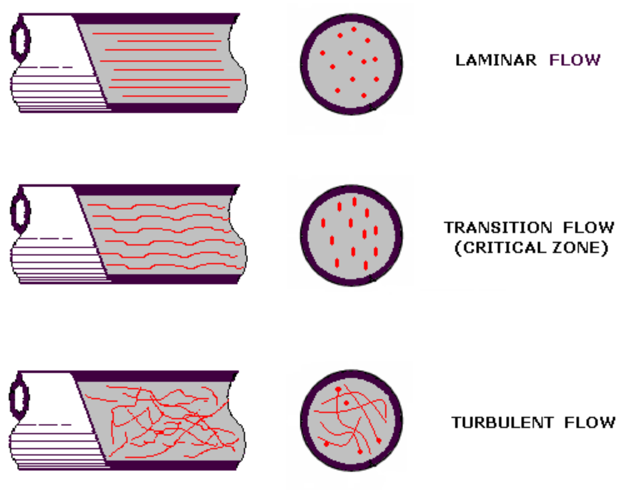


Figure 1 – Flow Regimes

Flow Variables

Part of the difficulty in working with fluid flow problems is the wide array of different variables that come into play. It is helpful to categorize these variables. The fluid dependent variables are:

- Fluid density;
- Fluid viscosity.

The system dependent variables are:

- Surface roughness of pipe interior;
- Length of flow path.

The basic flow parameters are:

- Flow velocity;
- Pipe size;
- Flow rate;
- Pressure loss due to frictional drag.

The fluid dependent variables remain essentially constant for most problems of interest. Similarly, the roughness is constant. Therefore, this leaves pressure drop and the four variables that have been characterized as the basic flow parameters, that must be balanced to select pipe size properly.[4]

Invariably, there are four basic equations which are used in the solution of pressurized fluid flow problems:

- Continuity equation modified for volumetric flow;
- Reynolds number;
- Colebrook friction factor and;
- the Darcy-Weisbach pressure drop.

Each of these will be used throughout the course. If necessary, the student should review PDHcenter.com Course No. M212 for a detailed treatment of the first two items above.

Fluid Friction

It is impossible to study pipe size selection without an understanding of fluid friction and in particular the fluid friction factor *f*. A great deal of technical literature is available, in print and on the Internet, on the subject of computing system friction. Any of this information can be successfully employed depending upon the student's preference, past experience, and applicability to the system and liquid being considered. In all cases, the line losses vary directly as a function of the square of the mean fluid velocity.

The friction factor is a critical variable regardless of the technique employed to determine frictional head loss. It is therefore important to have a background knowledge of its history and development. In 1938 British Engineer C.F. Colebrook demonstrated that for $N_{Re} > 3{,}000$, a fluid friction factor existed which was a function of both the value of N_{Re} and the relative roughness (ε/D) of the pipe. No approximations here; the Colebrook factor is extremely accurate. The Colebrook friction factor is defined as

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

This hideous looking mathematical creature is implicit in *f* which requires an iterative process for convergence to a solution. Going forward, a host of later experimental Engineers developed approximations of this friction factor concept that were explicit in *f*, culminating in a graphical representation of a friction factor relationship produced by L.F. Moody in 1944. A schematic diagram of what has become to be known as the Moody chart is shown in Figure 2. It is a collection of plots of friction factors versus Reynolds number for a variety of relative roughness values (ε/D).

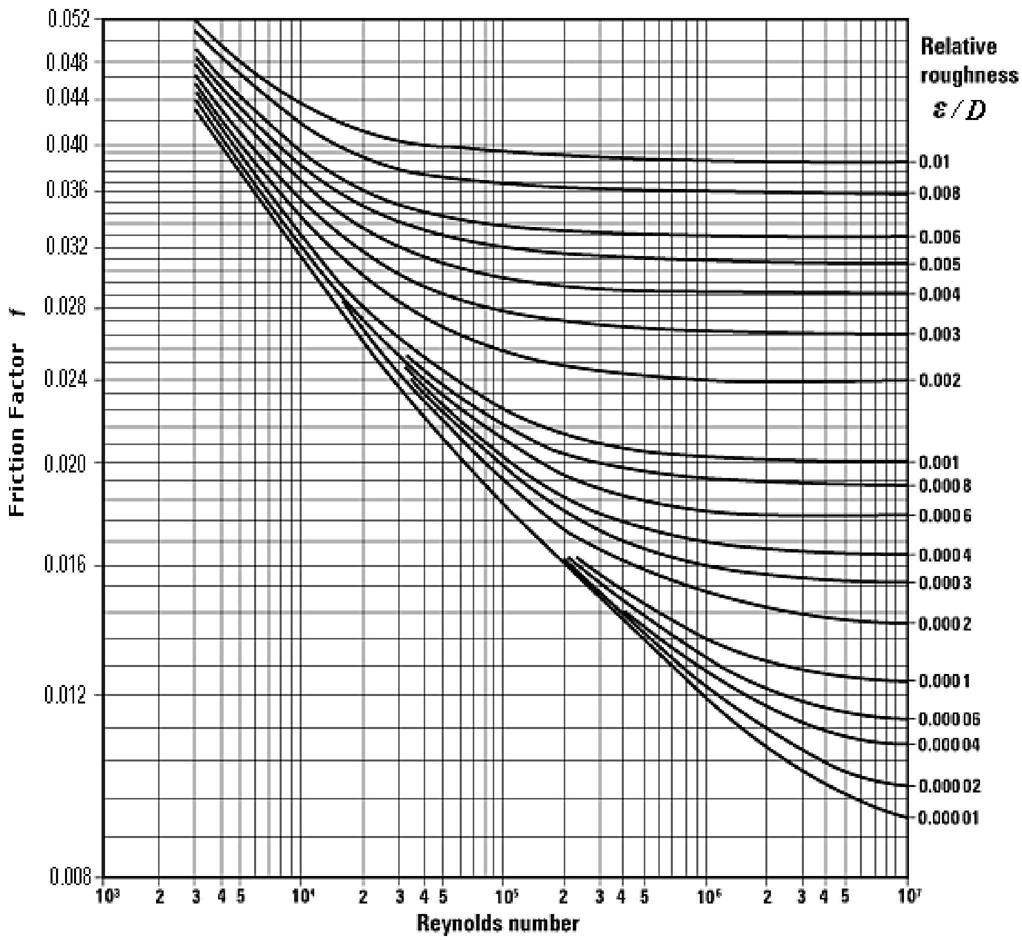


Figure 2 – Moody Chart or Diagram

The Moody chart was used extensively prior to the advent of the programmable calculator and the desktop computer. Present day desktop computing power has eliminated the tedious exercise of manually solving the Colebrook equation by iterative convergence necessitated by the fact that the Colebrook equation is implicit in f.

The drudgery (and resulting approximation) associated with the use of the Moody chart to determine friction factors has been gratefully replaced with any of a number of free applets available on the Internet.

Alternative methods and approximations for determining the value of f have been studied *ad infinitum* and the number of empirical relationships associated with it abound. No attempt will made in this course to list or describe these various formulae or recommend one particular formula over another. For no particular reason, when the need for a value of f is required, the Colebrook value will be referenced.

While the Moody chart provides a quick method of f determination, an extension of a concept derived from the Moody chart can be used to rid us of the burden of evaluating f at each flow rate. The Moody chart f values become asymptotic ($f \approx$ constant) in the zone of complete turbulence. With a close examination of Figure 2, and using water as the fluid, a fully turbulent fluid flow friction factor can be shown to approximately vary with pipe diameter in accordance with Table 4.

TABLE 4 Approximate Asymptotic Moody Friction Factors for Clean Commercial Steel Pipe with Flow in Zone of Complete Turbulence

Nominal Pipe Size (inches)	*Friction Factor (f)*
1	0.023
2	0.019
3	0.018
4	0.017
6	0.015
8-10	0.014
12-16	0.013
18-24	0.012
30-48	0.011

The Fanning friction factor can be cause for much consternation. Some N_{Re} versus ε/D fluid friction charts list values of the Fanning friction factor along the vertical axis in lieu of the Colebrook friction factor. It is important to remember that the Fanning factor = Colebrook ÷ 4.

Frictional Head Loss

If a universal equation exists with regard to the flow of fluids it would have to be the Darcy-Weisbach formula. Within limits, in can be used to analyze both incompressible and compressible fluids. All of its variables, with the exception of f, can be rationally derived.

$$h_f = f \frac{L}{D} \frac{V^2}{2g} \qquad [\ 2\ ]$$

Without derivation, the Darcy-Weisbach energy loss equation can be rewritten in terms of pipe diameter d with Q in gallons per minute, to appear as

$$d = 12 \left(\frac{0.0311 f L Q^2}{h_f} \right)^{0.2} \qquad [\ 3\ ]$$

This form allows for the direct determination of an optimum pipe size.

A Trade-Off

One can always trade-off pipe size against velocity and pressure drop. Greater flow for a given pipe size can be obtained if a higher pressure drop is acceptable. Given the fact that static heads are constant regardless of pipe size, comparable pressurized pumping costs are purely attributable to fluid friction. This leads us to the subject of selecting the most economic pipe size.

Economics

Comparative Analysis

Simply stated, there is no easy approach to determine an economic pipe size for pressurized systems. There have been volumes written on optimum *economic* pipe size. Any method which only looks at flow rate to determine pipe size (or vice versa) cannot cover all the cases. From a purely economic standpoint involving pressurized flow, the optimum pipe size is the diameter that minimizes the life cycle cost of a piping system. The first (installation) cost and the recurring costs associated with operation and maintenance are the components that are examined using the concept of time-value of money. It follows that there is a compromise between large diameter pipe (higher first cost, lower recurring costs) and small diameter pipe (lower first cost, higher recurring costs) for a given service. The only tried and true method involves detailed comparative analysis.

There is no shortage of coverage in the open technical literature on the subject of picking directly the most economic pipe size. Practically all of the methods presented depend on date sensitive cost data which quickly becomes archaic after publication. Simplified methods of determining economic pipe size such as nomographs which incorporate material, energy, and labor costs, are problematic for the same reason.

In order to illustrate fully the concept of hardware versus pumping cost, an example will be provided which introduces the comparative cost method known as break-even analysis.

Break-Even Analysis

Many capital investment alternatives cannot be evaluated by conventional means of worth. Such a case is one in which neither of the presented alternatives can be justified solely on the basis of an inherently derived savings. Break-even analysis becomes useful in such situations. For the same pressurized fluid flow application there is a tradeoff between a large pipe size (higher first cost C_i with lower operating cost C_o) and small pipe size (lower first cost C_i with higher operating cost C_o); consequently, from a total cost standpoint, there exists a point during the life of the project where the attributes of each are equal. This is known as the break-even point. When inherent savings are not applicable, it can be useful to identify the periods that each alternative is economic-ally attractive during the course of the installation's expected life, n. Piping systems often have a life spans of 20+ years.

The concept of break-even analysis uses the following thought process:

The total cost of one alternative, let's call it α, can be expressed as

$$C_\alpha = C_{\alpha i} + n C_{\alpha o} \qquad [\,4\,]$$

The total cost of the second alternative β could be indicated by

$$C_\beta = C_{\beta i} + n\,C_{\beta o} \qquad [\,5\,]$$

To express the relationship in terms of break-even point of these alternatives, let the total cost of each be equal:

$$C_{\alpha i} + n\,C_{\alpha o} = C_{\beta i} + n\,C_{\beta o} \qquad [\,6\,]$$

Rearranging this equation in terms of *n* results in a relationship in which the break-even point (in time) is indicated:

$$n_e = \frac{C_{\alpha i} - C_{\beta i}}{C_{\beta o} - C_{\alpha o}} \qquad [\,7\,]$$

This relationship is shown graphically in Figure 3.

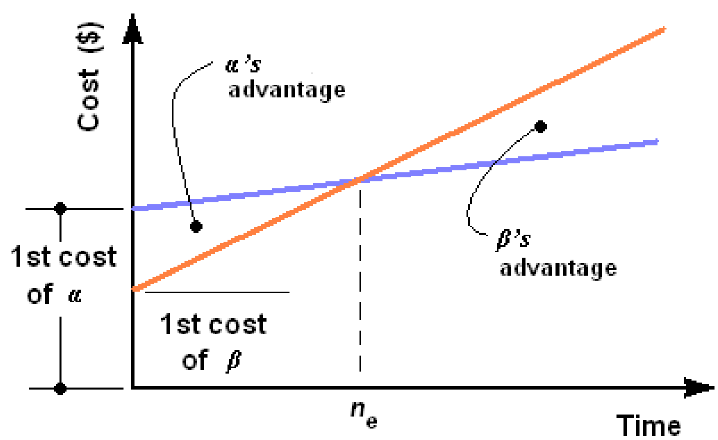


Figure 3 – Break-Even Point of Two Alternative Systems

Studying this diagram shows that the break-even point occurs where the total costs for the individual alternatives are equal. Put another way, the break-even point is where the total cost of piping system β (which represents the lowest first cost but has the higher annual operating expense) has reached the value of the total cost of piping system α (higher first cost but lower annual operating cost). Let us look at an example of break-even analysis for a hypothetical pressurized pipe system which offers two alternatives.

It should be noted that annual operation time can be very critical in the determination of optimum pipe sizes for externally pressurized flow. It is intuitively obvious that operating cost is of little significance where the annual duty cycle (pumping or compressing hours) is very low. In these systems the smallest pipe size otherwise allowable is the most economical and therefore the optimum.

Example 1 Evaluation of Two Alternative Piping Systems

Problem: A choice exists between investing in piping system $\boldsymbol{\alpha}$ or piping system β. Alternative α utilizes large diameter piping with an initial investment of $100,000 and an annual operating energy expense of $20,000. Alternative β employs smaller diameter piping resulting in an initial outlay of only $80,000 but due to the increased pumping cost, is estimated to have an annual operating expense of $25,000. Both systems have been estimated to have a useful life of 15 years. From an economic standpoint, which alternative system should be installed?

Given: $C_{\alpha i}$ = $100,000 ; $C_{\alpha o}$ = $20,000 ; $C_{\beta i}$ = $80,000 ; $C_{\beta o}$ = $25,000 ; n = 15 years.

Find: The break-even point and the most economical alternative system.

Solution:
Using Equation [7] and substituting the known values results in a break-even point of

$$n_e = \frac{C_{\alpha i} - C_{\beta i}}{C_{\beta o} - C_{\alpha o}} = \frac{100{,}000 - 80{,}000}{25{,}000 - 20{,}000} = 4 \text{ years}$$

Conclusion: Since the useful life of both systems is indicated at 15 years, piping system α is decidedly more attractive because of its lower yearly operating energy expense. Were the useful life of the two systems to be less than 4 years, alternative β would be the most attractive.

The present value of the total cost of each alternative in Example 1, using the time value of money, would need to be determined in order to properly evaluate the comparative costs. This leads to the concept of Life Cycle Cost Analysis.

Life Cycle Cost Analysis

The life cycle cost (LCC) of any piping system is the total *lifetime* cost to purchase, install, operate, and maintain that piping system. LCC analysis requires the evaluation of alternative systems based on the present value of these monetary outlays.

Piping system project scale must be considered when allocating time spent in the life cycle cost analysis exercise. It stands to reason that time expended should be proportional to the physical size of the piping system. To wit, the design of an overland gas transmission pipe line would be given heightened life cycle cost analysis emphasis compared to a simple in-plant piping system. A piping system of relatively short length may not warrant any consideration whatsoever in this respect.

Equivalent present value analysis requires the conversion of all future cash outflows (recurring annual costs) to the present. As such, it requires the consideration of time value of money using a term known as the uniform series present value factor. Comparison of the equivalent value of competing alternatives allows the choice of the most desirable alternative on the basis of economics.

When the estimated useful project life is identical for each alternative, the alternative with the lowest present net value should be selected. Recall from engineering economics that the uniform series present value factor is,

$$\frac{(1+i)^n - 1}{i(1+i)^n} \qquad [\,8\,]$$

The annual operation cost variable C_o must be modified by this factor in order to arrive at the total life cycle cost present value:

$$C = C_i + C_o \frac{(1+i)^n - 1}{i(1+i)^n} \qquad [\,9\,]$$

Here is a step-wise approach for determining the most economic pipe size:

1. Knowing the required process flow rate, select a trial pipe size from an appropriate equation listed in Table 2;
2. Using an accurate piping arrangement drawing, as opposed to a piping and instrument diagram (P&ID), determine the exact physical length, including equivalent lengths of the various valves, fittings, and equipment, of the system. Put another way, identify and quantify all components of the system which result in flow resistance;

3. With the given fluid properties, required flow rate, and pipe material and length, calculate the Reynolds number, relative roughness, friction factor, and head loss due to friction;
4. Assume appropriate equipment operating efficiencies. For pumps use maybe 60%; for electric motors use 90%. Determine the required energy input to overcome system fluid friction losses;
5. Based on the stated running hours, compute the system's annual operating energy cost based on the energy input calculated in Step 4;
6. Obtain a manufacturer's price quotation for the pump and driver based on the required flow rate and the calculated head from Step 3;
7. Assume that annual maintenance costs will be 4% of the price obtained in Step 6;
8. Sum the annual costs of energy and maintenance to arrive at the variable C_o, total annual recurring operating cost. (Include any other recurring costs that might be applicable here);
9. From the detailed piping arrangement, estimate the material and labor cost to install the trial pipe size system. Sum this cost with the equipment price quotation of Step 6 to arrive at the variable C_i, the initial capital cost;
10. Use Equation [9] to determine the total life cycle cost present value for the trial pipe size;
11. Repeat Steps 2 through 10 for one nominal pipe size below and one nominal pipe size above the trial pipe size selected in Step 1;
12. Compare the three separate alternative present values and select the pipe size corresponding to lowest cost.

An illustration of this step-wise economic analysis is shown in the worked example which follows. This example is abbreviated and only the details of the initial iteration cycle are presented; however, the results of the entire analysis are summarized in Table 5.

Example 2 Selection of Most Economic Pipe Size for a Given Process Condition

Problem: A clean fluid is to be continuously pumped at a rate of 200 gallons per minute through 960 feet of schedule 40 alloy steel pipe. The fluid has a specific gravity of 0.8 and a viscosity of 1.1 cP at the pumping temperature. There is no flow independent head, *i.e.* static head, in the process. If annual cost escalation is taken as 5%, annual maintenance cost is 4% of capital equipment cost, and energy cost is 7¢ per kwh, what is the most economic pipe size for this operation? The system is expected to have a useful life of 15 years.

Given: t_o = 8,760 hours ; Q = 200 gpm ; L = 960 feet of steel pipe (ε = 0.00015 feet) ; G = 0.8 ; μ = 1.1 cP ; i = 0.05 ; maintenance allowance = 0.04 ; power cost is \$0.07 per kwh ; n = 15 yr.

Find: The most economic pipe size d based on lowest life cycle cost present value of the system.

Methodology: The step-wise cost analysis approach just described will be used. In the interest of brevity, for this example many of the specific calculation details will not be presented.

Solution:

① For Q = 200 gpm, from the nominal pipe size equation in Table 2 determine that d = 3.503 inches. From Table 13 on Page 77 select a 3 inch NPS schedule 40 pipe with d = 3.068 inches as a trial size. Later, for comparison, examine 2½ inch NPS and 4 inch NPS as economic alternatives.

② With the given fluid properties, required flow rate, and pipe material and length, calculate,

$$V = 8.7 \text{ ft/sec}$$
$$N_{Re} = 149{,}694$$
$$\varepsilon/D = 0.00059$$
$$f = 0.01976$$
$$h_f = 87 \text{ feet of fluid}$$

③ Recall that input power to a pump is given by the formula presented below. Using the assumed pump and motor operating efficiencies offered in the course, determine the required horsepower to overcome the system friction in the 3 inch NPS schedule 40 pipe to be,

$$HP = \frac{Q h_f G}{3960\,\eta_p \eta_m} = \frac{(200)(87)(0.8)}{(3960)(0.60)(0.90)} = 6.51 \text{ hp}$$

④ The annual operating cost for this system is,

$$C_o = \frac{(\$0.07)}{\text{kwh}} \cdot \frac{\text{kwh}}{1.341\,\text{hp-hr}} \cdot \frac{6.51\,\text{hp}}{1} \cdot \frac{8{,}760\,\text{hr}}{1} = \$2{,}977 \quad \text{[round-up to \$3,000]}$$

⑤ Obtain a manufacturer's quotation for the pump/motor set. For this iteration (probably 7½ horsepower and 90 feet of head) assume this value to be $13,000. From this equipment cost we can estimate an annual maintenance cost to be (0.04)($13,000) = $520. The value of the recurring annual cost C_o in Step ④ becomes $C_o = \$3{,}000 + \$520 = \$3{,}520$.

⑥ From a detailed piping arrangement of the proposed system, estimate the material and labor cost to install the trial 3 inch NPS pipe system. For this example, assume this cost is $21,000. The initial capital cost is the sum of the piping and equipment cost, or $C_i = \$13{,}000 + \$21{,}000 = \$34{,}000$.

⑦ Total life cycle cost for the 3 inch NPS size system consists of the sum of the first cost and the present net value of the recurring annual costs:

$$C = C_i + C_o \frac{(1+i)^n - 1}{i(1+i)^n} = \$34{,}000 + \$3{,}520 \cdot \frac{(1+0.05)^{15} - 1}{i(1+0.05)^{15}} = \$70{,}540$$

⑧ When Steps ② through ⑦ are repeated for 2½ inch and 4 inch NPS schedule 40 pipe, the summarized data in Table 5 is generated.

TABLE 5 Summary of Economic Variables for Three Alternative Pipe Systems of Example 2

	Nominal Pipe Size (NPS)		
Parameter (variable, units)	***2½ inch***	***3 inch***	***4 inch***
Diameter (*d*, inches)	2.469	3.068	4.026
Mean fluid velocity (*V*, ft/sec)	13.4	8.7	5.0
Reynolds number (N_{Re})	186,012	149,694	114,074
Relative roughness (ε/D)	0.00073	0.00059	0.00045
Colebrook friction factor (*f*)	0.02001	0.01976	0.01975
Head loss due to friction (h_f, feet)	260	87	22
Power input (*HP*, hp)	19.45	6.51	1.65
Annual recurring costs (C_o, $)	$ 9,120	$ 3,520	$ 1,150
Initial capital cost (C_i, $)	$ 30,000	$ 34,000	$ 38,000
Life cycle cost present value (*C*, $)	$124,660	$70,540	**$ 49,940**

⑨ The alternative with the lowest present net value is the **4 inch NPS.** Therefore it is the most economic pipe size.

The content just presented was limited to pressurized piping systems. Gravity flow does not involve external energy input and therefore LCC analysis with regard to power consumption is inconsequential. Gravity flow design applications are often quite challenging however. Coverage of the topic of gravity flow also provides the opportunity to introduce the concept of flow patterns which, by-the-way, are manifested in both gravity and pressurized piping systems.

Gravity Flow

Introduction

Some of the trickiest situations concerning pipe line sizing involve flow solely under the influence of gravity. This is true in both single-phase (full pipe) and two-phase (partially filled) flow patterns. Two-phase flow occurs in a pipe when a gas and liquid are coexistent. This phenomenon is characterized by flow patterns.

Flow Patterns

Flow pattern is distinguished from flow regime where the fluid is said to be laminar or turbulent. Depending on the gas to liquid ratio and the mixture velocity, flow patterns range from bubble flow to dispersed flow. See Figure 4.

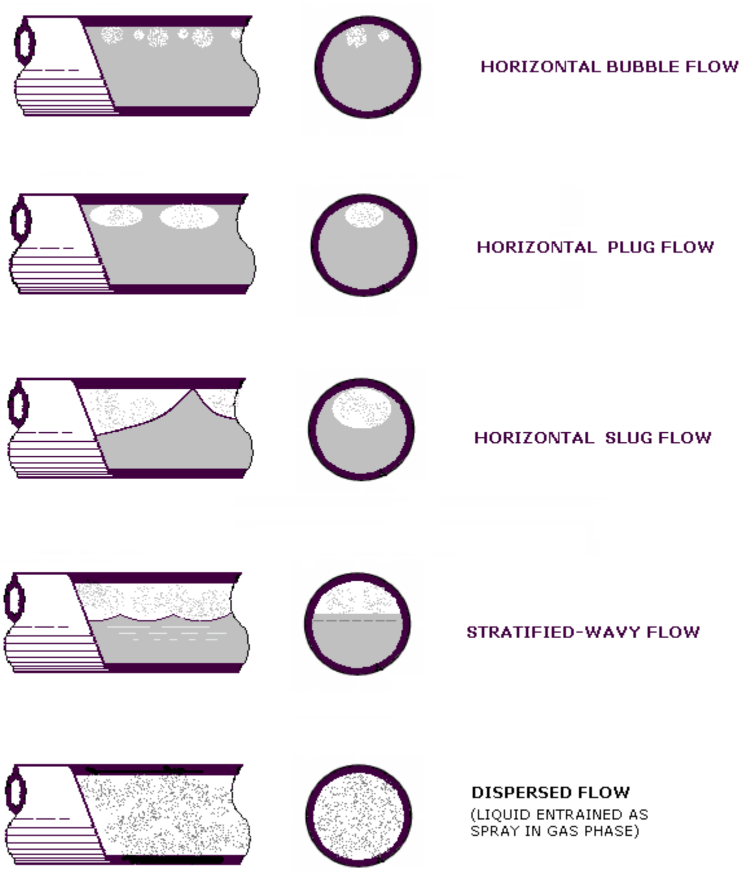


Figure 4 – Flow Patterns in Horizontal Pipes

Flow patterns can change with pipe orientation alone. See Figure 5.

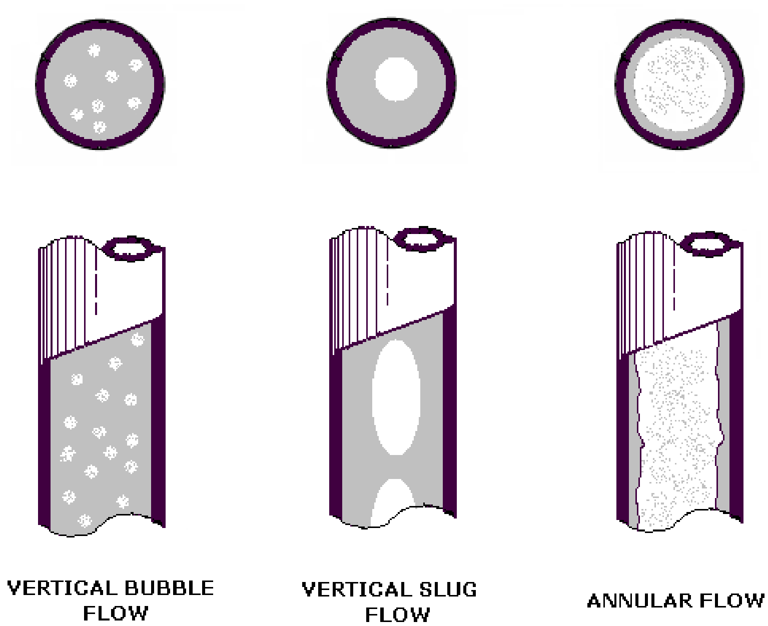
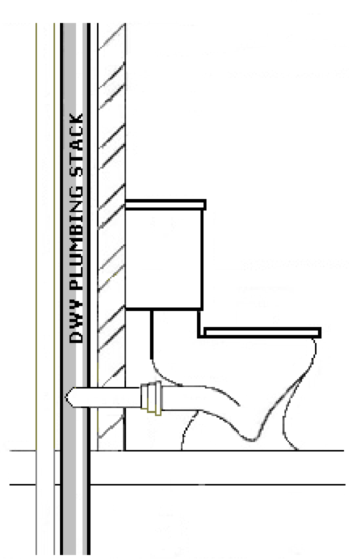


Figure 5 - Flow Patterns in Vertical Pipes

Plumbing Considerations



Obviously the fluid mechanics are the same in process piping and plumbing; however, the terminology and the design approach to select pipe size can be quite different. For instance, vertical runs of processing piping have no specific name; vertical runs of plumbing waste or drainage lines are often called ***stacks***, while supply lines are known as ***risers***. The constituents in the gas phase of two phase process piping flow are varied. The countercurrent core flow of gas in plumbing annular flow is essentially air. Venting in plumbing stacks is needed to preserve the *trap*, that liquid seal which prevents the introduction of obnoxious odiferous vapors into occupied spaces. Gravity flow process piping is not necessarily like plumbing applications where the optimum pipe size must be selected to produce a low velocity liquid flowing vertically downward as an annular film with displaced gas being allowed to flow counter-currently (vent stack). (The term DWV means drain/waste/vent). This is not true in industrial process piping where the gas stream contained within the core is often times flowing concurrently, not counter-concurrently, with the liquid annular film.

Building designers and building mechanical engineers use a simplified design procedure from building codes to select drainage pipe size within buildings. It is customary to size plumbing using a *cookbook* approach. Building codes generally dictate minimum pipe sizes based on the *fixture unit* concept. The procedure assigns fixture units (FUs) to various waste generating plumbing fixtures, such as sinks and showers, then establishes authoritatively the permissible pipe sizes and slopes for a given number of total fixture units contributing to a point in a drainage system. The designer determines the total number of FUs and selects an accommodating pipe size. Table 6 is a typical example of just such a code listing.

TABLE 6 Typical Plumbing Code Fixture Unit Loading Chart

Pipe Size (inches)	*Maximum Fixture Units*
1¼	1
1½	1
2	8
2½	14
3	35
4	216
6	720
8	2640
10	4680
12	8200

Stack flow has been studied extensively[5] because of the deleterious effect which can be created when critical flow rates are exceeded. It has been found that slugs of water and the resultant violent pressure fluctuations *do not* occur until the stack flow exceeds one-third full.[6] Plumbing stacks are generally sized to insure that flow corresponds to $r < \frac{1}{3}$ full because higher flows result in pressure fluctuations which compromise the trap seal. The maximum permissible flow rates in the stack to prevent slug flow can be expressed by the formula:

$$Q = 27.8 r^{5/3} d^{8/3} \qquad \textit{where} \qquad r = \frac{4A}{\pi d^2} \qquad [10]$$

The variable r should not be confused with hydraulic radius. It is the ratio of the cross sectional area of the annular flow to the total cross sectional area of the pipe. See Figure 6. Equation [10] rewritten in terms of d to allow for a direct solution of pipe size yields,

$$d = \left(\frac{Q}{27.8\, r^{5/3}} \right)^{3/8} \qquad [11]$$

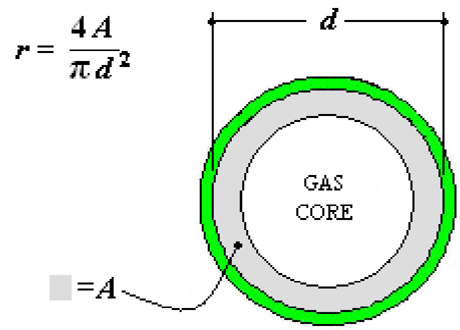


Figure 6 – Plan View of Stack Flow

With regard to gravity flow in partially filled pipes, the term *full* is used somewhat inconsistently. In vertical pipes and stacks, ¼ full and ⅓ full generally refers to the value of *r*, the ratio of flow area to inside pipe area. In horizontal and near-horizontal lines, ¼ full and ⅓ full generally refer to the value of *y*, the relative flow depth (see below).

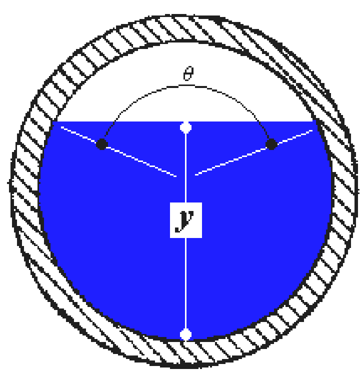


Figure 7 – Relative Flow Depth in Horizontal and Near-Horizontal Pipes

After about 30 feet, gravity flow liquid reaches terminal velocity in vertical runs of pipe. The flow rate in Equation [11] is in gallons per minute once terminal velocity is achieved. The formulas developed for terminal velocity and terminal length, without derivation[5, 6], are given in Figure 8.

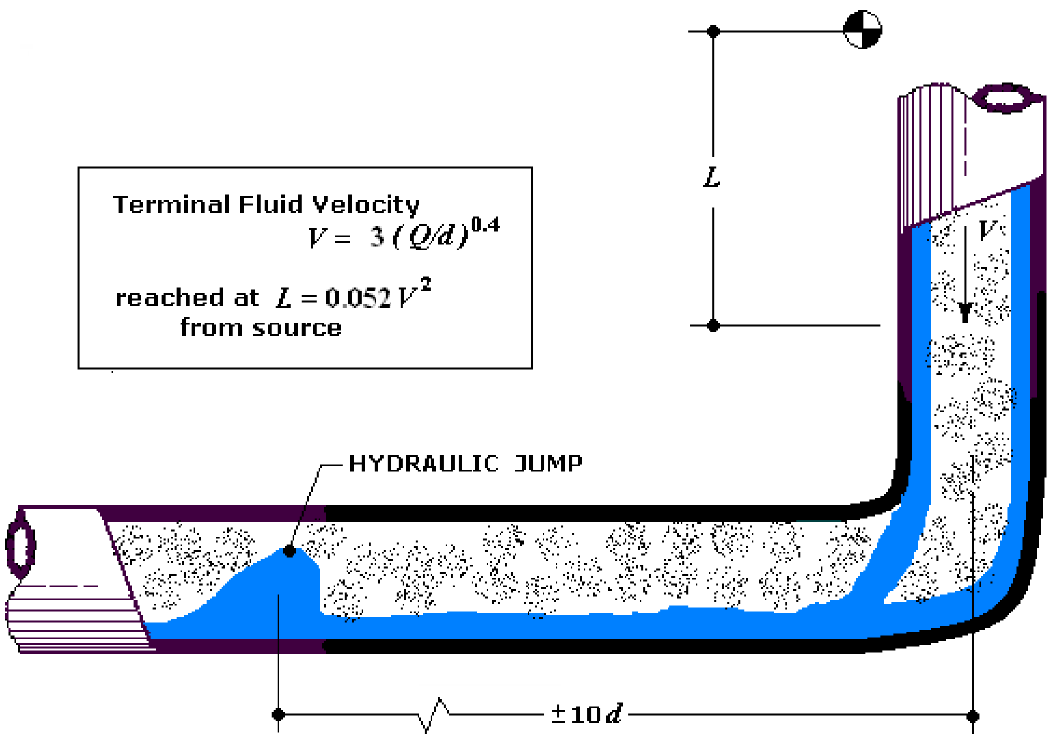


Figure 8 – Vertical to Horizontal Transition Flow

At the transition of vertical to horizontal flow at the base of the pipe, the fluid velocity decreases after turning horizontally. This change in kinetic energy results in a hydraulic jump. While the hydraulic jump is normally of little consequence, the fluid characteristics upstream of the jump can essentially close-off the vapor phase flow, ultimately resulting in potentially damaging slug/plug flow pulsations.

Example 3 Selection of Minimum Pipe Size for a Given Waste Flow Condition

Problem: A proposed high-rise building project's design specifications indicate that plumbing stacks shall be sized to flow no more than ¼ full to prevent damaging slug flow. What diameter pipe should be used to permit waste water to flow at the rate of 550 gallons per minute? Determine the terminal fluid velocity, the vertical distance from the 550 gpm flow source at which this velocity will be achieved, and the approximate lateral distance from the stack at which a hydraulic jump will form after the DWV line turns to a horizontal orientation.

Given: r = ¼ (0.25) [assumed, see below] ; Q = 550 gpm.

Find: A target pipe diameter d, terminal velocity V, terminal length L, and a horizontal distance to jump formation.

Solution:

① While it is not stated in the problem, as is customary practice, ¼ full shall be assumed to mean the ratio of flow area to total pipe area. Using Equation [11] and the given quantities, an inside pipe diameter is calculated to be,

$$d = \left(\frac{Q}{27.8\, r^{5/3}}\right)^{3/8} = \left(\frac{550}{(27.8)(0.25)^{5/3}}\right)^{3/8} = 7.285 \text{ inches}$$

② From Table 13 on page 77 select a 8 inch NPS schedule 40 pipe with **d = 7.981 inches**

③ From Figure 8, calculate the terminal velocity, vertical distance, and lateral displacement of the resulting hydraulic jump to be,

$$V = 3\left(\frac{Q}{d}\right)^{0.4} = 3\left(\frac{550}{7.981}\right)^{0.4} = \mathbf{16.3 \text{ ft/sec}}$$

$$L = 0.052\, V^2 = (0.052)(16.3)^2 = \mathbf{13.8 \text{ feet}}$$

$$10d = (10)(7.981)/12 = \mathbf{\pm 6⅝ \text{ feet}}$$

Civil Engineering Applications

Open channel flows are modeled most frequently by the Manning formula which in some special cases can also be used to select pipe size. Irish Engineer Robert Manning is attributed with the development of this simple, dimensionally homogeneous formula for gravity flow. The Manning formula[7] is semi-empirical. In its original form it is based in terms of fluid velocity,

$$V = \frac{1.486}{n} R^{2/3} S^{1/2} \qquad [12]$$

where V is the liquid cross-section mean velocity and R is the hydraulic radius. Hydraulic radius should not be confused to mean half of the pipe inside diameter. Hydraulic radius is,

$$R = \frac{A}{\text{wetted perimeter}} \qquad [13]$$

The Manning formula does not account for entrance or exit losses, bend losses, losses due to flow through a partially open/closed valve, loss through reducers or any other minor losses. It is best used for uniform steady state flows. Uniform means that the pipe size remains constant and steady state means that the velocity, discharge, and depth do not change with time. These assumptions are rarely ever strictly achieved in process conditions.

The formulas, equations, and inequalities presented for gravity flow are, for the most part, empirical for water at 68° F (μ = 1 cP). Stated reductions in flow rate of 10% have been noted in Reference 9 for fluids with viscosity on the order of 10 cP. Consequently, the information presented must be used with due consideration when very viscous fluids are encountered.

Experience has shown that the maximum gravity water flow does not occur when a near-horizontal pipe is flowing full. Fluid friction reduces flow rate when the ratio of depth to diameter is greater than $y/d \approx 0.938$. A rational mathematical solution verifies that the additional flow area gained by increasing the ratio's value greater than 0.938 is differentially small compared to the large additional pipe surface area presented for fluid friction. In a similar fashion, maximum fluid velocity can be shown to occur at $y/d = 0.81$. Armed with this knowledge, the standard form of the Manning formula presented in Equation [12] can be rewritten to allow the determination of a pipe size which offers maximum gravity flow for a given flow rate and energy gradient:

$$d = \left[1525 \frac{Qn}{\sqrt{S}}\right]^{0.375} \qquad [14]$$

Unlike civil engineering applications of open channel flow where it is often difficult to accurately estimate the roughness coefficient n of a natural channel bed, the Manning roughness coefficients for commercially available pipe materials is fairly well established, varying within a range. These are shown in Table 7. Another useful rewrite of the standard Manning formula is one in which the pipe is flowing full[8]:

$$d = \left[1639 \frac{Qn}{\sqrt{S}}\right]^{0.375} \qquad [15]$$

TABLE 7 Ranges of Manning's *n*

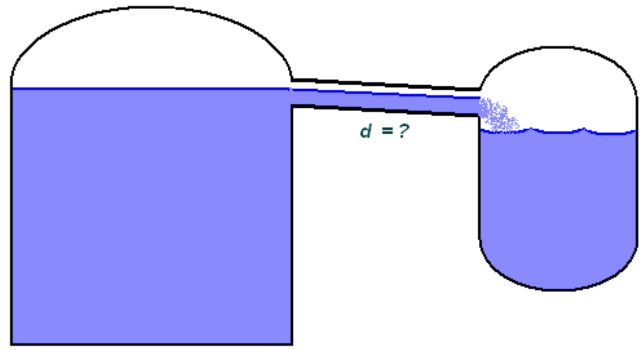
Pipe Material	*Manning's n*
Cement	0.011 – 0.017
Clay	0.010 – 0.015
Ductile Iron	0.011 – 0.016
Plastic	0.010 – 0.015
Steel	0.010 – 0.014

Example 4 Optimum Pipe Size for Near-Horizontal Gravity Flow Transfer Pipe

<u>*Problem:*</u> Cool beer wort (whose physical properties are very similar to H_2O) is to be transferred at the rate of 600 gallons per minute between two brewery vessels through a near-horizontal stainless steel transfer line. The liquid levels in the two vessels will be automatically maintained to establish uniform, steady flow in this crossover pipe. Spatial limitations dictate that the maximum slope (gradient) for this line be 1:14. Discounting vessel liquid level effects at the pipe entrance and exit, and without concern for gas entrainment, what is the smallest pipe inside diameter which will accomplish this clarified wort transfer?

Given: $n = 0.013$ (stainless steel) ; $Q = 600$ gpm = (1.34 ft³/sec) ; S = 1:14 (or 0.07).

Find: The optimum (smallest) pipe diameter d for the stated process conditions.



Methodology: Since no mention of pressurization is made, the decanting operation described implies a gravity flow partially filled pipe scenario. Accordingly, the Manning formula appears to be acceptable. The smallest pipe for this application will be the one which provides the maximum partially filled pipe flow depth for uniform 600 gpm steady flow.

Solution:

① Using Equation [10] and the given quantities, calculate a trial diameter of

$$d = \left[1525 \frac{Qn}{\sqrt{S}}\right]^{0.375} = \left[1525 \frac{(1.34)(0.013)}{\sqrt{0.07}}\right]^{0.375} = 5.63 \text{ inches}$$

② Find the nearest standard pipe size corresponding to the trial size found in Step ①. From Table 13 on page 77 select a **6 inch NPS schedule 10 standard pipe size with $d = 6.357$ inches**

Process Drainage

Basically, gravity liquid flow through pipe from relatively large source volumes can be approached from three distinct design perspectives. These are (1) full pipe single phase flow; (2) gas entrained flow; and (3) self-venting flow. Of the three conditions, the full single phase flow design generally results in the smallest pipe size but it can be the most difficult to maintain. Problems arise when actual flow conditions do not match the design approach used. When such pipes do not run full, considerable amounts of gas can be drawn down by the liquid. The amount of gas entrained is a function of pipe diameter, pipe length, and liquid flow rate.

Entrainment inhibits liquid gravity flow by raising the pressure drop through the piping above that for single-phase flow, and reduces the static head available for over-coming the pressure drop.

Figure 9 is a classical example of a design situation wherein a drainage pipe size has been determined on the basis of full pipe flow but in fact the situation is entraining gas because a critical minimum head fails to be maintained. In this case an absorption column is being drained through a vertical pipe whose size has been designed for full flow. Absorption columns are vessels which are used throughout the petrochemical industry to accomplish mass transfers of fluids and gases.

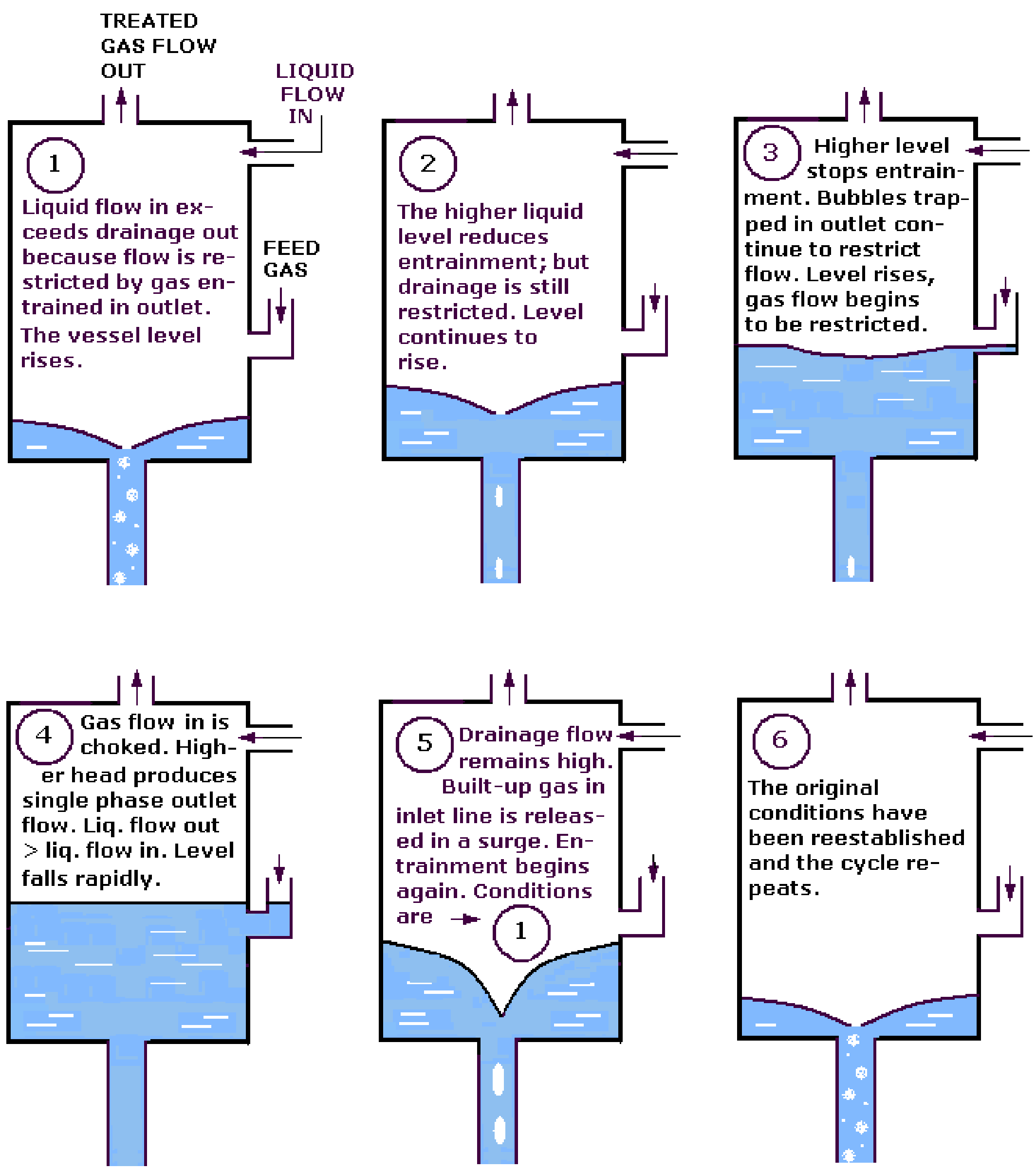


Figure 9 – Cyclic Surging in Gas Entrained Drain (adapted from Reference 9)

Froude Number

William Froude was a 19th century British Engineer, hydrodynamicist, and naval architect. He formulated reliable laws for the prediction of ship stability and for the resistance water offers to marine craft. He is credited with the development of the dimensionless number presented here. He developed a dimensionless number that compares inertial and gravitational forces.

In the context of this course, the Froude number is a dimensionless liquid flow rate term which can be used as a superficial volumetric comparator:

$$F_r = \frac{V}{\sqrt{\frac{d}{12}g}} \qquad [16]$$

Where V is based on the assumption of full-pipe flow.[10]

The number is useful in sizing gravity flow piping because extensive observations have yielded empirical relationships which define the interdependency of Froude number, static heads, flow rates, and pipe size.

Pipe Size for Flooded Gravity Flow

For full flow, the pipe size must be based on single-phase criteria. To avoid gas entrainment in the full-pipe-flow design, the motive force, *i.e.* static head, must be great enough to maintain a flooded inlet condition. According to Hills[9], two simultaneous conditions must be met in order to insure full pipe single phase flow free from gas entrainment. These conditions which involve static liquid heights and Froude number values are given by the following inequalities:

For full pipe single phase liquid flow from bottom outlets:

$$h > 10.704\left(\frac{12Q^2}{gd}\right)^{0.25} \quad \text{and} \quad F_r < 1.6\left(\frac{h}{d}\right)^2 \qquad [17]$$

Where Q is in ft^3/sec.

For full pipe single phase liquid flow from side outlets:

$$h > 201803 \frac{Q^2}{gd^4} \qquad \text{and} \qquad F_r < \sqrt{\frac{2h}{d}} \qquad [18]$$

The reference datums for these relationships are shown in Figure 10.

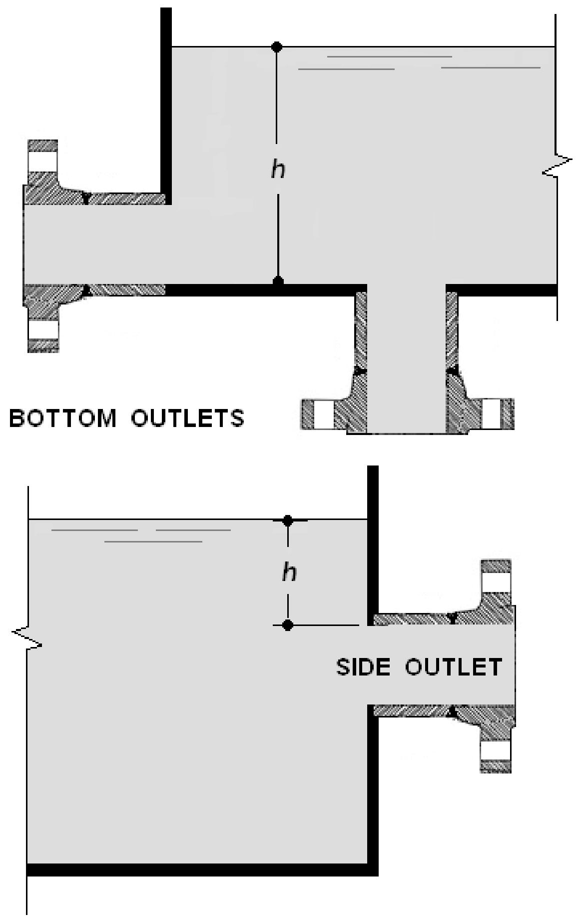


Figure 10 – Critical Static Head for Full Pipe Single-Phase Liquid Drainage

Pipe Size for Self-Venting Flow

Self-venting drainage is generally carried out by means of a side outlet arrangement. Hills[9] suggests that two simultaneous conditions must be met in order to insure side outlet self-vented flow. These conditions, which involve a diameter at the entrance boundary to the outlet and the Froude number, are given by the following inequalities:

$$d > 12\left(\frac{4Q}{0.3\pi\sqrt{g}}\right)^{0.4} \quad \text{with} \quad F_r < 0.3 \qquad [19]$$

The above inequalities insure that the depth of flow in the pipe will be less than half full at the entrance to the outlet. To insure self-vented gravity flow, a critical far field (away from the outlet) static fluid height less than 0.8*d* must be maintained. This fluid height datum is shown in Figure 11.

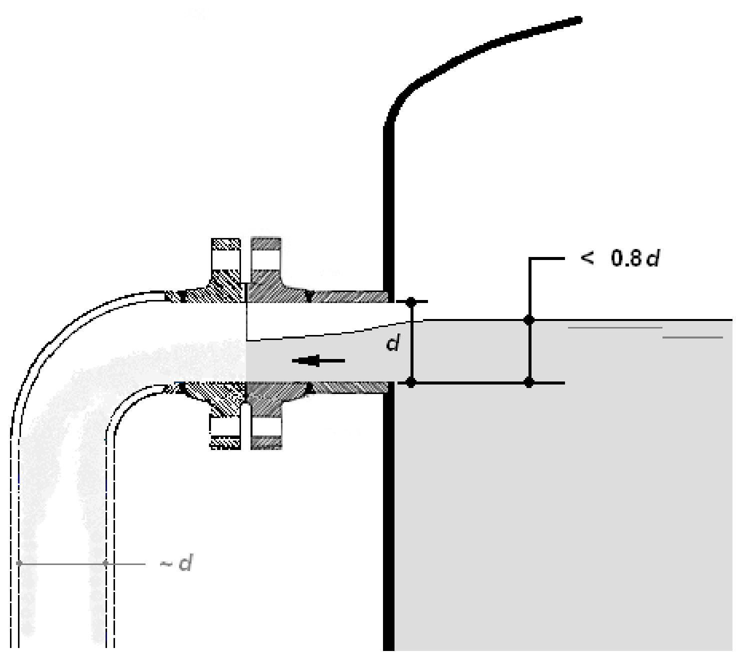


Figure 11 – Static Head to Promote Self-Venting Drainage

A regression analysis by Whitesides of Hills[9] pipe flow data resulted in the following flow capacity correlation equation which related nominal pipe size to side outlet gravity flow for the arrangement shown in Figure 11:

$$Q = 1.23\, d^{2.5} \qquad [\text{for } 2 \le d \le 18] \qquad [20]$$

Where Q is in gallons per minute

Written in terms of d, Equation [20] becomes,

$$d = 0.92\, Q^{0.4} \qquad [\textit{for } 5 < Q < 1{,}600] \qquad [21]$$

For a desired flow rate, a diameter for the outlet's entrance can be estimated directly from the Inequality [19], from which a fluid velocity and Froude number can be established. If the Froude number is less than 0.3, then it can be assumed that the selected opening size is acceptable. The design procedure for self-venting gravity flow is shown in Example 5.

Example 5 Optimum Pipe Size for Self-Venting Gravity Flow from Vessel

Problem: It is desired to vent liquid from an atmospheric tank at a minimum rate of 120 gallons per minute through a side outlet self-venting drain (overflow arrangement Figure 11). What size free opening should be made in the tank wall? What far field (away from the outlet's entrance) maximum liquid level should be allowed above the outlet's invert to avoid air entrainment?

Given: $Q = 120$ gpm $= 0.267$ ft^3/sec

Find: An acceptable opening diameter d and corresponding critical static head h which should not be exceeded to insure self-venting drainage without surging.

Methodology: Using the Inequality [19], estimate an initial opening size. Knowing the required flow rate, calculate a corresponding fluid velocity for this diameter. Test Inequality [19] to determine if the Froude number is acceptable for this size. Repeat this process, increasing the size, until an acceptable size is found.

Solution:

① Determine a trial pipe size from Inequality [19] for a self-vented side outlet configuration,

$$d \; > \; 12\left(\frac{4\,Q}{0.3\,\pi\sqrt{g}}\right)^{0.4} \; > \; 12\left[\frac{(4)(0.267)}{(0.3)(\pi)\sqrt{32.17}}\right]^{0.4} \; > \; 6.30 \text{ inches}$$

② Find the nearest standard pipe size corresponding to the trial size found in Step ①. From Table 13 on page 77 select a **6 inch NPS schedule 10 standard pipe size with $d = 6.357$ inches**

③ With the required flow rate, determine the fluid velocity for the selected standard pipe size assuming full pipe flow. Use a rearrangement of the continuity equation.

$$V \; = \; \frac{Q}{A} \; = \; \frac{4\,Q}{\pi\,d^2} \; = \; \frac{(4)(0.267)}{\pi(6.357\,/\,12)^2} \; = \; 1.21 \text{ ft / sec}$$

④ Calculate the Froude number for the selected standard pipe size and the computed fluid velocity,

$$F_r \; = \; \frac{V}{\sqrt{\frac{d}{12}\,g}} \; = \; \frac{1.21}{\sqrt{(6.357\,/\,12)(32.17)}} \; = \; 0.29$$

$F_r = 0.29 < 0.3$ ∴ by Inequality [19], the selected 6 inch NPS Schedule 10 pipe is o.k. for side outlet self-vented flow

⑤ Maximum static head to insure ½ flow depth in the outlet will be $h = (0.8)(6.357)$ **≈ 5 inches**

⑥ Use Equation [20] to determine the probable flow rate for the selected drain outlet pipe size,

$$Q \; = \; 1.23\,d^{\,2.5} \; = \; (1.23)(6.357)^{\,2.5} \; = \; \textbf{125 gal/min} \; > \; \textbf{120 gal/min required, } \therefore \textbf{ o.k.}$$

Another popular name for the self-venting gravity flow side outlet arrangement is *overflow arrangement.* Another popular name for the bottom outlet arrangement is *drain arrangement.* The entrance to an overflow arrangement is always above the horizontal surface. The entrance to a drain arrangement is always flush with a horizontal surface.

Horizontal Fully Vented Flow

In many instances gravity flow is desirous in horizontal pipe lines running partially full. Obviously this cannot occur unless the line is slightly inclined to provide static head sufficient to overcome the frictional losses. This application is termed *near-horizontal line flow.* Hills[9] has recommended a minimum slope of 1:40 for these process situations with adequate free area remaining in the pipe to allow counter-current gas flow.

While it has been stated that the theoretical maximum flow rate occurs in near-horizontal pipes at a flow depth of $y = 0.938d$, sufficient free area must be reserved in the upper region of the pipe to preclude gas carryover in the liquid and to allow for gas phase counterflow. Liquid depths in these two phase flows should not exceed more than ½ the inside pipe diameter for nominal pipe up to 8 inch NPS and not more than ¾ of the diameter for larger pipes.

A hydraulic jump, see Figure 8, will occur at the base of vertical runs when near-horizontal pipe lines flowing partially full turn to a vertical orientation, and then resume a near-horizontal orientation downstream. As long as there is sufficient distance to the first flow obstruction encountered in the near-horizontal pipe, the surge associated with this hydraulic jump is of no consequence. The jump normally occurs at approximately $10d$ downstream of the change in the flow orientation from vertical to near-horizontal.

Pressurized Flow

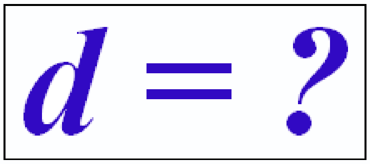
Introduction

From confined space automotive applications to mammoth systems in industrial facilities, fluids are transferred by piping to and from and through the systems that are now taken for granted as part of our daily lives. Needless to say, a huge amount of energy is expended in accomplishing these transfers. Pressurized flow systems account for nearly 20% of the world's electrical energy demand and range from 25-50% of the energy usage in many industrial plant operations.[11]

In the context of this course, pressurized flow means flow that is induced artificially by a pump, compressor, or a higher pressure source such as a boiler.

Design Approach versus Analysis

Flow equations published in the open literature are invariably based on determining pressure loss due to fluid friction based on a known pipe size. As such, they represent an ***analysis approach***. In many cases the requirements placed on a system are established in terms of a maximum allowable pressure drop or energy loss, a required volume or mass flow rate, the fluid properties, and the material composition of the pipe to be employed. Accordingly, a suitable pipe size must be determined which will meet these requirements. As shown in previous sections of this course, situations where the determination of an optimum pipe size is required present a true ***design approach*** that can necessitate the rearrangement of the normal flow equations in terms of *d*.



Estimation of Pipe Size for Liquids

Adams[12] developed two mathematic equations which very closely describe the pressurized flow of liquids in typical pipes within the realm of normal supply pressures. They offer a simple way to do preliminary pipe sizing. If used for typical flow situations for normal fluids, they provide a quick size determination.

The Adams equations in their original form are cubic in terms of pipe diameter, and as such are more suited for determining the approximate flow corresponding to a known nominal pipe size. However, with manipulation they can provide a quick means to determine an approximate nominal pipe size. A least squares geometric regression analysis of Adams' equation for lower flow rates yields the following correlation equation:

$$d = 0.25\sqrt{Q} \qquad \text{[for } Q < 100 \text{ gal/min]} \qquad [22]$$

Adams' equation for higher flows, rewritten in terms of d to allow for a direct solution yields,

$$d = \left(\frac{Q}{1.2}\right)^{1/3} - 2 \qquad \text{[for } Q > 100 \text{ gal/min]} \qquad [23]$$

Example 6 Quick Determination of Pipe's Flow Capacity

Problem: Nominal pipe sizes have yet to be indicated on a preliminary P&ID (process and instrumentation diagram). Two separate process lines with non-viscous liquid flow rates of 85 gpm and 260 gpm are shown. What are the probable nominal pipe sizes for these two lines?

Given: $Q = 85$ gal/min ; $Q = 260$ gal/min.

Find: A target pipe diameter d for the stated flow rates.

Solution:

① Because handbook or table data are not available, use the reformations of the equations given by Adams[12] to make quick determinations. In the first instance the stated flow rate is less than 100 gpm ∴ use equation [22]:

$$d = 0.25\sqrt{Q}$$
$$d = 0.25\sqrt{85}$$
$$d = 2.3 \text{ inches}$$

∴ A preliminary nominal pipe size for the first pipe line would be **2½ inches.**

② In the second instance the stated flow rate is greater than 100 gpm, ∴ use equation [23]:

$$d = \left(\frac{Q}{1.2}\right)^{1/3} - 2$$
$$d = \left(\frac{260}{1.2}\right)^{1/3} - 2$$
$$d = 4 \text{ inches}$$

∴ A preliminary nominal pipe size for the second pipe line would be **4 inches.**

Pressure Conservation

Invariably, in the case of many process fluids, the size determination becomes one that is driven by pressure conservation - that is to say, the size that will result in a specified minimum pressure value at some terminal point. System design problems usually contain too many unknowns to allow a direct solution. The resulting design approach can require a procedure known as iteration; a trial-and-error solution method in which a trial value is assumed for a given unknown, thus allowing the computation of a related unknown. The iteration procedure provides a means of checking the accuracy of the assumed trial value and also indicates the new trail value to be used if an additional computational cycle is required. The following example illustrates the application of this design procedure.

Instructor's Note: While extremely useful, automated computational methods often preclude a clear understanding of the underlying engineering. The example that follows employs a classical method of design which was required prior to the availability of powerful desktop computing resources. It is presented here solely to illustrate the concepts of fluid flow relationships and their importance in selecting pipe size.

Example 7 Optimum Pipe Size for Pressure Conservation (Classical Method)

Problem: In a petrochemical unit operation, *para*-xylene [$C_6H_4(CH_3)_2$] at 86° F is required to be transferred at a rate of 100 gallons per minute through a schedule 40 steel pipe segment which has a length of 188 feet. The *p*-xylene has density and viscosity values of 53.6 lb/ft³ and 0.6 cP respectively. What is the smallest nominal pipe size that can be installed that will result in a pressure drop through the pipe segment not exceeding 2 psi?

Given: Q = 100 gal/min (0.223 ft³/sec) ; L = 188 feet of steel pipe (ε = 0.00015 feet) ;
ρ = 53.6 lb/ft³ ; μ = 0.6 cP ($\upsilon = 7.5 \times 10^{-6}$ ft²/sec) ; ΔP = 2 lb/in² (h_f = 5.37 feet of *p*-xylene)

Find: The smallest pipe diameter d for the stated process requirements.

Methodology: The flow velocity V and the pipe diameter d are unknown so that neither the Reynolds number N_{Re} nor the relative roughness ε/D can be calculated. The procedure will be to assume a value of f and calculate a corresponding value of d using the energy loss equation and the continuity equation. A Reynolds number and a relative roughness will then be calculated based on this value of d. Using the Moody diagram (similar to Figure 2), a new value of f will be obtained. This procedure will be repeated unit the value of f is repeated (or remains appreciably unchanged) and all of the flow equations are satisfied. During the design procedure, care must be taken to remember that classical flow equations employ the variable D (in feet).

Solution:

① The flow velocity, Reynolds number, and the relative roughness are all dependent on the pipe diameter. Without these values the friction factor cannot be determined directly. Iteration is required to solve this system design problem since there are too many unknowns.

② Using a form of the continuity equation [$V = Q/A$] and the flow area formula [$A = \pi D^2/4$], rewrite the Darcy-Weisbach energy loss equation to appear as

$$h_f = f\,\frac{L}{D}\frac{V^2}{2g} = f\,\frac{L}{D}\frac{16\,Q^2}{\pi^2 D^4 (2g)} = \frac{8\,L\,Q^2}{\pi^2 g}\,\frac{f}{D^5}$$

③ Rewrite the resulting equation above in terms of *D,* insert the known values, and evaluate:

$$D^5 = \frac{8 f\,L Q^2}{g\pi^2 h_f} = \frac{(8)(188)(0.223)^2}{\pi^2 (32.17)(5.37)} f = 0.049 f$$

④ Use the continuity equation and the known quantities to write a form of the Reynolds number containing D:

$$N_{\mathrm{Re}} = \frac{V D}{\nu} = \frac{4Q}{\pi \nu D} = \frac{(4)(0.223)}{7.5 \times 10^{-6} \pi D} = \frac{3.79 \times 10^4}{D}$$

⑤ Using the absolute roughness value for steel, the relative roughness in terms of D becomes

$$\frac{\varepsilon}{D} = \frac{0.00015}{D}$$

⑥ A trial value of f must now be assumed. Since both N_{Re} and ε/d are unknown, the Moody diagram yields no specific help with the assumption of the initial value. Referring to the left abscissa of Figure 2, assume a mid-range value of the Colebrook friction factor of 0.03.

⑦ Compute the value of D from the Step 3 equation based on the initial assumption of f

$$D^5 = 0.049\,f$$

$$D = \left[(0.049)(0.03)\right]^{0.2} = 0.271 \text{ feet}$$

⑧ For this value of D, the following values can be computed

$$N_{\text{Re}} = \frac{3.79 \text{ x} 10^4}{D} = \frac{3.79 \text{ x} 10^4}{0.271} = 1.4 \text{ x} 10^5 \quad ; \quad \frac{\varepsilon}{D} = \frac{0.00015}{D} = \frac{0.00015}{0.271} = 0.00055$$

⑨ Figure 2 (or a more accurate version thereof) yields a Colebrook friction factor of $f = 0.02$ corresponding to the above calculated Reynolds number and relative roughness. Repeating the process for $f = 0.02$

$$D = \left[(0.049)(0.02)\right]^{0.2} = 0.25 \text{ feet}$$

$$N_{\text{Re}} = \frac{3.79 \text{ x} 10^4}{0.25} = 1.52 \text{ x} 10^5$$

$$\frac{\varepsilon}{D} = \frac{0.00015}{0.25} = 0.0006$$

⑩ The Moody diagram yields a Colebrook friction factor of $f = 0.019$ corresponding to the above calculated Reynolds number and relative roughness. Since this value of f is not appreciably different from the previous iteration, $D = 0.25$ feet ($d = 3$ inches) is probably the optimum size. From Table 13, select as the optimum pipe size a 3 inch NPS Schedule 40 pipe with **d = 3.068 inches** . As a check, the actual pressure drop for this selected pipe size will be computed to insure that the maximum allowable pressure drop has not been exceeded:

$$V = \frac{(0.223)(4)}{\pi\,(3.068\,/\,12)^2} = 4.35 \text{ ft/sec} \quad \text{and}$$

$$h_f = f\,\frac{L}{D}\frac{V^2}{2g} = 0.195 \cdot \frac{(188)(4.35)^2}{(3.068\,/\,12)(2)(32.17)} = \mathbf{4.21 \text{ feet } \textit{p}\text{-xylene} = 1.57 \text{ lb/in}^2 \ \therefore \ \text{o.k.}}$$

Pipe Size via Fluid Velocity

The most common pipe size selection criteria is most usually based on fluid velocity. Years of experience have generated rules-of-thumb regarding normal velocities for given fluids. Knowing the flow rate, optimum pipe size is back-calculated from the traditional continuity equation which can be rewritten in the form

$$d = \sqrt{\frac{4Q}{\pi V}} \qquad [24]$$

The generally accepted body of data for normal velocities is presented in the Table 8.

TABLE 8 Widely Published Recommended Flow Velocities

Service	*Mean Flow Velocity (feet per Minute)*	*Mean Flow Velocity (feet per second)*
Boiler Feed Water	500 – 600	8 - 10
General Water Service	200 – 500	3 - 8
Process Water	300 – 600	5 - 10
Pump Discharge	300 – 600	5 - 10
Pump Suction	200 – 500	3 - 8
Saturated Steam (high pressure)	6,000 – 10,000	100 - 165
Saturated Steam (low pressure)	4,000 - 6,000	65 - 100
Superheated Steam (high pressure)	10,000 – 15,000	165 - 250

For low-viscosity liquids economic optimum velocity is typically in the range of 6 to 8 feet per second. For gases with density ranging from 0.015 to 1.25 lb/ft^3 the economic optimum velocity is about 130 down to 30 feet per second.

The term optimum velocity has often been incorrectly limited to mean the velocity for a certain flow rate and pipe size (flow area) which results in an economic balance between hardware and energy costs. Optimum velocity may also very well mean the velocity which will maintain a certain degree of agitation; or will produce an efficient or improved heat transfer; or may limit abrasion or conserve the suspension of solids.

Pipe Size based on Fluid Velocity Effects

Not all fluid particles travel at the same velocity within a pipe. The shape of the velocity curve (the velocity profile across any given section of the pipe) depends upon whether the flow is laminar or turbulent. If the flow in a pipe is laminar, the velocity distribution at a cross section will be parabolic in shape with the maximum velocity at the center being about twice the average velocity in the pipe. In turbulent flow, a fairly flat velocity distribution exists across the section of pipe, with the result that the entire fluid flows at a given single value. Figure 12 helps illustrate the above ideas. The velocity of the fluid in contact with the pipe wall is essentially zero and increases with the distance away from the wall.

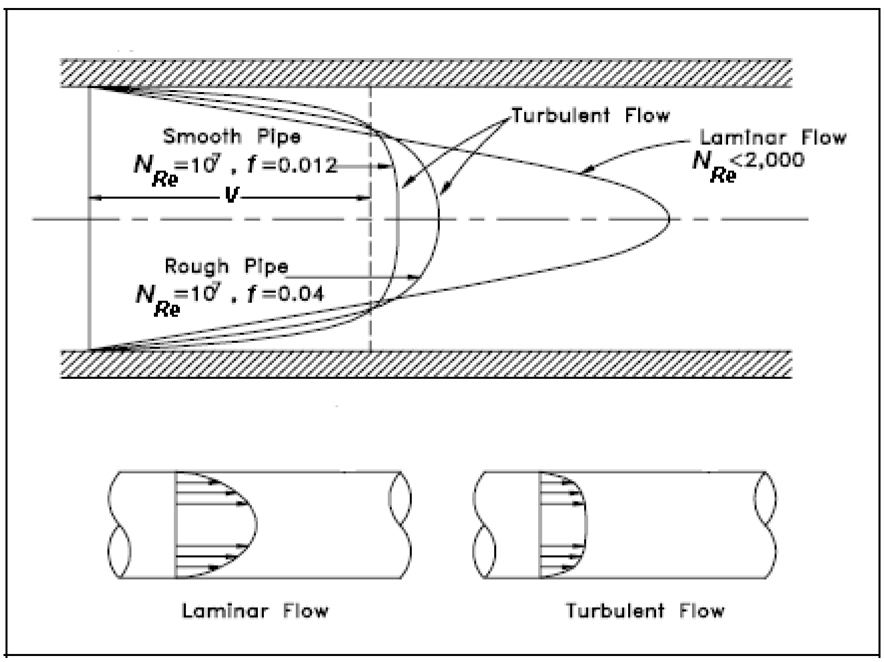


Figure 12 – Velocity Profiles (adapted from Reference 13)

In many process applications pipe size is dictated by fluid velocity effects. Fluid velocity can affect noise generation, vibration, erosion, and the fluid's transport efficacy. Flow velocity is normally maintained high enough such that liquid flows are swept along avoiding the slug flow pattern.

This section of the course addresses two important considerations of fluid velocity and therefore pipe size selection. These considerations are flow induced erosion and settling velocity.

For a given required mass flow rate, pipe size can have a significant bearing on internal pipe erosion. An important consideration in the selection of pipe to provide pressure resisting strength is the tolerable amount of erosion, in terms of loss of pipe wall thickness. Erosion occurs in both single and multiphase flow patterns. With regard to single phase flow, studies have shown that gas flows are generally more damaging than liquid flows.[14] With that said, flow of concentrated liquid acids have been found to be extremely erosive.

It logically follows that erosion in straight pipes is less severe than that in elbows and other fittings for a comparable set of operating conditions. In elbows fluid can possess enough momentum to traverse the flow streamlines and make direct impinge against the pipe wall. Erosion can also occur in straight sections of pipe even though there is no mean velocity component directed toward the wall. This occurs when turbulent fluctuations in the flow provide the matter with momentum in the radial direction, forcing it into the pipe wall.

In the oil and gas industries, the presence of contaminants in produced oil and natural gas represents a major concern because of the associated erosive wear occurring in various flow passages. The commonly used practice for controlling erosion in gas and oil producing wells is to select a pipe size to limit production velocities following the provisions of *American Petroleum Institute. Recommended Practice 14E*.[15] This document contains a simplified, very conservative formula to calculate a threshold velocity, and thereby a corresponding pipe size, below which an allowable amount of erosion occurs. Rewritten in terms of d with gas flow Q in ft^3/sec this expression becomes,

$$d = 12\sqrt{\frac{4Q\rho_m^{0.5}}{\pi c}} \qquad [25]$$

While providing the advantage of computational ease, this approach has some disadvantages. The variable c is an *empirical* constant, intended to represent the normally occurring corrosive and erosive effects of entrained contaminants in the fluid.

A misconception of the API erosive velocity formula is that values of *c* account for varying degrees of sandy petroleum flows. In fact, assumption of values for *c* is intended to allow for a range of clean service conditions.

The original formula from which Equation [25] is derived has been found to be very conservative for relatively clean fluids and inapplicable for corrosive/sand containing processes. The definition of its use and the validity of the assigned values of 100, 125, 150, and 200 for *c* have generated a host of supplemental third party guidance literature.[15] The value of *c* is irrespective of volume or weight fraction of gas-to-liquid in the petroleum. Multiphase flow is accounted for in the formula by the incorporation of the variable representing the density of the fluid mixture. In real-world applications many factors, in addition to the density of the mixture, influence the erosion rate. These include flow geometry, particle size, and Reynolds number. Shirazi *et al.*[15] have presented a method to calculate threshold velocities to overcome these limitations by accounting for many of the physical variables in the flow and erosion processes and includes a way to predict the maximum penetration (pipe wall thickness reduction) rate for sand erosion. In reality erosive material can travel in a non-uniform, transient slug flow pattern, which settles and forms deposits. As the deposits increase, the flow cross-sectional area decreases, and the resulting erosive velocity increases, entraining and slugging through the deposits that have previously settled.

Gas erosional velocity and pipe size selection will be discussed in more detail later in the course.

Solids Suspension, Settling, and Sedimentation

If a fluid, such as water, is flowing, it can carry suspended particles. The *settling velocity* is the minimum velocity a flow must have in order to transport, rather than deposit, sediments, and is given by Stokes Law.

If the flow velocity, often in these instances referred to as shear velocity, is greater than the settling velocity, sediment will be transported downstream as *suspended load*. As there can be a range of different particle sizes in the flow, some will have sufficiently large diameters that they settle on the lower pipe wall, but still be displaced downstream. In many cases of process design, the terminal velocity of spherical particles (the settling velocity) must be ascertained. In most cases the application of Stokes' law is appropriate; however, in some cases other flow regimes apply which require a different form of the equation to determine settling velocity.

McCabe and Smith[16] give a method for determining the correct flow regime using physical data. A dimensionless constant K is employed which has the value of:

$$K = 34.81\, D_p \left[\frac{\rho_f\,(\rho_p - \rho_f)}{\mu^2} \right]^{1/3} \qquad [26]$$

Flow regimes are categorized in ranges of either **Stokes range**, **intermediate range**, or **Newtonian range**.

TABLE 9 Constants *b* and *n* in Equation [27] for three operative flow regimes

Law	*Range*	*b*	*n*
Stokes	$K < 3.3$	24.0	1.0
Intermediate	$3.3 \geq K \leq 43.6$	18.5	0.6
Newton's	$K > 43.6$	0.44	0

Caution: computed values of K exceeding 2,360 result in a regime which exhibits abrupt changes in the drag coefficient with minimum changes in velocity, nullifying the equations presented here.

After the appropriate flow regime has been determined, the sediment or solids single particle settling velocity can be found from

$$V_S = \left[\frac{D_p^{1+n}\,(\rho_p - \rho_f)}{b\,\mu^n \rho^{(1-n)}} \cdot \frac{43.87\,(1/12)^{1+n}}{(6.72\,\mathrm{x}\,10^{-4})^n} \right]^{1/(2-n)} \qquad [27]$$

using the constants b and n given in Table 9. Single particle settling velocity by itself is of limited usefulness since practical applications generally involve agglomerations. Fortunately, Perry[17] has provided a relationship which accounts for hindered settling due to the interaction of proximate particles

$$V_{SH} = V_S\,(1 - c)^m \qquad [28]$$

Where m has the value shown in Table 10:

TABLE 10 Single Particle Settling Velocity Modification Exponent for Hindered Settling Equation [28]

Reynolds No. Range	***m***
$N_{Re} < 0.5$	4.65
$0.5 \geq N_{Re} \leq 1300$	$4.375\ N_{Re}^{-0.0875}$
$N_{Re} > 1300$	2.33

For a rigorous treatment of this topic it is recommended that Reference 18 be read. Barring its availability, the practical concept of solids suspension conservation is best illustrated by the example that follows.

In fluid mechanics two-phase flow occurs in a system containing gas and liquid. The term *multiphase flow* tends to be used to characterize two-phase flow in cases where the two phases are not chemically related (*e.g.* dusty gases).

Example 8 Conservation of Suspended Solids in a Transfer Pipe

Problem: It is proposed to transport iron particulate through a pipe via a gas stream at a rate of 50 ft³/min. The iron particles, which will constitute a mean volume fraction of 10% in the resulting mixture, have been determined to be of uniform size and fairly spherical, measuring 0.01 inches in diameter. The particle density is 500 lb/ft³. The transport fluid (gas) density and viscosity are 0.08 lb/ft³ and 0.02 cP respectively. Without regard to pressure drop or erosion, what is the probable nominal diameter pipe that can be used to accomplish the multiphase transport of this metal?

Given: Q = 50 ft³/min ; c = 0.10 ; D_P = 0.01 inches ; ρ_P = 500 lb/ft³ ; ρ_F = 0.08 lb/ft³ ; μ = 0.02 cP

Find: A target pipe diameter d for the stated process application.

Solution:

① In order to have unsedimented transport, the shear (transport) flow velocity must be greater than the suspended iron particle hindered settling velocity. First determine the appropriate flow regime by computing the McCabe-Smith constant using Equation [26],

$$K = 34.81\,D_p\left[\frac{\rho_f\,(\rho_p - \rho_f)}{\mu^2}\right]^{1/3} = (34.81)(0.01)\left[\frac{(0.08)(500-0.08)}{0.02^2}\right]^{1/3} = 16$$

Since K = 16, from Table 9 it can be seen that the ***intermediate flow regime*** is applicable and consequently the constants b and n are assigned the values of 18.5 and 0.6 respectively.

② Next, calculate the single particle settling velocity from Equation [27],

$$V_S = \left[\frac{D_p^{1+n}\,(\rho_p - \rho_f)}{b\,\mu^n\,\rho^{(1-n)}}\cdot\frac{43.87\,(1/12)^{1+n}}{(6.72\,\mathrm{x}\,10^{-4})^n}\right]^{1/(2-n)} = \left[\frac{(0.01)^{1+0.6}\,(500-0.08)}{(18.5)(0.02)^{0.6}\,(0.08)^{1-0.06}}\cdot\frac{43.87\,(1/12)^{1+0.6}}{(6.72\,\mathrm{x}\,10^{-4})^{0.6}}\right]^{1/(2-0.6)}$$

$$V_S = 12 \text{ ft/sec}$$

③ Compute the particle Reynolds number using

$$N_{Re} = \frac{124\,D_p\,V_S\,\rho_f}{\mu} = \frac{(124)(0.01)(12)(0.08)}{0.02} = 60$$

(the coefficient of 124 converts inconsistent units)

④ Since N_{Re} = 60, from Table 10 find m from the expression

$$m = 4.375\, N_{Re}^{-0.0875} = (4.375)(60)^{-0.0875} = 3.06$$

⑤ Determine the hindered settling velocity of the iron solids using Equation [28]

$$V_{SH} = V_S\,(1 - c)^m = 12(1 - 0.10)^{3.06} = 8.7 \text{ ft / sec}$$

⑥ Select a suitable pipe diameter which will produce a flow velocity that exceeds V_{SH} . Use the re-arranged continuity equation previously presented to arrive at a *target* size. Be careful to convert quantities to insure consistent units.

$$d = \sqrt{\frac{4Q}{\pi V_{SH}}} = \sqrt{\frac{(4)(50\,\text{ft}^3)(\text{sec})(\text{min})(144\text{in}^2)}{(\text{min})(\pi)(8.7\,\text{ft})(60\,\text{sec})(\text{ft}^2)}} = \textbf{4.2 inches}$$

⑦ Since this inside diameter will result in a flow velocity ***equal*** to the hindered settling velocity, a value actually smaller than 4.2 inches must be used in order to produce a flow velocity that ***exceeds*** the hindered settling velocity.

Compressible Flow

Flow where the density varies by more than 5% is considered compressible. An incompressible flow is flow in which the density of the fluid is constant or nearly constant. Liquid flows are normally treated as incompressible. Unlike liquid steady-state flow where the mean flow velocity remains relatively constant for a given pipe size, gas flow velocity can vary along the pipe's length. This is due to compressibility effects, and variation in pressure and temperature.

The Compressibility factor *Z* is a dimensionless parameter less than 1.00 that represents the deviation of a real gas from an ideal gas. At low pressures and temperatures Z is nearly equal to 1.00 whereas at higher pressures and temperatures it may range between 0.75 and 0.90. The actual value of Z at any temperature and pressure must be calculated taking into account the composition of the gas and its critical temperature and pressure.

In a given pipe size of fixed cross sectional area were the mass flow rate of the gas is constant, pressure, temperature, and density changes in the gas will modulate the velocity. If the gas flow at some upstream point in the pipe is at a velocity less than Mach 1 (subsonic), the pressure, temperature, and density of the gas can decrease to a point were the velocity becomes Mach 1. It is at this point within the pipe where the flow becomes sonic and any intended means of upstream flow control, *e.g.* pressure relief valve, is rendered ineffectual. This flow condition is said to be *choked*, a term that it is semantically misleading because the flow with this condition is far from being totally obstructed.

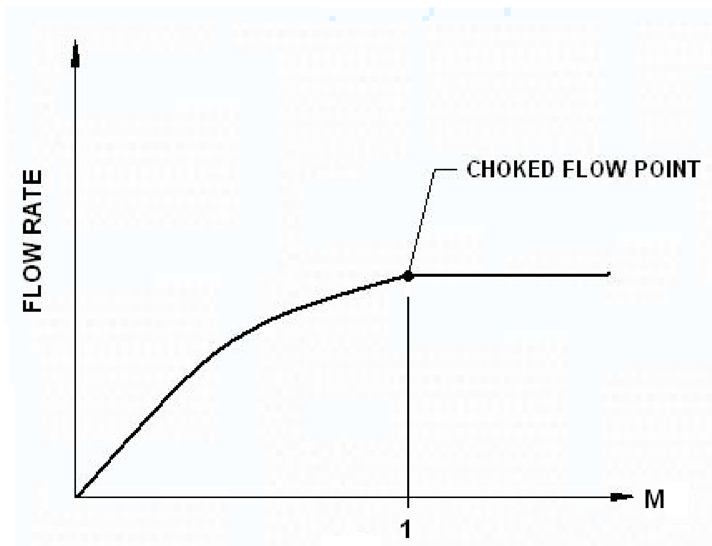


Figure 13 – Sonic Flow

Compressible flow problems can be complex and very difficult to model. For simplification, often times if the anticipated compressible flow pressure and/or temperature changes are small, the flow is analyzed as incompressible. Additionally, transmission of gas in overland pipe lines is generally considered to be carried-out isothermally. In contrast, shorter length compressible gas flows are assumed to occur adiabatically (no heat transfer). Of course in reality flow conditions probably fall somewhere between these two thermodynamic processes.

Owing in great part to the natural gas industry, there is an extensive number of empirical flow equations which are in place which attempt to model compressible flow in pipe. The general gas flow equation rewritten in terms of d is,

$$d = \left[\frac{Q}{8306\sqrt{\dfrac{P_1^2 - P_2^2}{GTLZf}}} \right]^{0.4} \qquad [29]$$

Where the subscripts 1 and 2 denote upstream and downstream conditions, respectively.

Static head of gas column (+ or −) correction factors must be introduced to the general gas flow formula when significant topographical elevation differences exist between the source (inlet) and terminus of an overland gas pipe line.

Important considerations when dealing with gas flow formulae to determine optimum pipe size are:

1. Pressures and temperatures are <u>always</u> taken at their absolute value;
2. Volumetric flow rate is in standard ft^3/day, often times abbreviated SCFD;
3. Gas specific gravity is relative to air at standard conditions;
4. Pipe lengths are in miles in lieu of the customary feet.

According to Menon[19] the velocity in an isothermal compressible gas flow at any point is described by,

$$V = 6 \times 10^{-5} \frac{ZTQ}{Pd^2} \qquad [30]$$

Where Q is in standard ft^3/day

and the limiting erosional velocity for such a flow is given by the empirical formula,

$$V_e = 100\sqrt{\frac{ZTR}{29PG}} \qquad [31]$$

The variable R in Equation [31] is the universal gas constant, not the hydraulic radius.

Equating expressions [30] and [31] and evaluating the constants results in the following useful mathematical relationship to select a pipe size which mitigates gas flow pipe wall erosion:

$$d = 0.001\sqrt{Q}\left(\frac{ZTG}{P}\right)^{0.25} \qquad [32]$$

Example 9 Optimum Pipe Size for Gas Flow with Limiting Velocity

Problem: Natural gas is to be transferred at the rate of 3.8 x 10^8 standard cubic feet per day through 11 miles of transmission pipe line. Assuming the average pressure is 900 lb/in^2 and the flow is carried-out isothermally at 70° F, what pipe size should be employed to minimize erosion to the pipe wall? The compressibility factor and specific gravity for this natural gas have been determined to be 0.85 and 0.60 respectively. Calculate the maximum allowable erosional velocity for these flow conditions and estimate the actual fluid velocity for the selected pipe size.

Given: Q = 3.8 x 10^8 ft^3/day ; L = 11 miles ; P = 900 lb/in^2 ; T = 70° F ; Z = 0.85 ; G = 0.60.

Find: The smallest nominal pipe diameter d for the stated duty which will minimize erosion.

Solution:

① Convert pressure and temperature to absolute values:

$P = 900 + 14.7 = 914.7$ lb/in^2 absolute

$T = 70 + 460 = 530°$ R

② Use the given values and Equation [32] to find that,

$$d = 0.001\sqrt{Q}\left(\frac{ZTG}{P}\right)^{0.25} = 0.001\sqrt{3.8 \text{ x } 10^8}\left[\frac{(0.85)(530)(0.60)}{914.7}\right]^{0.25} = 14.373 \text{ inches}$$

③ From Table 13 on page 77 the closest standard weight nominal pipe size is a:

16 inch NPS schedule 30 with d = 15.250 inches

④ The limiting erosional velocity is found from Equation [31]:

$$V_e = 100\sqrt{\frac{ZTR}{29\,PG}} = 100\sqrt{\frac{(0.85)(530)(10.7316)}{(29)(914.7)(0.60)}} = 55.1 \text{ ft/sec}$$

⑤ The resulting velocity for the selected optimum pipe size is,

$$V = 6 \text{ x } 10^{-5}\frac{ZTQ}{Pd^2} = 6 \text{ x } 10^{-5}\frac{(0.85)(530)\left(3.8\,x\,10^8\right)}{(914.7)(15.250)^2} = 48.3 \text{ ft/sec}$$

⑥ Since $V < V_e$ (48.3 < 55.1), then the selected pipe size is satisfactory.

The Weymouth gas flow equation is the oldest (1912) and most common of the empirical formulas. A variant of its original form, rearranged to allow a direct determination of pipe size is,

$$d = \left[\frac{W^2 G L}{786.8\left(P_1^2 - P_2^2\right)}\right]^{0.1876} \qquad [33]$$

Example 10 Gas Pipe Line Size based on the empirical Weymouth Formula

Problem: Gas is to flow at the rate of 85,000 lb/hr through an overland transmission line. The guaranteed delivery pressure 9½ miles downstream is 20 lb/in^2. The compressor will discharge the gas, which has a specific gravity of 0.80, at 200 lb/in^2. It has been estimated that the average operating temperature will be 60° F. What size pipe should be used for this service?

Given: W = 85,000 lb/hr ; L = 9.5 miles ; P_1 = 200 lb/in^2 ; P_2 = 20 lb/in^2 ; T = 60° F.

Find: A suitable pipe diameter d for the stated utility application.

Solution:
① Convert pressures and temperature to absolute values:

P_1 = 200 + 14.7 = 214.7 lb/in^2 absolute

P_2 = 20 + 14.7 = 34.7 lb/in^2 absolute

T = 60 + 460 = 520° R

② Use the given values and Equation [33] to find that,

$$d = \left[\frac{W^2 G L}{786.8\left(P_1^2 - P_2^2\right)}\right]^{0.1876} = \left[\frac{(85{,}000)^2(0.80)(9.5)}{786.8\left(214.7^2 - 34.7^2\right)}\right]^{0.1876} = 3.970 \text{ inches}$$

③ From Table 13 on page 77 select a **4 inch NPS schedule 40 pipe with *d* = 4.026 inches**

Two-Phase Flow

The majority of two-phase (liquid-gas) flows manifest in patterns known as bubble, plug, slug, wavy, and dispersed. These are shown in Figures 4 and 5. Multiple phase flows, even when restricted to simple pipeline geometry, are in general quite complex, and several features may be identified which make them more complicated than single-phase flow. As shown in the two figures, because of phase density differences, vertical flow patterns are different from horizontal flow patterns, and horizontal flows are generally asymmetric. Even when phase equilibrium is achieved by good mixing in two-phase flow, the changing equilibrium state as the pressure drops with distance, or as heat is added or lost, may require that inter-phase mass transfer, and changes in the relative amounts of the phases, be considered.

Because two-phase flow applications and their flow equations can be so challenging with regard to selecting pipe size directly, an indirect approach is often employed in which a trial pipe size is selected and the corresponding pressure drop is evaluated. This cycle is repeated until a size is found that results in an acceptable energy loss for the required flow rate. Lockhart & Martinelli[20] developed a semi-empirical method that conservatively approximates a two phase pressure drop that is applicable to the four major two phase flow patterns of stratified, wavy, slug, and dispersed. Although still an excellent approximation, the method has been found to under estimate the actual pressure drop for the annular flow pattern. The Lockhart & Martinelli correlation is based on the application of a factor derived from the independent single phase pressure drops of the two phases. This factor is applied to the single phase pressure drop of either phase to arrive at the mixture pressure drop. Like most other analytical exercises, determination of an appropriate friction factor presents the single biggest burden.

The Lockhart & Martinelli method uses normal flow equations to compute common values associated with single phase pressure drop as if each phase was flowing singly in the pipe. The Blackwell[21]/Lockhart & Martinelli method then applies Equations [34], [35], [36], and [37] to arrive at the total mixture pressure drop. The subscripts *l* and *g* refer to liquid and gas respectively:

Two-phase mixture velocity[21]: $$V = 0.0509\,\frac{W_g/\rho_g + W_l/\rho_l}{d^2} \qquad [34]$$

Lockhart & Martinelli two-phase flow modulus[22]: $$X = \sqrt{\frac{\Delta P_l}{\Delta P_g}} \qquad [35]$$

Blackwell correlation of Y_g from Ref. 22: $$Y_g = \left(e^{A_0 + A_1 \ln X + A_2 \ln X^2 + A_3 \ln X^3}\right)^2 \qquad [36]$$

Where, $A_0 = 1.4659$; $A_1 = 0.4914$; $A_2 = 0.0489$; $A_3 = -3.487 \times 10^{-4}$

Lockhart & Martinelli two-phase mixture pressure drop: $$\Delta P = \Delta P_g\, Y_g \qquad [37]$$

The above equations are applicable to the fully turbulent flow regime.

Rearrangement of Equation [34] allows for the computation of a suitable pipe diameter when a mean fluid mixture velocity is known:

$$d = 0.2256\sqrt{\frac{W_g/\rho_g + W_l/\rho_l}{V}} \qquad [38]$$

The best way to illustrate the Blackwell/Lockhart & Martinelli methodology is through a worked-out example:

Example 11 Optimum Pipe Size for Two-Phase Flow with Specified Velocity

Problem: A fluid mixture of gas and liquid must be conveyed at a mass flow rate of 650,000 lb/hr. The weight fraction of the liquid which has a density and viscosity of 33.5 lb/ft^3 and 0.10 cP respectively, is 46%. The gas has a density of 2 lb/ft^3 and a viscosity of 0.01 cP. If a minimum fluid mixture velocity of 66 feet per second is required to preclude damaging slug flow, approximate the probable diameter of pipe that must be installed to meet these conditions. Calculate and evaluate the resulting pressure drop assuming this steel pipe is 100 feet long.

Given: W = 650,000 lb/hr ; W_l = (0.46)(650,000) = 299,000 lb/hr ; ∴ W_g = 351,000 lb/hr ; ρ_l = 33.5 lb/ft³ ; μ_l = 0.10 cP ; ρ_g = 2.0 lb/ft³ ; μ_g = 0.01 cP ; V = 66 ft/sec ; L = 100 feet of steel pipe (ε = 0.00015 feet).

Find: The target pipe diameter d and a corresponding pressure drop for the given process flow.

Solution:

① Using Equation [38] and the given quantities, calculate an inside pipe diameter to be,

$$d = 0.2256\sqrt{\frac{W_g/\rho_g + W_l/\rho_l}{V}} = 0.2256\sqrt{\frac{351{,}000/2 + 299{,}000/33.5}{66}} = 11.93 \text{ inches}$$

② Knowing that $D = d/12 = 11.93/12 = 0.994$ feet, calculate the relative roughness ε /D to be 0.000151.

③ Assume the gas is flowing alone and calculate a Reynolds number[23] of,

$$N_{\mathrm{Re}} = 6.31\frac{W}{d\mu} = 6.31\left[\frac{351{,}000}{(11.93)(0.01)}\right] = 1.86\,\mathrm{x}\,10^7$$

④ Knowing these values of ε /D and N_{Re} , use a Moody chart, internet applet, or any accepted empirical formula to compute a Colebrook friction factor of f = 0.013.

⑤ Calculate the gas pressure drop using a form[23] of the Darcy-Weisbach formula,

$$\Delta P_g = 3.36x10^{-6}\frac{fLW_g^2}{d^5\rho_g} = 3.36x10^{-6}\frac{(0.013)(100)(351{,}000)^2}{(11.93)^5(2)} = 1.1134 \text{ lb/in}^2$$

⑥ Now assume the liquid is flowing alone and repeat Steps ④, ⑤, and ⑥ and calculate:

$$N_{Re} = 1.59 \text{ x } 10^6$$
$$f = 0.014$$
$$\Delta P_l = 0.0515 \text{ lb/in}^2$$

⑦ Next, find the Lockhart & Martinelli two-phase flow modulus using Equation [35]:

$$X = \sqrt{\frac{\Delta P_l}{\Delta P_g}} = \sqrt{\frac{0.0515}{1.1134}} = 0.215$$

⑧ Using this value of X, calculate Perry's flow parameter Y_g from Equation [36],

$$Y_g = \left(e^{A_0 + A_1 \ln X + A_2 \ln X^2 + A_3 \ln X^3}\right)^2 = 5.237$$

⑨ Now compute the mixture pressure drop using Equation [37] and evaluate its suitability:

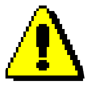
$$\Delta P = \Delta P_g Y_g = (1.1134)(5.237) = \mathbf{5.83\ lb/in^2}$$

Pressure Relief Valve Discharge Pipes

One of the most important pipe size selection exercises is that associated with pressure relief valves. The inlet, and especially the outlet piping of pressure relief valves, is critical to their intended proper performance. The discharging fluid friction in the discharge piping produces back pressure on the relief valve; the larger the pipe diameter, the lower the resulting back pressure.

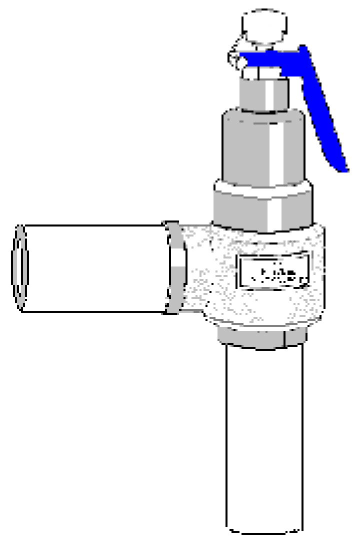
Relief valves are designed to work with back pressure from 10% (conventional valves) to 40% or more (balanced bellows valves) of their set pressure. For example, a conventional valve relieving at 150 psig will work reliably if the back pressure is limited to 15 psig.

The simultaneous discharge from multiple relief valves makes line sizing more complex. Relief discharge pipes sized for protecting tanks against external fires requires the assumption that all valves associated with a manifold will relieve at once.[24] Safety relief systems where the relieving conditions are based on other criteria, *e.g.* runaway chemical reactions, must be analyzed to determine how many devices could conceivably relieve simultaneously. The actual task of pipe sizing can begin once it has been determined which valves will relieve simultaneously.



The relief valve sizing procedure often times necessitates the selection of a relief valve flow orifice which results in a larger relieving flow rate than that required by the process. Pipe sizing must be carried out for the flow rate corresponding to the actual relief valve orifice size selected at set pressure, and not at the originally envisioned required process relieving rate. See PDHcenter.com course no. M112 for a detailed treatment of the selection and sizing of pressure relief valves.

Two-phase flow through pressure relief valves is common-place. Assumption of a complete vapor condition will result in conservative results where multiphase flow is apparent and the total flow rate is known. Going further, assumption of isothermal flow conditions has shown to result in higher pressure drops (more conservative estimates) than an adiabatic process assumption. Actual flow more than likely lies between these two thermodynamic processes. Because two-phase pressure drop prediction models are inherently uncertain, the conservatism is welcomed.



The rapid expansion of a single phase liquid through an actuated pressure relief valve can, and often does, result in flashing flow in the discharge pipe. Contrasted to two-phase flow which denotes steady-state conditions, flashing is the term used to describe the special category of liquid vaporization which involves a fluid's rapid phase change from liquid to vapor without the return of the fluid to the liquid phase.

Relief valve discharge manifolds are used over separate dedicated pipe runs to minimize costs and protect the environment. Pipe size section in relief manifolds is critical to the reliable operation of a safety system. While time consuming, manual pipe sizing is straightforward. Each pipe segment in the manifold network is analyzed based on the flow rates summed from contributing upstream relief devices. It is often necessary to perform the computations using different scenarios. The proper way to size the piping is to proceed in reverse (upstream) from the point where the manifold discharges to a collection element or atmosphere. The best approach is to make estimates for trial pipe sizes based on the equations presented in this course. Detailed calculations are then performed to determine the pressure drop through each pipe segment. The allowable back pressure for each relief valve is compared with the back pressure calculated at the valve. Technical reasoning is used to adjust pipe sizes, larger or smaller, in the network; the calculations are repeated and the procedure continued until all back pressures are satisfactory. Keep in mind that critical (sonic) velocity must not be exceeded in a given pipe segment in order to preclude choking flow.

As in single pipe lines, actual flow in relief valve discharge manifolds also normally takes place somewhere between adiabatic and isothermal conditions. The assumption of an isothermal model is a conservative approach in that isothermal equations yield higher manifold backpressures for both subsonic and sonic flow.[25]

Equation [43] presented on page 60 offers an iterative, manual approach to selection of pipe sizes for flare and relief discharge manifolds. Students who are less interested in the development of the methodology may wish to proceed there now.

Kandell[26] through the combination of sonic and actual gas flow equations and the incorporation of the ideal gas law showed that the Mach number for isothermal flow is given by:

$$M = 1.336 \times 10^{-5} \frac{W}{PA} \sqrt{\frac{ZT}{M_w}} \qquad [39]$$

Knowing that $A = \pi d^2/4$ allows the transformation of Equation [39] to appear as:

$$M = 0.00244951 \frac{W}{Pd^2} \sqrt{\frac{ZT}{M_w}} \qquad [40]$$

which permits the direct insertion of an assumed or selected pipe size. Rearrangement of Equation [40] in terms of d allows for the direct selection of a pipe size with an assumed upper limit for the Mach number:

$$d = \left[\frac{W\sqrt{ZT/M_w}}{408.245 MP} \right]^{0.5} \qquad [41]$$

[with the value of M usually assigned in the range of 0.6 to 0.8]

For isothermal flow is can be shown that: $M_1 P_1 = M_2 P_2$ [42]

An adaptation from Kandell and Lapple[27] shows that for isothermal flow based on upstream conditions:

$$\frac{P_1^2}{P_2^2} = M_2^2\left(\frac{12\,f\,L}{d} + \ln\frac{P_1^2}{P_2^2}\right) + 1 \qquad [43]$$

Where the subscripts 1 and 2 denote upstream and downstream conditions, respectively, for a given pipe segment.

Relief discharge manifold networks contain too many unknowns to allow for a direct solution for pipe size selection. A rigorous design approach requires the procedure known as iteration which has been previously explained. The use of Equation [43] provides an iterative design approach and pipe size selection method for two-phase relief discharge manifolds. This is best illustrated by a worked out example which will follow momentarily. Here is the step-by-step analytical approach to the selection of pipe sizes for flare and relief discharge manifolds where compressible flow is evident and simultaneous valve actuation is probable:

1. Construct an arrangement drawing of the discharge piping arrangement and assign location labels, A;B;C; *etc.* to each relief valve discharge point. Label the pipe intersections as well;
2. Designate on the drawing the known relief flow rates, temperatures, molecular weights, and viscosities at the relief valve discharge points. Creation of a separate data table is also helpful;
3. Based on the style of relief valve and its set pressure, compute the maximum allowable back pressure (MABP) at the discharge point of each relief valve. Add this data to the arrangement drawing and the data table;
4. Calculate the combined flow rates downstream of each pipe intersection and add this data to the drawing and the data table;
5. Estimate the fluid mixture properties downstream of each pipe intersection by the flow rate weighted average method using the following formulas:

$$M_w \approx \sum_{i=1}^{n} W_i \Bigg/ \sum_{i=1}^{n} \left(W/M_w\right)_i \quad [\,44\,]\,, \qquad T \approx \sum_{i=1}^{n} W_i\,T_i \Bigg/ \sum_{i=1}^{n} W_i \quad [\,45\,]\,,$$

$$\mu \approx \sum_{i=1}^{n} \chi_i\,\mu_i\left(\sqrt{M_w}\right)_i \Bigg/ \sum_{i=1}^{n} \chi_i\left(\sqrt{M_w}\right)_i \quad [\,46\,]$$

   Where n represents the number of constituent fluid streams. Add these fluid mixture properties to the drawing and the data table. *Keep in mind that compressible flow equation variables must be evaluated with absolute values;*
6. Begin the analysis at the terminal point (or flare tip) of the manifold, and progress backward. In order to conduct the analysis, a system terminal point pressure must be assumed. This assumption is simplified if the terminal point of the manifold is

vented to atmosphere at sea level: the value of the terminal pressure is 14.7 lb/in^2 absolute.

7. Based on the known (or otherwise determined) terminal point pressure and the estimated fluid properties in the final pipe segment, select a trial pipe size for this segment based on Equation [41]. Use good engineering judgment in the assignment of a value for the Mach number which is appropriate for the process. *Keep in mind that it is desirous to limit fluid velocities such that $M < 1$ (subsonic) in order to preclude excessive vibration and unacceptable noise generation.* In order to introduce a factor of safety, limiting the flow velocity at any point in the manifold corresponding to a maximum allowable Mach 0.6 is advisable. Design values range from 0.6 to 0.8;
8. Using Equation [40] calculate the terminal point Mach number corresponding to the trial pipe size selected. If it is greater than 0.8, then the velocity is too close to sonic and the assumed pipe size is too small;
9. Based on the selected trial pipe size, compute the pipe's relative roughness (ε/D) and the pipe segment Reynolds number using an appropriate equation. In combination these values allow for the determination of the pipe segment's fluid friction factor;
10. Substitute known values into Equation [43] to establish an iterative test equation which will allow for the solution of the upstream or inlet pressure for this pipe segment;
11. Once the inlet pressure is computed, calculate the pipe segment's inlet Mach number using Equation [42]; if it is greater than 0.8, then the velocity is too close to sonic and the assumed pipe size is too small. Select the next larger nominal pipe size and repeat Steps 9, 10, and 11 until an acceptable Mach number is obtained;
12. Working backward, the inlet pressure established in Step 10 should now be thought of as the outlet or downstream pressure for the preceding upstream pipe segment which will be analyzed. Similarly, the inlet Mach number determined in Step 11 now becomes the Mach number at the downstream point of the preceding pipe segment.
13. Repeat this exercise beginning with Step 9, for each pipe segment in the manifold, working backward toward each relief valve, until the pressure at each relief valve has been established;
14. The resulting calculated pressure at each relief valve is the back pressure *seen* by the relief valve. It should be compared to the MABP for that valve. The back pressure must be less than the maximum allowable;
15. If the back pressure exceeds the MABP, a larger pipe size must be selected for the pipe segment under consideration.
16. If there is a significant positive difference between the calculated back pressure and the MABP, the longest pipe segment should be decreased in size until the calculated back pressure approaches the MABP.

The following example illustrates the application of the manual design procedure just outlined. As an alternative to this manual method, commercially available computer programs which utilize looping iteration features to converge on solutions to implicit mathematical relationships handle complex flow problems like this quickly and accurately.

Instructor's Note: While extremely useful, automated computational methods often preclude a clear understanding of the underlying engineering. The example that follows employs a classical method of design which was required prior to the availability of powerful desktop computing resources. It is presented here solely to illustrate the concepts of fluid flow relationships and their importance in selecting pipe size.

Example 12 Selection of Pipe Sizes for a Pressure Relief Discharge Manifold

Problem: To minimize piping installation costs, the pressure relief discharge manifold shown in Figure 14a which ultimately vents out-of-doors, will be constructed in a chemical process plant. The worst case scenario for the process has been determined to be loss of cooling water; this will result in the simultaneous actuation of all four relief valves. Flashing compressible flow discharge conditions are documented. The known fluid properties and relief rates at each valve discharge point are indicated in the figure. Assuming fully turbulent isothermal flow conditions of ideal gases, what size pipe should be used for each segment of the manifold in order to insure that maximum allowable back pressures are not exceeded and subsonic flow conditions are maintained?

Given: The process information shown in Figure 14a and assumption of fully turbulent isothermal relief discharge flow conditions with $Z = 1$. Material: alloy steel (ε = 0.00015 feet). Segmental lengths given are equivalent pipe lengths that account for fittings and entry and exit losses.

Find: The smallest pipe size for each of the indicated pipe segments which will result in acceptable relief valve back pressures and universal subsonic flow.

Methodology: The same analytical sequence presented in the course will be used in this example; however, the numbered steps in the solution that follows do not coincide with the step-wise enumeration in the course. In the interest of brevity, sample calculations are presented in lieu of repetitive computation details. Nevertheless, the final completed computational data and design results for the entire system are indicated in Figure 14b and summarized in Table 12.

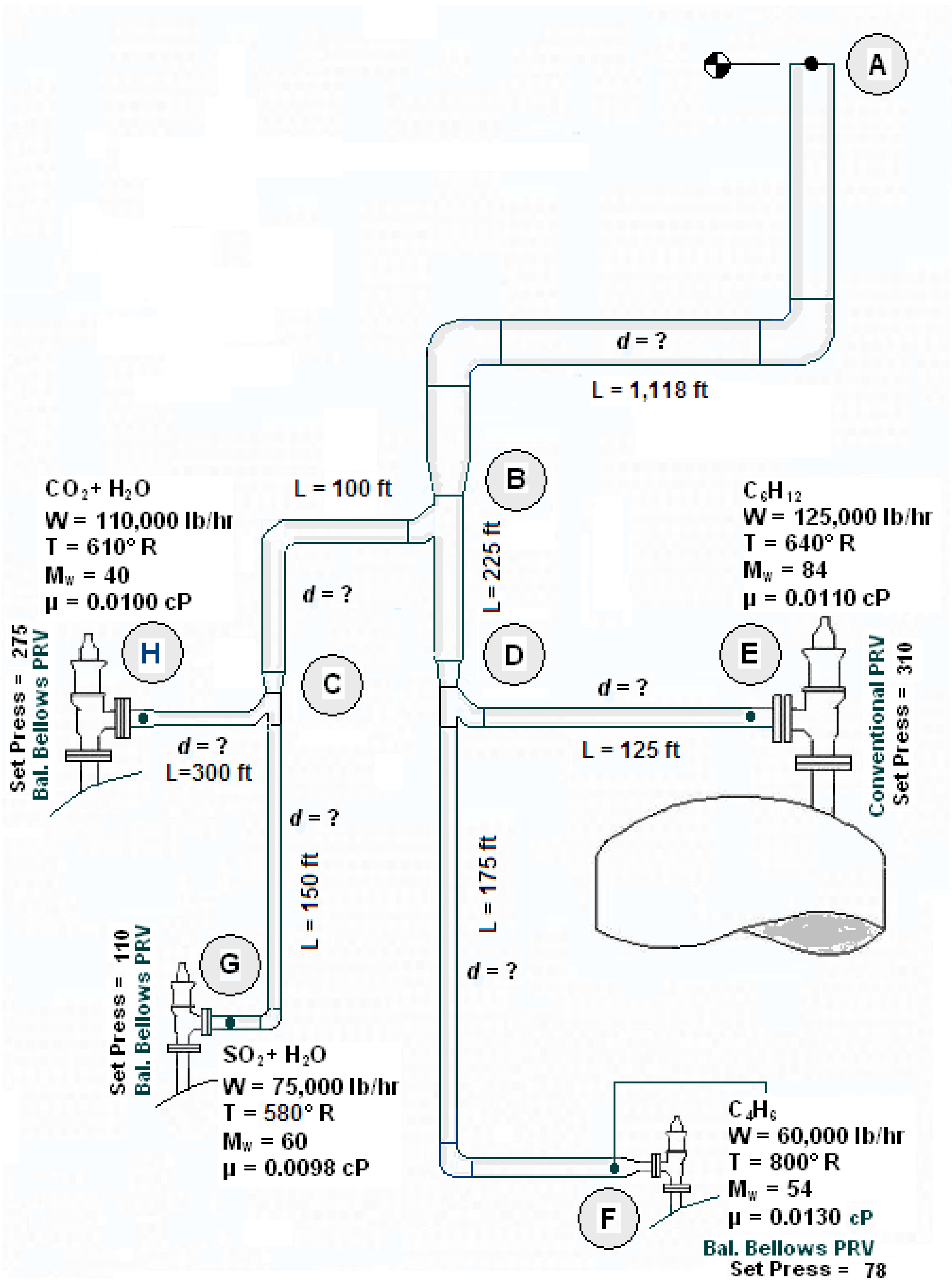


Figure 14a – Relief Discharge Manifold Diagram for Example 12

Solution:

① In relief valve parlance the maximum allowable back pressure (MABP) is equal to a fractional reduction of the valve's set pressure (generally 10% for conventional valves, 40% for balanced bellows valves). Sample Calculation: For the conventional valve at point **E**:

$$\text{MABP} = (310)(0.10) + 14.7 = 45.7\ \text{lb/in}^2\ \text{absolute}$$

Refer to Figure 14b for the calculated values of MABP for valves at points **F, G,** and **H**.

② Sample Calculation: By inspection, the mass flow rate downstream of point **C** will be:

$$W = 75{,}000 + 110{,}000 = 185{,}000\ \text{lb/hr}$$

Refer to Figure 14b for the calculated values of mass flow rate W, downstream of points **B** and **D**.

③ Sample Calculation: The estimated fluid properties downstream of point **C** are:

$$\text{Equation [44]: } M_w \approx \sum_{i=1}^{n} W_i \Big/ \sum_{i=1}^{n} (W/M_w)_i \approx \frac{75{,}000 + 110{,}000}{75{,}000/60 + 110{,}000/40} \approx 46$$

$$\text{Equation [45]: } T \approx \sum_{i=1}^{n} W_i T_i \Big/ \sum_{i=1}^{n} W_i \approx \frac{(75{,}000)(580) + (110{,}000)(610)}{75{,}000 + 110{,}000} \approx 598^\circ\ \text{R}$$

By inspection, the mixture fluid viscosity will be $\mu \approx 0.0099$ cP.

Refer to Figure 14b for the calculated values of fluid mixture properties downstream of points **B** and **D**.

④ By definition, the pressure at point **A** is $P_A = 14.7$ lb/in^2 absolute. Determine a trial pipe size for pipe segment **A→B** based on P_A and Mach 0.6 using Equation [41]:

$$d = \left[\frac{W\sqrt{ZT/M_w}}{408.245MP_A}\right]^{0.5} = \left[\frac{370{,}000\sqrt{(1)(645)/55.8}}{(408.245)(0.6)(14.7)}\right]^{0.5} = 18.69\ \text{inches}$$

From Table 13 on page 77, select a **20 inch NPS schedule 40 pipe with $d = 18.814$ inches**

⑤ Knowing that $D = d/12 = 18.814/12 = 1.568$ feet, calculate the relative roughness ε /D to be 9.57×10^{-5}. Sample Calculation: the Reynolds number for pipe segment **A→B** is:

$$N_{\mathrm{Re}} = 6.31 \frac{W}{d\mu} = 6.31 \left[\frac{370{,}000}{(18.814)(0.0111)} \right] = 1.12 \times 10^{7}$$

Sample Calculation: knowing these values of ε /D and N_{Re}, use a Moody chart, internet applet, or any accepted empirical formula to compute a Colebrook friction factor of $f = 0.01206$.

⑥ Compute the Mach number at point **A** using Equation [40] to be:

$$M_A = 0.00244951 \frac{W_A}{P_A d^2} \sqrt{\frac{ZT_A}{M_w}} = \frac{(0.00244951)(370{,}000)}{(14.7)(18.814)^2} \sqrt{\frac{(1)(645)}{55.8}} = 0.592$$

⑦ Substitute the calculated and known values for pipe segment **A→B** into Equation [43] to develop an expression which can be used to find the value of P_B:

$$\frac{P_1^2}{P_2^2} = M_2^2 \left(\frac{12 f L}{d} + \ln \frac{P_1^2}{P_2^2} \right) + 1$$

$$\frac{P_B^2}{P_A^2} = 0.592^2 \left[\frac{(12)(0.01206)(1{,}115)}{18.814} + \ln \frac{P_B^2}{P_A^2} \right] + 1$$

$$\left(\frac{P_B}{P_A} \right)^2 = 0.3507 \left[8.577 + \ln \left(\frac{P_B}{P_A} \right)^2 \right] + 1$$

Since this expression is implicit in P_B/P_A, use iteration to solve for P_B. Begin the iteration process by arbitrarily assuming that the pressure at point **B** is twice that at point **A**:

$$P_B = 2 P_A \rightarrow P_B / P_A = 2$$

Substitute this trial value for pressure ratio into the test equation and get,

$$(2)^2 = 0.3507 \left[8.577 + \ln (2)^2 \right] + 1$$

$$4 < 4.494$$

Try $P_B/P_A = 2.5$ to compare the equality:

$$(2.5)^2 = 0.3507\left[8.577 + \ln(2.5)^2\right] + 1$$
$$6.250 \quad > \quad 4.651$$

Because the inequality sign changed direction, it is obvious that $2 < P_B/P_A < 2.5$.
Try $P_B/P_A = 2.25$ to observe the directional change, if any, of the inequality sign,

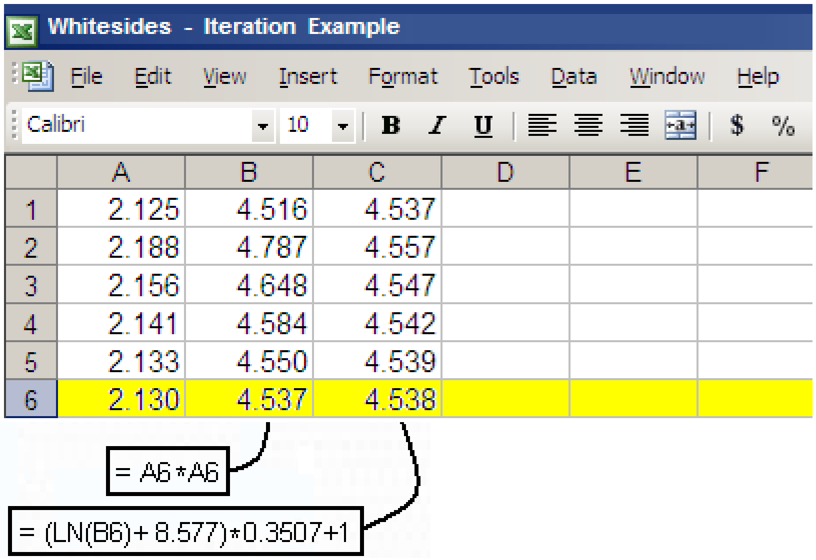
$$5.063 \quad > \quad 4.577$$

No change indicates a lower value for P_B/P_A. See the table that follows for iterative cycles:

TABLE 11 Iterative Cycles for values of P_B/P_A to Test the Inequality of Equation [43] for Example 12

P_B/P_A	*Left side of Equation*	*Sign*	*Right side of Equation*
2.125	4.516	<	4.537
2.188	4.787	>	4.557
2.156	4.648	>	4.547
2.141	4.584	>	4.542
2.133	4.550	>	4.539
2.130	**4.537**	**≈**	**4.538**

As one might imagine, the foregoing exercise could easily be automated somewhat by use of a programmable calculator or spreadsheet:



	A	B	C	D	E	F
1	2.125	4.516	4.537			
2	2.188	4.787	4.557			
3	2.156	4.648	4.547			
4	2.141	4.584	4.542			
5	2.133	4.550	4.539			
6	2.130	4.537	4.538			

The inlet pressure to pipe segment **A→B** is therefore,

$$P_B = (2.130)(14.7) = 31.3 \ \text{lb/in}^2 \ \text{absolute}$$

⑦ Proceeding upstream and in an identical fashion, a pipe size will now be selected for pipe segment **B→C** with P_B now becoming the downstream pressure value.

$$d = \left[\frac{W\sqrt{ZT/M_w}}{408.245 M P_A}\right]^{0.5} = \left[\frac{185{,}000\sqrt{(1)(598)/46}}{(408.245)(0.6)(31.3)}\right]^{0.5} = 9.33 \ \text{inches}$$

From Table 13 on page 77, select a **10 inch NPS schedule 40 pipe with d = 10.020 inches**
For this size,

$$N_{Re} = 1.18 \times 10^7$$
$$\varepsilon/D = 1.79 \times 10^{-4}$$
$$f = 0.01354$$
$$M_B = 0.520$$
$$P_C = 38.8 \ \text{lb/in}^2 \ \text{absolute}$$

⑦ Proceeding upstream and in an identical fashion, a pipe size will now be selected for pipe segment **C→H** with P_C now becoming the downstream pressure value.

$$d = 7.981 \ \text{[8 inch NPS schedule 40]}$$
$$N_{Re} = 8.69 \times 10^{6}$$
$$\varepsilon/D = 2.26 \times 10^{-4}$$
$$f = 0.01419$$
$$M_C = 0.426$$
$$P_H = 59.6 \ \text{lb/in}^2 \ \text{absolute}$$

Since there is a large positive difference between the MABP and the calculated pressure at point **H** (124.7 versus 59.6), try the next smaller nominal pipe size for pipe segment **C→H**:

$$d = 6.065 \ \text{[6 inch NPS schedule 40]}$$
$$N_{Re} = 1.14 \times 10^{7}$$
$$\varepsilon/D = 2.96 \times 10^{-4}$$
$$f = 0.01497$$
$$M_C = 0.737$$
$$P_H = 102.9 \ \text{lb/in}^2 \ \text{absolute}$$

P_H < MABP, therefore use the 6 inch NPS schedule 40 pipe size for pipe segment **C→H.**

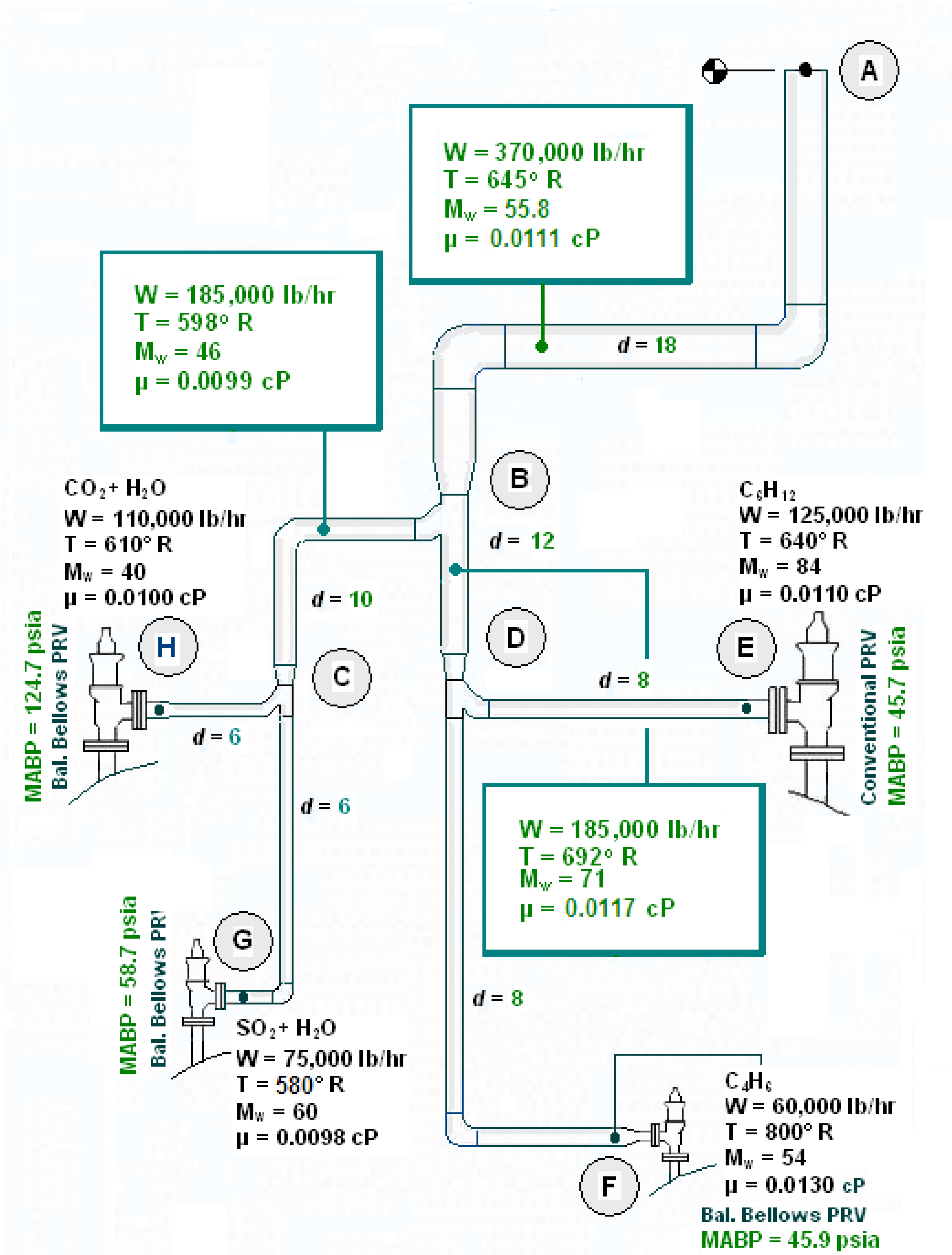
⑧ See Figure 14b and Table 12 for the remainder of the pressure relief manifold's pipe size selections.

TABLE 12 Summary of Final Data for Pressure Relief Manifold Pipe Size Selection - Example 12

Line	**A→B**	**B→D**	**D→F**	**D→E**	**B→C**	**C→H**	**C→G**
NPS	18	12	8	8	10	6	6
Schedule	40	40	40	40	40	40	40
d	18.814	12.000	7.981	7.981	10.020	6.065	6.065
L	1,115	225	175	125	100	300	150
W	370,000	185,000	60,000	125,000	185,000	110,000	75,000
T	645	692	800	640	598	610	580
M_w	55.8	71	54	84	46	40	60
μ	0.0111	0.0117	0.0130	0.0110	0.0099	0.0100	0.0098
MABP	---	---	45.9	45.7	---	124.7	58.7
P_2	14.7	31.3	35.7	35.7	31.3	38.8	38.8
M_2	0.578	0.320	0.249	0.371	0.520	0.737	0.320
N_{Re}	1.12×10^{7}	8.31×10^{6}	3.65×10^{6}	8.98×10^{6}	1.18×10^{7}	1.14×10^{7}	7.96×10^{6}
f	0.01280	0.01312	0.01434	0.01418	0.01354	0.01497	0.01500
P_1	31.3	35.7	40.2	42.7	38.8	102.9	47.8

Calculated values are indicated in black; given values are indicated in red.

Figure 14b –



Summary Diagram for Example 12

Steam Pipes

Water is not only one of the most common substances and indispensable to life, it is also one of the most important media in engineering applications. Steam, the hot gas or vapor phase of water, provides process heating, pressure control, mechanical drive, component separation, and is the source of hot water for many process reactions.[28] Steam systems can be broken down into the four distinct subsystems of generation, distribution, end use, and recovery. Piping transports steam between the subsystems. Optimum pipe size determination in the steam distribution and recovery components can be particularly challenging in that they offer a unique set of problems. The fluid contents of recovery (condensate) piping is a dynamic creature that can have a mixed phase flow of a non-Newtonian compressible fluid. Distribution and condensate systems will be dealt with separately within this course.

Steam distribution piping is generally sized with consideration for pressure drop and fluid velocity. One simplified approach often used is to arbitrarily select a pipe size and then estimate the resulting pressure drop over the length of pipe under analysis, considering steam as an incompressible fluid.

Steam modeled as an incompressible fluid in long lengths of pipe can result in inaccurate results because it assumes that the density remains reasonably constant.

The simplified method utilizes a form of the Darcy-Weisbach formula[23] which has been modified to accommodate the insertion of the customary steam flow and specific volume units of pounds per hour and cubic feet per pound respectively:

$$\Delta P = 3.36 \times 10^{-6} \frac{f L W^2 v}{d^5} \qquad [47]$$

Further simplification of Equation [47] with assumed fully turbulent flow, transformed to allow for the direct determination of steam pipe size results in,

$$d = 1.75 \sqrt{\frac{W v_g x}{V}} \qquad [48]$$

Where x is the steam quality expressed as a decimal fraction.

Saturated steam **is a term used to designate 100% vapor saturation.** ***Dry steam*** **is another term for saturated steam.** ***Steam quality (x)*** **is the proportion of saturated steam in a saturated water/steam mixture (mass fraction of gas phase). A steam quality of zero (0) indicates 100% water while a steam quality of one (1) indicates 100% steam.** ***Wet steam*** **is a term that designates steam with a quality less than one (1).**

Equation [48] is of extreme utility when the recommended steam velocities previously listed in Table 8 on page 42 are felt to be applicable. An example follows:

Example 13 Selection of an Appropriate Steam Pipe Size

Problem: A relatively short length of 200 lb/in^2 steam pipe is to added in a power plant. The line will flow 80,000 lb/hr of steam with a moisture content of 9% w/w. Without regard to pressure drop, what is the probable nominal pipe size that should be installed to limit noise, vibration, and pipe wall erosion?

Given: P = 200 lb/in^2 ; W = 80,000 lb/hr ; wet steam.

Find: A nominal pipe diameter d for the stated power plant utility application.

Solution:

① Without more information, the operating pressure of 200 lb/in^2 will be considered high pressure steam for the assumption of a suitable fluid velocity. From Table 8, select a median flow velocity of 8,000 ft/min.

② The stated 9% moisture content translates to a steam quality of $x = 1 - 0.09 = 0.91$

③ Referring to any of a number of available steam tables such as Reference 29 (Keenan & Keyes), find that the specific volume of saturated steam at 200 lb/in^2 gage pressure (214.696 lb/in^2 absolute) is v = 2.137 ft^3/lb.

④ Using Equation [48], the pipe size to limit noise, vibration, and erosion is,

$$d = 1.75\sqrt{\frac{W v_g x}{V}} = 1.75\sqrt{\frac{(80{,}000)(2.137)(0.91)}{8{,}000}} = 7.717 \text{ inches}$$

⑤ From Table 13 on page 77 select an **8 inch NPS schedule 40 pipe with d = 7.981 inches**

Once steam has done its work of surrendering latent heat, it condenses to the liquid phase referred to as steam condensate. Condensate retains residual useful energy and its physical recovery is important for efficient operation. It is *returned* to the generation unit by means of pumping of the hot liquid or through pipe lines at reduced pressures. Pumped condensate return lines in properly engineered systems only carry water and as such their pipe size can be determined by the conventional incompressible flow methods previously outlined. It is in the low pressure return lines, often referred to as *headers*, where pipe sizing for condensate recovery takes on a new dimension: two-phase flow brought about by the larger effective volume and reduced pressure of the fluid. When hot condensate under pressure is released to a lower pressure, its temperature drops rapidly to the saturation temperature of that lower pressure. The sensible heat released from the liquid condensate causes some re-evaporation into flash steam, and the two phases coexist within the pipe. Neglecting the presence of flash steam yields undersized pipe sizes that result in increased back pressure and velocity, with a liquid phase that could lead to water hammer. Intense water hammer can cause damage to personnel, pipes, and equipment. Consequently, the sizing exercise must be one which provides some type of two-phase-flow correlation.

It has been shown that most flow patterns for condensate return pipe lines lie within the annular region[30] although on some occasions they may fall within the dispersed, or even slug-flow regions. Generally, the flashed steam constitutes less than 30% w/w of the fluid mixture. Ruskin[31] developed a method for calculating pressure drop of flashing condensate which is based on the assumption of a single homogenous phase of fine liquid droplets dispersed in the flashed vapor. This method provides pressure drops that are comparable to those obtained by more rigorous methodologies. Using Ruskin's method, Blackwell[32] developed the following flashing correlation equations based on the assumption that the vapor-liquid mixture throughout the steam/condensate header is represented by the mixture for end-pressure conditions. The subscripts l and g refer to liquid and gas, respectively; the subscripts 1 and 2 denote upstream (higher) pressure and downstream (low or header) pressure, respectively. See Figure 15.

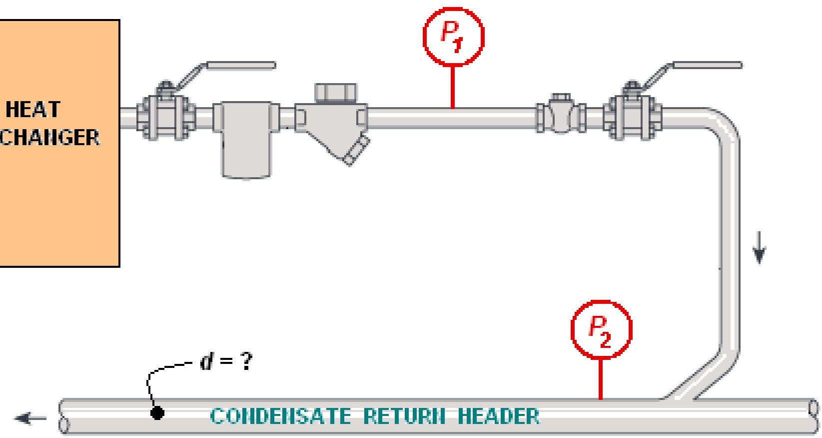


Figure 15 – Relationship between High and Low Pressure Condensate Lines

The density of the condensate can be correlated to a cubic form in terms of the header pressure:

$$\rho_l = -1.3 \times 10^{-6} P_2^3 + 4.8 \times 10^{-4} P_2^2 - 0.078 P_2 + 60.827 \qquad [49]$$

The density of the flash steam can be correlated to the header pressure by:

$$\rho_g = 0.0029 P_2^{0.938} \qquad [50]$$

The steam quality of the two-phase mixture is approximated by the expression,

$$x = [\ln P_1]^2 \left[0.0001 e^{6.122-(16.919/\ln P_2)} + 0.0088\right] - \left[0.00671 (\ln P_2)^{2.27}\right] \qquad [51]$$

Rearranging the original Blackwell[32] velocity equation allows for the computation of a suitable pipe diameter when a mean fluid mixture velocity is known:

$$d = 1.7476\sqrt{\frac{W_g/\rho_g + W_l/\rho_l}{V}} \qquad [52]$$

The recommended maximum velocity of flash steam/condensate return lines to limit physical pipe deterioration is 5,000 feet per minute with optimum velocities in the range of 3,000 to 3,600 feet per minute. A worked example which illustrates the use of these equations to determine an optimum pipe size for a condensate return header which is expected to have flashing flow follows:

Example 14 Selection of an Appropriate Steam Condensate Return Pipe Size

Problem: A low pressure steam condensate return header which is designed to operate at 10 lb/in^2 will receive 10,000 lb/hr of 100 lb/in^2 condensate from a process equipment steam trap arrangement similar to that shown in Figure 15. Flash steam generation in the header from sensible heat re-evaporation is expected. Approximate the probable header pipe size that should be installed based on normally recommended fluid velocities.

Given: P_1 = 100 lb/in^2 ; P_2 = 10 lb/in^2 ; W = 10,000 lb/hr ; flashing conditions.

Find: A nominal pipe diameter d for the return header portion of the system.

Solution:

① Convert the pressures to absolute values:

P_1 = 100 + 14.7 = 114.7 lb/in^2 absolute

P_2 = 10 + 14.7 = 24.7 lb/in^2 absolute

② Estimate the density of the liquid phase in the header using Equations [49],

$$\rho_l = -1.3\,\mathrm{x}\,10^{-6} P_2^3 + 4.8\,\mathrm{x}\,10^{-4} P_2^2 - 0.078\,P_2 + 60.827$$

$$\rho_l = -1.3\,\mathrm{x}\,10^{-6}(24.7)^3 + 4.8\,\mathrm{x}\,10^{-4}(24.7)^2 - 0.078(24.7) + 60.827$$

$$\rho_1 = 59.17\ \mathrm{lb/ft^3}$$

③ Estimate the density of the vapor phase in the header using Equations [50],

$$\rho_g = 0.0029 P_2^{0.938} = (0.0029)(24.7)^{0.938} = 0.0587 \text{ lb/ft}^3$$

④ Approximate the steam quality of the two-phase fluid in the header using Equation [51],

$$\begin{aligned} x &= [\ln P_1]^2 \left[0.0001 e^{6.122-(16.919/\ln P_2)} + 0.0088\right] - \left[0.00671(\ln P_2)^{2.27}\right] \\ x &= [\ln 114.7]^2 \left[0.0001 e^{6.122-(16.919/\ln 24.7)} + 0.0088\right] - \left[0.00671(\ln 24.7)^{2.27}\right] \\ x &= 0.1044 \end{aligned}$$

⑤ Calculate the mass flow rate of each phase in the two-phase fluid:

$$\begin{aligned} W &= 10{,}000 \text{ lb/hr} \\ W_g &= xW = (0.1044)(10{,}000) = 1{,}044 \text{ lb/hr} \\ \therefore \quad W_l &= W - W_g = 10{,}000 - 1{,}044 = 8{,}956 \text{ lb/hr} \end{aligned}$$

⑥ Select a suitable fluid velocity based on the recommended range of 3,000 to 3,600 ft/min stated in the course: $\rightarrow V = 3{,}300$ ft/min.

⑦ Determine a optimum pipe size from Equation [52] based on the calculated data,

$$d = 1.7476 \sqrt{\frac{W_g/\rho_g + W_l/\rho_l}{V}} = 1.7476 \sqrt{\frac{1{,}044/0.0587 + 8{,}956/59.17}{3{,}300}} = 4.074 \text{ inches}$$

⑧ From Table 13 on page 77 select a **4 inch NPS schedule 40 pipe with *d* = 4.026 inches**

Special Pipe Size Design Criteria

Various industry-specific, non-mandatory special pipe sizing criteria, are often adopted by local building codes and thus become regulatory. A good example of a widely adopted method is that of the Hazen-Williams formula in the fire protection industry. It is used to calculate pipe size based on pressure differential, length of pipe, and pipe material roughness.

$$d = \left[\frac{4.52\, L\, Q^{1.85}}{\Delta P C^{1.85}} \right]^{0.205} \qquad [53]$$

Where Q is in gallons per minute

The Hazen-Williams formula, written in its original form [NFPA 13 §6-4.2.1],

$$\Delta P = \frac{4.52\, Q^{1.85}}{C^{1.85} d^{4.87}} \qquad [54]$$

is more commonly used for pressure drop calculations in sprinkler systems, water distribution systems, and irrigation systems. The variable C is a friction loss coefficient which is dimensionless - the higher the C factor, the smoother the pipe. Typical C values used in design, which take into account some increase in roughness as pipe ages, range from concrete and cast iron at C =100 to C = 150 for PVC and FRP.

While similar in value, the variable C of Equation [53] should not be confused with the variable c in the API RP 14E formula, Equation [25].

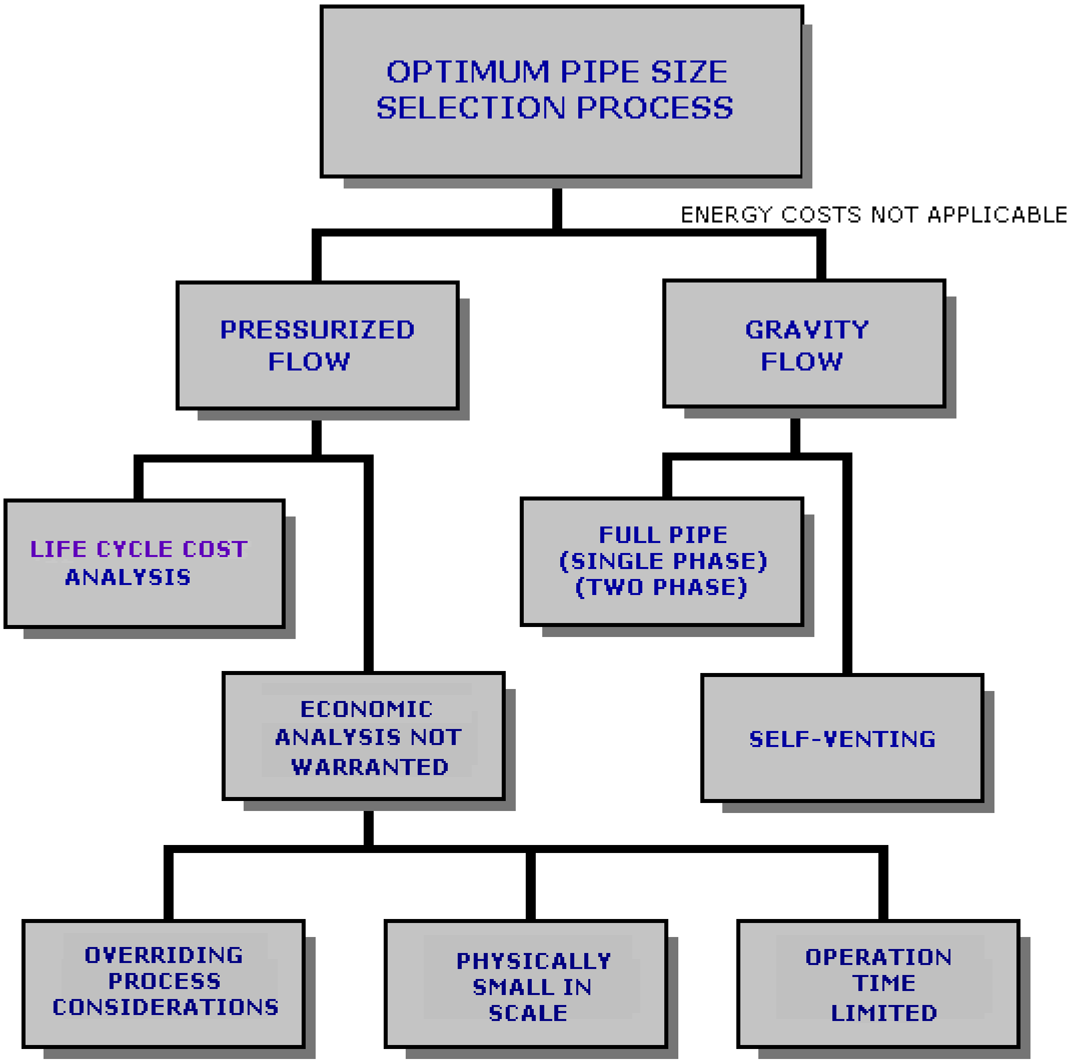
The Hazen-Williams formula is empirical and lacks physical basis. The formula's application is severely limited to *normal* conditions of water flowing at moderate temperatures (40° F < T < 75° F) with a mean fluid velocity $V \approx 3$ ft/sec, $N_{Re} > 10^5$. Even the use of the formula with hot water which exhibits a lower viscosity, can result in significant error. The Darcy-Weisbach method should be used for other liquids or gases.

TABLE 13 Inside Pipe Diameter of NPS Pipe

Summary

Nominal Size	Schedule	d	Nominal Size	Schedule	d
1	10	1.097	14	10	13.624
	40	1.049		30	13.250
	80	0.957		60	12.814
1½	10	1.682	16	10	15.500
	40	1.610		30	15.250
	80	1.500		60	14.688
2	10	2.157	18	10	17.500
	40	2.067		30	17.124
	80	1.939		60	16.500
2½	10	2.635	20	10	19.500
	40	2.469		30	19.000
	80	2.323		60	18.376
3	10	3.260	22	10	21.500
	40	3.068		ST	21.250
	80	2.900		XS	21.000
4	10	4.260	24	10	23.500
	40	4.026		20	23.250
	80	3.826		40	22.626
6	10	6.357	26	ST	25.250
	40	6.065		XS	25.000
	80	5.761		-	24.500
8	10	8.329	28	ST	27.250
	40	7.981		XS	27.000
	80	7.625		-	26.500
10	10	10.420	30	ST	29.250
	40	10.020		20	29.000
	80	9.564		30	28.750
12	10	12.390	32	ST	31.250
	40	12.000		XS	31.000
	80	11.750		-	30.500

This course is best summarized by the block diagram shown below:



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