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1.0 SCOPE

This guideline is not meant to be a rigid set of rules governing the manner in which the calculations are done; it is meant as a means of improving the quality of work and of minimizing the likelihood that a significant error is made. This guideline is meant to insure that engineering judgment and thought is put into the calculations.

Pressure drop and velocity limits for specific services are given in this document followed by guidelines for hydraulic circuits. Finally, a discussion of two phase flow concludes this guideline.

2.0 RESPONSIBILITIES

Procedures PTD-PRO-105 ("Procedure Requirement for Checking Process Technology Work.") and PTD-PRO-107 ("Process Calculations") should be followed for all hydraulic calculations.

The process engineer doing the initial work associated with the hydraulic calculations has the responsibility to ensure that the work is done properly. Although all work must be checked by a second engineer, this does not in any way reduce the obligation of the primary process engineer to do the calculations properly the first time.

The checking engineer has the responsibility to review all calculations performed by the responsible engineer and must resolve all perceived discrepancies. After all discrepancies have been resolved to the checking engineer's satisfaction, this second engineer must sign off all calculations to indicate concurrence.

In the event that discrepancies cannot be resolved between the two engineers, a third supervisory engineer should be consulted to resolve the dispute.

3.0 ABBREVIATIONS

Unless otherwise noted, all abbreviations and symbols used in this guideline are defined as follows:

btu:	British thermal unit
c_p:	fluid specific heat, in btu/pound ^{o}F
cs:	centistokes
d:	internal diameter of the pipe, in inches
	impeller diameter, in inches
d_n:	nominal diameter of the pipe, in inches
F_c:	piping complexity factor
fps:	feet per second
ft^3:	cubic feet
g:	acceleration of gravity, 32.2 fps

gpm: gallons per minute
H: total head, in feet of fluid
k: ratio of specific heat at constant pressure to specific heat at constant volume
L: linear length of piping, feet
lb: pound
L_{eq}: equivalent length of piping, feet
MW: molecular weight
m: mass of fluid, lb
NPSH: net positive suction head
OSBL: outside battery limits
P: pressure, in pounds per square inch gauge
P': pressure, in pounds per square inch absolute
P&ID: piping and instrument drawing
PFD: process flow diagram
psi: pounds per square inch
Q: volumetric flow rate, gpm
R: individual gas constant = 1544 / MW
scf: standard cubic feet
t: temperature, in ^{o}F
T: absolute temperature, in ^{o}Rankine (460 + t)
v: velocity, in feet per second
v_S: sonic velocity, in feet per second
W: mass flow rate, pounds per hour
W.C.: Water Column
Z: compressibility factor

Subscripts

A: available
AVG: average
L or l: liquid
MAX: maximum
MIN: minimum
mix: mixture
R: required
v: vapor

Greek Characters

Δ: delta, differential between two points
ρ: rho, density at flowing temperature and pressure, in pounds per cubic foot

4.0 GLOSSARY OF TERMS

Equivalent Length (of a valve or fitting): The length of straight pipe that would give the same pressure drop as a valve or fitting of the same nominal diameter under the same flow conditions.

Liquid Holdup: Ratio of the volume of liquid in a pipe segment to the total volume of that pipe segment. The volume fraction of the pipe segment occupied by liquid. Non-slip liquid holdup is the ratio of the liquid volumetric flow rate to the total combined volumetric flow for the stream. The difference between the non-slip liquid holdup and liquid holdup is the degree of slippage between the liquid and vapor phases.

Net Positive Suction Head Available ($NPSH_A$): The total suction head in feet of liquid (absolute at the pump centerline or impeller eye) less the absolute vapor pressure (in feet) of the liquid being pumped.

Normal Flow Rate: The flow rate at which the pump is expected to operate. This flow normally corresponds to the value found in the heat and material balance on the Process Flow Diagram.

Process Engineer. The term "Process Engineer" is used to denote the department, group, or engineer who performs line sizing calculations. The term "Process" could refer to Process Systems Engineers or Mechanical Systems Engineers as well as Process Engineers.

Control Systems. The term "Control Systems" is used to denote the department, group, or engineer who handles the instrumentation. The words "Instrumentation or Instrument" are often substituted for "Control Systems".

Slip Ratio: The ratio of vapor velocity to liquid velocity in a two-phase flow line. A homogeneous mixture would have a slip ratio of 1.0.

Sonic or Critical Velocity: The maximum possible velocity that a gas or gas-liquid mixture can attain in a pipe at a given upstream pressure, regardless of the discharge (or destination) pressure. For gases, this velocity is equal to the sonic velocity calculated as:

$$V_s = (kgRT)^{1/2} \quad \text{(fps)}$$

$\Delta P/100'$: Pressure loss per 100 feet of pipe is a common line sizing criteria. This quantity is expressed in psi throughout the guideline.

5.0 GENERAL DISCUSSION

The pipe sizing criteria presented should be used in the design and installation of new systems. The economic criteria for revamping existing systems is different than that used to design and build new installations. Therefore, although these criteria should be used to evaluate existing installations it should not be used as the ultimate test in determining whether or not piping or equipment should be replaced. However, whenever there are significant deviations from these guidelines, e.g., excessive fluid velocities or pressure drops in lines, they should be noted for client review and discussion.

5.1 Basis

General line sizing criteria is based upon normal flow rates unless otherwise noted. Pressure drop criteria are presented in psi per 100 equivalent feet (ΔP/100') and velocity criteria are presented in feet per second (fps). In all cases total pressure drop available is a primary consideration in line sizing. If designing per one of the following pressure drop guidelines results in inadequate pressure at the fluid destination, obviously a larger pipe is required.

5.2 Branching Lines

Unless covered by other specific guidelines, the minimum sum of the transverse internal areas of the branching lines should be equal or greater than the transverse internal area of the main line.

5.3 Sonic Velocity, Vapor

In sizing certain lines, such as vacuum heater transfer lines or flare header lines, limitations due to sonic velocity must be observed.

In a pipe of constant diameter, a compressible fluid cannot move faster than the speed of sound in that fluid. Many correlations do not warn of violations to this fact and the engineer, if not aware of this possibility, may calculate a frictional pressure drop less than that which actually exists. With a high-velocity vapor stream, whenever the calculated pressure drop approaches or exceeds the following values of ΔP as a percentage of upstream pressure, the possibility of sonic flow should be investigated:

Diatomic Gases:	$\Delta P = 47\ \%$
Heavier Gases:	$\Delta P = 45\ \%$
Saturated Steam:	$\Delta P = 42\ \%$

In situations where it is determined that the sonic velocity has been exceeded, the downstream pressure in the pipe will actually rise until the increased gas density permits the flow to proceed at exactly sonic velocity.

$$V_s = (kgRT)^{1/2} \quad \text{(fps)}$$

5.4 Maximum Velocity

Max velocity should not exceed erosional velocity.

$$V_e = \frac{c}{\sqrt{\rho_m}}$$

c =100 for continuous service
c =125 for intermittent service

NOTE: c must be reduced if solids are present. c may be increased if specific application studies have shown it appropriate.

The homogenous mixture density is:

$$\rho_m = \frac{m_V - m_L}{\frac{m_V}{\rho_V} + \frac{m_L}{\rho_L}}$$

For additional information, refer to API Recommended Practice 14 E.

6.0 LINE SIZING CRITERIA

The following sizing guidelines are based upon both economic and operating criteria. Clients may have alternate guidelines, which supersede the ones listed below. Nevertheless, even in those circumstances it is recommended that these guidelines be consulted. Past experience indicates that these guidelines will provide adequate operating flexibility without significant impact on installation or operating costs.

6.1 Gravity Flow Lines

The following guidelines are for general services normally encountered in process piping. Refer to PTD-DGS-121 “Gravity Fluid Flow Design Guide” for more specific information on designing gravity flow piping systems.

A. General Drains and Draw-Offs

Guideline: $(\Delta P/100')_{MAX} = 0.20$ psi

Two flow regimes exist within drain piping. Siphon flow causes pressure pulsations and should be avoided. Self venting flow, the desirable gravity flow condition, allows any normally entrained vapors in the pipe leg to disengage and move freely apart from the liquid. The following correlation is taken from reference (Simpson, Larry L. and Martin L Weirwick, “Designing Plant Piping” Chemical Engineering: Deskbook Issue, April 3, 1978).

Guideline: $d = 0.92\ Q^{0.4}$ (min)

B. Gravity Flow to Kettle or Thermosyphon Reboiler (non-pumped)

Generally, the heat exchanger specialists will conduct the hydraulic calculations and determine the piping design requirements for the reboiler system. Nevertheless, the following guidelines can still serve as a reference.

Guideline: $(\Delta P/100')_{MAX} = (0.0062)\,(\rho_l - \rho_v) - 0.06$ (psi)

where ρ_l = density of liquid at bottom of column, lb/ft^3
ρ_v= density of vapor in reboiler return line, lb/ft^3

This line and the reboiler return line should be sized at the same time to insure hydraulic integrity of the reboiler circuit. Detailed hydraulics of the overall reboiler system must still be reviewed.

C. Gravity Flow to Pump Suction, stream temperature ≤ 50 ^{0}F below bubble point

Guideline: $(\Delta P/100')_{MAX} = 0.20$ psi
V_{MAX} Centrifugal Pumps = 2-3 fps
V_{MAX} Reciprocating Pumps = 1-2 fps

Exception: For streams carrying solids, such as catalyst fines: $v_{MIN} = 3.0$ fps

Following this minimum velocity requirement will sometimes result in sizing lines which exceed the pressure drop guideline; in those instances choose the largest line size which will achieve the minimum velocity requirement. Slower velocities will result in the settling of solids.

D. Gravity Flow to Pump Suction, stream temperature ≥ 50 ^{o}F below bubble point

Guideline: $(\Delta P/100')_{MAX} = 0.35$ psi

Caution: Beware of viscosity changes for heavy liquids that could cool in the suction line. A higher viscosity can affect the pressure drop and subsequently, the $NPSH_A$.

6.2 Pump Discharge Lines

A. Pump Discharge Line, immediately downstream of reducer

Guidelines: $(\Delta P/l00')_{MAX}$ = 4.0 psi
v_{MAX} =11.7 fps

Both guidelines must be met.

B. Pump Discharge Line, downstream branches

Treat the same as Section 6.2 A; however, use caution in sizing low-capacity branches. A high pressure drop in such a line might force a large-capacity pump to be sized for this higher head when the main flow does not require the higher discharge pressure. In such cases it may be economical to over-size the lower-capacity branch.

C. Pump Discharge Line, downstream branches at alternate conditions

Treat the same as Section 6.2 B. Use caution in sizing low-capacity lines (see Section 6.2 C).

D. Reboiler Line

For fired reboiler heater transfer lines, when minimum furnace outlet process temperature is desired:

$(\Delta P/100')_{MAX}$ = 2.5% of column bottom absolute pressure, psia.

Caution: If the stream is in vertical upward two-phase flow, such as feed to a column, check for slug flow both at normal flow and turndown. The complete system pressure drop needs to be calculated including the heater ΔP.

6.3 General Gas Lines, inside battery limit

Guideline:

Operating Pressure, psig	$(\Delta P/100')_{MAX}$, psi
<0	0.25
0 to 100	0.50
100 to 500	1.00
500 to 1000	2.00
>1000	0.002P

where P = operating pressure, psig

Caution: V_{MAX} should be 60-80 fps to minimize noise.

6.4 Cooling Water

A. Headers

$(\Delta P/100')_{MAX}$ = 7.0 psi

v_{MAX} = 11.7 fps

B. Branches

Flow Rate				
0 - 650	gpm:	$(\Delta P/100')_{MAX}$	=	1.3 psi
651 - 2800	gpm:	v_{MAX}	=	8.3 fps
2801 - 7000	gpm:	v_{MAX}	=	10.0 fps
7001 +	gpm:	v_{MAX}	=	11.7 fps

0 - 650 gpm:	$(\Delta P/100')_{MAX}$	= 1.3 psi	
651 - 2800 gpm:	v_{MAX}	= 8.3 fps	
2801 - 7000 gpm:	v_{MAX}	= 10.0 fps	
7001 + gpm:	v_{MAX}	= 11.7 fps	

Caution: Note that actual pressure drops in fouled systems can significantly exceed the 1.3 psi/100' guideline.

6.5 Lines with High-Available ΔP

(≥ 25 psi for piping friction loss, within the battery limit)

Guidelines: The general guideline is to use up but not exceed the available ΔP while still observing the following constraints (not applicable to flare headers):

For 100% Liquid: $(\Delta P/100')_{MAX}$ = 7.0 psi
v_{MAX} = 11.7 fps

For 100% Vapor: $(\Delta P/100')_{MAX}$ = 7.0 psi
$v_{MAX} = 100 / \rho_v^{1/2}$ (fps)

where ρ_v = density of vapor at flowing temperature and pressure, lb/ft^3

With CO_2 concentrations above 1%, limit the maximum velocity to 30-50 fps.
Minimum gas velocity is 10-15 fps to prevent liquids gathering in low points.

Caution: Noise Specifications may limit gas velocity in non-emergency piping to 60-80 fps.

For Two-Phase: $(\Delta P/100')_{MAX}$ = 5.0 psi
$V_{MAX} = 100 / \rho_{mix}^{1/2}$ (fps)

Caution: For flashing streams, it is imperative that vaporization be taken into account in the calculation of the pressure drop as the absolute pressure decreases.

Noise Specifications may limit two-phase velocity in non-emergency piping.

6.6 Steam Lines

A. Steam Lines, general

Guidelines:

Saturated Steam, ≤ 15 psig:	$(\Delta P/100')_{MAX}$	=	0.11 psi
	v_{MAX}	=	100 fps

If allowable pressure drop is available, these limits can be increased nevertheless, the velocity should be kept below 170 fps.

Saturated Steam,> 15 psig to 250 psig:	$(\Delta P/l00')_{MAX}$	= (0.013)P' (psi)
	v_{MAX}	= 170 fps
Saturated Steam, > 250 psig:	$(\Delta P/100')_{MAX}$	= (0.020) P' (psi)
	v_{MAX}	= 170 fps
Superheated Steam, ≥ 200 psig:	$(\Delta P/100')_{MAX}$	= (0.012) P' (psi)
	v_{MAX}	= 200 fps

B. Condensing Steam Turbine Exhaust Line

The exact configuration of this line must be accurately known to determine its size. The operating pressure of the turbine exhaust and the condenser inlet must also be known. For simplicity, the calculations should be based on the entire flow from the turbine assuming steam quality of 100%. Size the line in order to use the differential pressure between the turbine and the condenser.

Guideline:

$v_{MAX} = 20 / \rho_v^{1/2}$ (fps)

where ρ_v = the density of steam at the inlet to the condenser, lb/ft^3

C. Atomizing Steam to Heaters

If available, use burner vendor recommended values for flow rates. In lieu of vendor information, estimate the steam requirements at 3-4 pounds of steam per gallon of fuel oil burned. Estimate the line pressure at 75 psig (downstream of the valve, if present).

Guideline: Use the same sizing criteria as that for Steam Lines (Section A above).

6.7 Steam Condensate Lines

There are numerous operating schemes possible in the operation of condensate removal piping. The following discussions highlight the major concerns, which should be considered by the process engineer.

A. Steam Condensate Lines, upstream of control valve or steam trap

Guideline: v_{MAX} = 3.0 fps

B. Steam Condensate Return Lines, downstream of control valve or steam trap

General
To size the condensate line the process engineer must first determine the destination requirements (pressure and elevation), and the initial condensate conditions (temperature and pressure).

The initial condensate conditions can depend upon numerous variables: the process-side temperatures, the temperature approach to the process side, the exchanger fouling, the capacity utilization of the exchanger, and the steam control strategy.

Steam Regulation
If the control valve is regulating the steam flow to the exchanger, it is imperative that the process engineer review the possible operating scenarios to insure that the condensate will have sufficient pressure to flow to its destination. The condensate temperature must be high enough such that the saturation pressure is sufficiently above the condensate destination pressure to permit flow. Lower condensate temperatures can be permitted when flow is to a condensate drum operating under a vacuum.

It is important that the process engineer review turndown scenarios. When the unit is operating at turndown conditions, heat exchangers are oversized. Therefore, the condensate condensing temperature is lowered and can approach the process-side temperature. Low temperature condensate will have less flashing after the steam trap. Nevertheless, low condensate temperatures will often govern for condensate line sizing because those operating scenarios have low upstream pressures and subsequently, the available ΔP is less than for other cases.

Condensate Pot
If a level-controlled condensate pot is used in lieu of a steam trap the pot can be placed so as to permit partial flooding of the exchanger. This would permit greater operating flexibility for turndown situations and provide better operating stability. During turndown, the condensate level is maintained in the exchanger to partially block off surface area. The steam subsequently condenses at a higher pressure, which allows the condensate to flow to its destination.

When the condensing pressure of the condensate is too low to permit flow to its destination, a pump or pressured-powered pump (pumping trap) may be required.

Condensate Regulation
If the control valve is placed in the condensate line, the pressure before the control valve will be full steam pressure less the exchanger pressure drop (usually small). The temperature of the condensate can vary from header saturation temperature to very close to process-side temperature. For condensate line sizing, the higher temperature will be governing due to the increased flashing for this case. The assumption that the condensate temperature is equal to steam saturation temperature would be conservative with respect to line sizing.

Pressure Drop Guideline
For the pressure drop calculation it is recommended that the line be broken into incremental sections in order to account for the increased flashing as the pressure

drops. It is recommended that the Parsons Steam Program be used to calculate the pressure drop in the condensate line downstream of the steam trap or control valve.

The following general guidelines apply:

Guidelines: The total pressure drop due to pipe frictional losses should not account for more than 50% of the total available pressure drop. Pressure drop per length of pipe will be greatest close to the destination point where the pressure level is the lowest. It is here that more liquid will have flashed to vapor and the vapor density will be the lowest. The guideline with respect to instantaneous pressure drop is:

$(\Delta P/100')_{MAX} = 7.5$ psi

In calculation of the pressure drop, use a safety factor of 1.5 times the calculated value for clean, commercial steel pipe. This will compensate for the corrosive action of the condensate, which will roughen the piping walls.

If the length of the condensate line is unknown, use 300 equivalent feet for the distance from the valve or trap to the condensate header or drum.

6.8 Fuel Oil to Heaters

Fuel oil temperature should be regulated to achieve a viscosity of about 50 cs. At that temperature, it is assumed that the flowing gravity will be 0.9. Generally, the oil recirculation rate will be about three times the consumption rate. With those comments in mind, the following should be used to size lines.

Guidelines:

FlowRate, gpm	Line Size
0 to < 6	1” Sch 80
6 to <20	1.5” Sch 80
20 to <35	2” Sch 80
35 to <100	3” Sch 40
100 to 300	4” Sch 40

The same criteria should be used for the fuel oil return lines to storage, surge drums, or refinery fuel oil system.

6.9 Fuel Gas to Heaters

Determine the amount of gas to the heater based on:

1) the process specification for the heater duty
2) the heater efficiency
3) the fuel gas heating value.

Guidelines:

Before the Control Valve: $(\Delta P/100')_{MAX} = (0.05)\ P$ (psi)

where P = the pressure before the control valve, psig

After the Control Valve: $(\Delta P/100')_{MAX} = (0.10)\ P$ (psi)

where P = the pressure after the control valve, psig

If the operating pressure is unknown, use 10 psig for fuel gas with a gross heating value of ≥ 500 btu/scf or 20 psig for fuel gas with a gross heating value < 500 btu/scf.

Fuel Gas Header: $(\Delta P/l00')_{MAX} = (0.05)\ P$ (psi)

where P = the pressure in the header, psig

Caution: If more than one process stream is to be heated in the same heater box, the heat release should be calculated for each set of burners.

6.10 Reboiler Return Line to Column

Generally, the heat exchanger specialists will conduct the hydraulic calculations and determine the piping design requirements for the reboiler system. Nevertheless, the following guidelines can still serve as a reference.

Guidelines:

Kettle Type Reboiler: $(\Delta P/100')_{MAX} = 0.20$ psi

Thermosyphon
(or once-through, non-pumped): $(\Delta P/100')_{MAX} = (0.0062)\ (\rho_l - \rho_v) - 0.016$ (psi)

where ρ_l = density of liquid in return line to column, lb/ft^3
ρ_v = density of vapor in return line to column, lb/ft^3

The flow area of the outlet line should be 100-150% of the total tube side flow area. The size of the process inlet may be estimated by:

$(ID)^2/Q = 9$ to 11

where: Q = heat duty in MMBtu/hr
ID = pipe internal diameter in inches.

Caution: Do not size the return line larger than 30". If the calculated diameter exceeds this, split the flow in half and size dual return lines for this reduced flow rate.

For non-kettle reboilers, the return line should be checked for slug flow. Slug flow should be avoided by allowing for more pressure drop in the line or revising the relative amounts of vapor to liquid (by specifying more vapor).

This line and the reboiler supply line should be sized at the same time. Detailed hydraulics of the overall reboiler system must still be reviewed.

6.11 Compressor Suction and Discharge Lines

Compressor piping should be sized to minimize noise, vibration, and pulsation. For critical installations, an engineering study is needed for each compressor. Below are guidelines for calculating pipe sizes.

A. Reciprocating Compressor Suction Line

This line is defined as the piping connecting the suction drum and the compressor, including the branch lines to the individual cylinders. It does not include the lines feeding into the suction drum.

Guidelines:

Operating Pressure, psig	(ΔP/100')MAX, psi
<0	0.05
0 to 100	0.05-0.19
100 to 500	0.19-0.49
500 to 1000	0.49-0.85
>1000	$0.025(P')^{1/2}$

$$V_{MAX} = 33\,(28.8/MW)^{1/2}\ \text{(fps)}$$

where P' = suction line pressure, psia

Both guidelines should be met.

B. Centrifugal Compressor Suction Line

Guidelines:

$(\Delta P/100')_{MAX}$ = See table from section A

$$v_{MAX} = 59\,\rho_v^{-1/4}\ \text{(fps)}$$

where P' = suction line pressure, psia
ρ_v = gas density at flowing temperature and pressure in the suction line, lb/ft^3

Both guidelines should be met.

C. Reciprocating Compressor Discharge Line

This line is defined as the piping connecting the compressor and the first piece of equipment downstream of the compressor such as an exchanger or knockout drum or until a junction with a line carrying other fluids. This criterion does include the branch lines from the individual cylinders.

Guidelines:

$(\Delta P/l00')_{MAX} = 0.043\ P'^{1/2}$ (psi)

$(\Delta P/100')_{MAX} = 1.0$ psi

$v_{MAX} = 50\ (28.8/MW)^{1/2}$ (fps)

where P'= discharge line pressure, psia

All three guidelines should be met.

D. Centrifugal Compressor Discharge Line

Guidelines:

$(\Delta P/100')_{MAX} = 0.043\ P'^{1/2}$ (psi)

$(\Delta P/l00')_{MAX} = 1.0$ psi

$v_{MAX} = 100\ /\ \rho_v^{1/2}$ (fps) – but not more than 60-80 fps

where P' = discharge line pressure, psia
ρ_v = gas density at flowing temperature and pressure in the suction line, lb/ft^3

All three guidelines should be met.

6.12 Make-up Gas Lines to a Hydrotreater or Hydrocracker

Guidelines:

$(\Delta P/100')_{MAX} = 1.0$ psi

$v_{MAX} = 100\ /\ \rho_v^{1/2}$ (fps)

Caution: For long lines it may be advisable to size for a smaller pressure drop.

6.13 Crude Oil Vacuum Unit Furnace Transfer Line

The exact configuration of this line must be accurately known to determine its size. Observing sonic velocity limitation, and allowing for changing pressure along the length of the line, size the line so as to use up the available pressure drop between the furnace and the flash zone. This is usually less than 3 psi. Generally, this line is sized to have gradual increases along the route until reaching the column. If the pressure at the downstream end of the line is raised because of sonic velocity limitation, the nozzle at the inlet to the column should be enlarged until sonic velocity or less can be attained in the nozzle at flash zone pressure (i.e., the inlet nozzle sizing guideline: $v_{MAX} \leq$ Mach 1). The fired heater specialist should be consulted for a rigorous hydraulic evaluation.

6.14 Feed to a Column

Guidelines:

Pump Discharge Line or Downstream of Pump Discharge Line: Treat as described in Section 6.2 as appropriate with the following constraint:

For 100 % Liquid: v_{MAX} = 8.3 fps

Lines with high-available ΔP ($\geq$ 25 psi for piping friction loss): Treat as described in Section 6.4 with the following constraint:

For 100 % Liquid: v_{MAX} = 8.3 fps

For lines, which do not fall within the above listed categories, the following guidelines apply:

$(\Delta P/100')_{MAX}$ = 2.0 psi

For 100 % Liquid: v_{MAX} = 8.3 fps

For 100% Vapor: v_{MAX} = 100 / $\rho_v^{1/2}$ (fps) – not to exceed 60-80 fps

where ρ_v = density of vapor at flowing temperature and pressure, lb/ft^3

Caution: If flow is two-phase, the line should be checked for slug flow at normal and turndown flow conditions. If at all possible, slug flow should be avoided by allowing for more pressure drop in the line.

6.15 Column Overhead Vapor Lines

A. General

Guidelines:

$(\Delta P/100')_{MAX}$ = 0.070 $P'^{1/2}$ psi
v_{MAX} =100 / $\rho_v^{1/2}$ (fps) – not to exceed 60-80 fps

where P' = pressure at the overhead receiver, psia
where ρ_v = density of vapor at flowing temperature and pressure, lb/ft^3

For columns operating at very low pressure where minimum pressure drop is critical, use the following guideline:

$(\Delta P/100')_{MAX}$ = 0.035 $P'^{1/2}$ psi

where P' = pressure at the overhead receiver, psia

B. Crude Oil Vacuum Column Overhead Line

The exact configuration of this line must be accurately known in order to size. As a general guideline, the pressure drop over the entire length should not be greater than 7% of the flash zone absolute pressure.

6.16 Column Overhead Condenser Rundown Lines

A. Column Overhead Condenser Rundown Lines, two-phase flow

Guidelines:
$(\Delta P/100')_{MAX}$ = 0.14 $P'^{1/2}$ psi

where P' = pressure at the overhead receiver, psia

For columns operating at very low pressure where minimum pressure drop is critical, use the following guideline:

$(\Delta P/100')_{MAX}$ = 0.07 $P'^{1/2}$ psi

Caution: Since this line could be vertical up flow, it should be checked for slug flow. If possible, slug flow should be avoided by decreasing the line size. It may be necessary to alter the piping arrangement and/or relax the pressure drop criteria to minimize the likelihood of slug flow.

If this rundown line is part of a pressure relief route, then the process engineer must review the line sizing with respect to emergency relief requirements. This situation would occur when a relief valve has not been located on the column or on the column overhead vapor line. The column must then vent through this line to reach the vent or pressure relief valve on the overhead accumulator.

B. Column Overhead Condenser Rundown Lines, totally condensing

Guidelines:

v_{MAX} = 3.3 fps

$(\Delta P/l00')_{MAX}$ = 0.14 $P'^{1/2}$ psi

where P' = pressure at the overhead receiver, psia

For columns operating at very low pressure where minimum pressure drop is critical, use the following guideline:

$(\Delta P/l00')_{MAX}$ = 0.07 $P'^{1/2}$ psi

where P' = pressure at the overhead receiver, psia

Caution: If the rundown line contains traces of a second, heavier liquid (e.g. water), and the piping arrangement would permit pockets of this second liquid to accumulate, then:

v_{MIN} = 5.8 fps

This higher velocity will help to keep the line clear of the heavier fluid; however, if this rundown line is also the path for column relief, further consideration must be given to the overall system before finalizing the design.

6.17 Relief Valve Discharge Line to Atmosphere

When discharging flammable vapors into the atmosphere, it is desired to entrain sufficient air so as to dilute the mixture below the flammable limit. With an adequate exit velocity the gas will turbulently mix with air to achieve the objective:

Guideline: v_{MIN} = 500 fps

Caution: Do not exceed erosional velocity. Low-pressure systems may not allow high velocities.

6.18 Amine, Carbonate, and Sour Water

A normal maximum velocity of 7.0 fps should be observed to prevent flashing.

6.19 Sulfur

A. Pumped

v_{MAX} = 10 fps

$(\Delta P/100')_{MAX}$ = 3.0 psi

B. Gravity

v_{MAX} = 2.5 fps

C. Slurry (square duct 1/3 full)

v_{MAX} = 4 fps

6.20 Sulfuric Acid

A. General

v_{MAX} = 4 fps

B. 93% to 99.6% (cast iron 8 in and larger piping at 180°F and below)

v_{MAX} = 6 fps

C. 93% to 99.6% (steel piping at 120°F and below)

v_{MAX} = 1 fps

D. Weak Acid 0%-30% (Plastic or Plastic Lined Pipe)

v_{MAX} = 6 fps

6.21 Sodium Hydroxide

A. 0%-30%

v_{MAX} = 6 fps

B. 30%-50%

v_{MAX} = 5 fps

C. 50%-73%

v_{MAX} = 4 fps

6.22 Flare Lines

The flare relief system must be sized to limit the back pressure (superimposed and built-up) to a value which will neither affect the set pressure, in the case of unbalanced relief valves, nor reduce the discharge capacity of relieving devices (balanced and unbalanced). Refer to API 520 for detailed information on relief system design.

<u>Conventional Pressure Relief Valves</u>

For conventional valves the variable back pressure (built-up) should not exceed 10 % of the set pressure. In fact, the set pressure of this type of valve depends on the value of the superimposed back pressure. Its discharging capacity is reduced rapidly if the built-up back pressure increases beyond certain values (10 to 15 % of set pressure).

Therefore, by limiting the maximum built-up back pressure to 10 % of set pressure, both negative effects (variation of set pressure and reduction of capacity) are kept within acceptable limits.

Balanced Pressure Relief Valves
The set pressure of a balanced valve is independent of back pressure. However, its capacity is reduced when the built-up back pressure exceeds 30 to 50 % of set pressure.

Summary
Careful design of the relieving header system is important to keep the back pressure for all emergencies with acceptable limits to insure proper operation of relieving devices.

Guidelines:

Branches:	v_{AVG}	~ 0.25 Mach
	v_{MAX}	= 0.65 Mach
Main Header:	v_{AVG}	~ 0.25 Mach
	v_{MAX}	= 0.65 Mach
Flare Stack:	v_{AVG}	~ 0.20 Mach
	v_{MAX}	= 0.50 Mach

7.0 HYDRAULIC CIRCUITS

The preceding section provides general guidelines for determining line size. Piping systems are made up of hydraulic circuits, not individual lines. The following will guide the engineer in properly designing piping for a hydraulic circuit.

7.1 Documentation Requirements

The process engineer has the option of doing rigorous calculations for all of the cases (see Section 5.3) or generating hydraulic calculations for the alternate cases by the use of flow ratios using the rigorous model as the primary basis.

A Pressure Profile Tabulation is required for all systems with pressure equipment downstream of the pump. The Pressure Profile Tabulation should include the normal, rated, turndown, and relief cases as a minimum. Optionally, other cases can be included (e.g., fouled exchangers versus clean exchangers, plugged reactor versus unplugged reactor, etc.).

Reference materials or back-up documentation is required for:

- Equipment pressure drops (exchangers, filters, etc.) [data sheets, vendor literature, etc.; see Section 5.7 for estimating values]

- Equipment elevations [equipment drawings]

- Pipe routing and fittings [EFDs, isometric drawings, plot plan and/or calculations with fitting factors; see Section 5.6]

  (Note: If alternate routes are available for the fluid stream, the route chosen for the hydraulic calculations should be highlighted on the EFD or isometric drawing.)

- Pump performance (existing pump) [pump data sheet with head-capacity curve; impeller size must be verified]

- Stream properties [simulation printout or PFD heat and material balance]

- Source operating conditions [simulation or PFD]

- Destination requirements [PFD, simulation, notegram, etc.]

- Two-phase flow calculations [printout of 2-phase calculation results]

- Pipe specifications (for pipe inside diameter)

All documentation must be kept in the file for review by the checking engineer. After the review has been completed and the results of the calculations have been issued, the results should be retained in a filing system.

7.2 System Sketch

A simplified sketch of the pump circuit must be drawn and attached to the calculations. This sketch must include the following:

- Source vessel with elevation and operating pressure
- Destination point with elevation and operating pressure
- The routing of all principle streams
- All major pieces of equipment (with numbers) on the piping route
- All control valves (with tags if available) [showing control strategy is optional]
- All flow elements (with tags if available)

The following items can be optionally shown on the sketch:

- Line segment numbers
- Reducers (normally at pump and control valve inlets/outlets)
- Line sizes

7.3 Operating Cases

The following hydraulic cases should be reviewed for each pumping circuit:

- Normal operating
- Rated or maximum operating
- Turndown (default is 50 %)
- Special operating scenarios which could fall outside of the range covered by the above listed cases

Note that all product destination cases should be reviewed and documented even if they do not determine the pump head (the non-governing lower head cases).

It is not necessary to conduct rigorous hydraulic calculations on all of the above noted cases. Rigorous hydraulics can be done for one case (usually the normal operating case) and the other cases can be generated using the following flow ratios:

- Piping Losses: $\Delta P_2 = (\Delta P_1)(W_2 / W_1)^{2.00}$
- Exchanger Shell Side Loss: $\Delta P_2 = (\Delta P_1)(W_2 / W_1)^{1.85}$
- Exchanger Tube Side Loss: $\Delta P_2 = (\Delta P_1)(W_2 / W_1)^{1.80}$
- Other Equipment Losses: $\Delta P_2 = (\Delta P_1)(W_2 / W_1)^{2.00}$

Use of pressure profile tabulation on a spreadsheet can facilitate the generation of the hydraulics for these additional operating cases by the input of these pressure drop ratios into the spreadsheet. Key control points can be fixed and the available control valve pressure drops will be automatically calculated.

7.4 Piping Segments

In performing the hydraulic calculations for the pump circuit, the circuit must be divided into piping segments. The following guidelines will apply:

New segments are required for:

- a change in flow
- a change in line size
- a change in stream properties
- a change in the number of phases

Segments should be broken at:

- each piece of equipment
- each control valve

7.5 Piping Lengths & Fittings

The best method to determine piping lengths and fittings is via the use of isometric drawings. The process engineer should request "hydraulic" isometric drawings from Piping at the earliest opportunity. "Hydraulic" isometric drawings are sketches that represent the piping designer's best idea of the expected piping layout at that given time. The final design will vary from this preliminary design, but the equivalent length of piping in the two designs will be very close.

The final or stress isometric drawings should be used to conduct a final check of the hydraulic calculations to insure that the assumptions made by the process engineer with respect to piping were reasonable. Unfortunately, stress isometric drawings are often issued well after the pumps have already been ordered. Even in those cases a final review must be made. In the event that a major error is discovered, there is still an opportunity to resolve the problem before installation. Albeit the discovery of a major error at this stage in the design will not be welcomed, it is still better than trying to resolve the problem in the field during start-up. The purpose of the final check is to validate that the system will operate in a acceptable manner. In order to avoid major piping geometry changes, pump impeller modifications, or other physical changes late in the project, a reasonable compromise between original design requirements and cost/schedule impacts may be required. Good judgment can often avoid major changes in fabricated piping, control valves, or equipment late in a project.

In the absence of isometric drawings to determine equivalent piping lengths, four alternate methods are presented. Method 1 is exclusively for the calculation of pressure drops in pump suction lines. Method 2 is used for short piping runs. Methods 3 and 4 are essentially the same method but are based upon different data bases and therefore result in slightly different values. The more conservative value should be used.

Method 1. For pump suction piping the following table should be used for the estimate of piping equivalent length in the absence of isometric drawings.

Pipe Size	Equivalent Length of Pipe, ft
< 6"	200
6” – 12”	500
≥ 14"	200 + 300 pipe diameters

At first glance these equivalent lengths may seem high, however, these values are consistent and required to insure that NPSH calculations are properly accounted for. Reducers at the pump suction, which may be one to three sizes smaller than the line size, add significantly to piping equivalent length. Expansion loops to handle high temperatures or to minimize pump suction nozzle stresses, since pump manufacturers have steadily decreased the allowable values, also add significantly to piping equivalent length.

Method 2. The process engineer uses the P&ID, plot plan and vessel drawings to visualize the piping layout. In other words, the process engineer simply assumes the role of the piping designer to estimate the number of fittings and valves and calculates the linear length of piping from the plot plan. This method is reasonably accurate for short piping runs when operating temperatures are near ambient (and expansion loops are not required). For long runs, high operating temperatures, or cryogenic operating temperatures, the process engineer can underestimate the equivalent length of pipe. The engineer can estimate the number of expansion loops to obtain an initial pipe size estimate. This estimate should be checked against the isometric pipe drawings.

Method 3. Use the following formula to estimate total equivalent length of piping for a given piping run. This method is based on an article by Grant S. Brown in *Chemical Engineering*, March 1987.

$$L_{eq} / L = 1 + (0.347 \times d_n^{1/2} + 0.216) F_c$$

The value of the complexity factor, F_c, depends upon the type of piping:

Very complex manifolds,	F_c	= 4.00
Manifold-type piping,	F_c	= 2.00
Normal piping,	F_c	= 1.00
Long, straight-run piping,	F_c	= 0.50
OSBL utility supply lines,	F_c	= 0.25

First determine the estimated linear length of pipe for a given segment (L); then multiply that estimated length by the right-hand side of the equation to obtain the total equivalent length including fittings. Do the pressure drop calculation based upon that total equivalent length of piping.

7.6 Equipment Allowances

For all instances, it is better to use calculated values, vendor information (e.g., charts for pressure drop across filters), or a combination of both for equipment pressure drop allowances. Unfortunately, such information may not be available at the time the pump hydraulics are performed. In such instances, the following table can be referenced to obtain allowable values for equipment pressure drops. Once reliable information is available, the hydraulics must be reviewed to insure that the hydraulics of the overall system was not compromised by poor estimated values.

Equipment	ΔP, psi
Coalescers	10
Desiccant Driers	15

Desalters	25-40 (includes mixing valve)
Exchangers:	
Shell & Tube: shell side	10 (or 5 psi per shell for multi-shell)
Shell & Tube: tube side	10 (or 5 psi per shell for multi-shell)
Air Coolers	10
Double Pipe	10 (each side)
Fixed Bed Reactors	20-50
Filters	10 dirty (or 2 psi clean)
Temporary Pump Suction Screens	300 equivalent feet of pipe
Permanent Pump Suction Screens	1 or Max. Allowable pressure drop
Spray Nozzles	20-40 (must check vendor's catalog)

The allowance for temporary pump suction screens should only be used if the client intends to leave the screens in place after start-up.

For systems involving complicated heat exchanger networks, such as the crude preheat train for a Crude Distillation Unit, it is recommended that the process engineer refer to prior projects of a similar nature to obtain better pressure drop estimates. Great care should be taken when applying these past pressure drop estimates.

Equipment pressure drops at flow rates other than equipment design values can be estimated based upon flow ratios when no other information is available. See Section 5.3 for calculation methods.

7.7 Regulating Control Valve Allowances

The following guidelines will apply to regulating control valves in pump discharge lines (when sizing new pumps not reviewing existing ones):

In general the allowable pressure drop will be the largest of:

- 15 psi
- 50 % of the dynamic pressure drop or system friction drop excluding the valve (or 33 % including the valve), or
- 10 % of the pump differential.

This guideline is based upon the rated capacity hydraulic case and not the normal case. All pumps should be reviewed to insure that there is no significant cost penalty in using these control valve allowances. In other words, if a pump size or motor size is altered by the use of these allowances, the system should be further reviewed to determine whether the allowances can be lowered to reduce equipment costs.

If the hydraulic system consists of a pump pumping back to itself, in other words, the source and destination points are the same (including operating pressures and static heads), the control valve allowances can be reduced below the above noted guidelines. Even in these situations, the control valve pressure drop allowance should not be set below 10-15 psi for the rated hydraulic case.

7.8 Diversion Control Valve Allowances

Diversion control valves are valves which direct flow through one line or another as opposed to actually regulating the amount of flow through the line. Temperature control valves which direct flow through a heat exchanger or through the bypass line are a common use of diversion valves.

For valves diverting flow to equipment, the allowable pressure drop should be set at:

- 3.0 psi for the normal operating case
- $(\Delta P_{NORMAL})\ (Q_{MAX} / Q_{NORMAL})^{2.0}$ for the maximum operating case

For valves diverting flow around equipment, the allowable pressure drop will also include the pressure drop through the equipment. The hydraulics through the equipment and associated piping must be calculated in order to determine the pressure drop available for the bypass valve.

7.9 Flow Element Allowances

Generally, flow elements will be designed with a 100" W.C. pressure differential. Control Systems will occasionally modify this differential (in 25" W.C. increments) to meet certain client criteria. Only in unusual circumstances will an instrument's pressure differential exceed 200" W.C. or be less than 25" W.C. (both require client approval).

The following <u>permanent</u> pressure drop losses should be allowed (at 100" W.C.) for the <u>normal</u> flow rate:

- Orifice plates: 2.0 psi
- Wedge meters: 1.0 psi
- Turbine meters: 2.0 psi
- Venturi meters: 0.5 psi
- Annubars and pitot tubes: 0.0 psi

For instruments calibrated with a 200" W. C. pressure differential, these values should be doubled (once again for the <u>normal</u> flow rate).

When developing the hydraulics for flow conditions other than the normal rate, the estimated permanent pressure drop should be adjusted per the following:

- $\Delta P_2 = (\Delta P_{NORMAL}) (W_2 / W_{NORMAL})^{2.0}$

Caution: These pressure drop allowances are for the flow element alone and do not include friction drop losses in the piping. Several of these elements have velocity requirements which result in a significant reduction in line size with a corresponding large increase in pressure drop. (For instance, turbine meter bore diameters will typically be one-two line sizes smaller than the corresponding piping. Straight piping with line size equivalent to the turbine meter bore is required a minimum of ten pipe diameters both upstream and downstream of the meter.) The process engineer must work with Control Systems to insure that such requirements are accounted for in the hydraulic calculations.

8.0 NON-PUMPED HYDRAULIC CIRCUITS

Non-pumped hydraulic circuits include any hydraulic systems which do not include a pump, compressor, or ejector. The fluid flows by pressure or gravity from the source to the destination point. The requirements for non-pumped hydraulic calculations are similar with the obvious exception of not requiring equipment specification sheets. The engineer cannot assume that a less rigorous approach is acceptable since non-pumped circuits often require more precise engineering than more forgiving circuits with pumps that have design margins to permit engineering flexibility.

9.0 TWO-PHASE FLOW

The purpose of this guideline is to provide direction to the process engineer who is presented with a hydraulic circuit that involves two-phase flow. Given the myriad of research articles that have been written on this subject, there is no intent by this guideline to edify the process engineer on the various correlations that have been developed. Process engineers who are interested in this subject are encouraged to read articles themselves. A good review of the topic can be found in the Gas Processors Suppliers Association Engineering Data Book.

All discussions concerning two-phase flow in this hydraulics guideline is pertinent for typical inside-the-battery-limit process piping applications and should not be applied to pipeline situations.

9.1 Flow Regime / Slug (Intermittent) Flow

Flow regime prediction is a complicated problem and numerous correlations with corresponding flow regime maps have been developed. The Taitel/Dukler (horizontal) and Taitel/Dukler/Barnea (vertical and horizontal) are generally accepted as the best. Their correlations allow for analytical prediction of the transition between the different regimes and are applicable to a wide range of pipe sizes and fluid properties. Other flow regime maps, such as the Baker chart, should not be used.

Horizontal flow regimes include stratified, wavy, annular dispersed, dispersed bubble, and intermittent or slug flow. Vertical flow regimes include bubble, annular, dispersed bubble, and slug flow. Of primary interest to the process engineer is slug flow. Slug flow develops when liquid bridges the flow area of the pipe thereby blocking the passage of the vapor portion. The liquid slug is then picked up and accelerated to the velocity of the vapor following it. The resulting stream flow is made up of alternating liquid-gas flow with large pressure pulsations due to the high-velocity slugs. Initiation and propagation of slug flow is more apt to occur in vertical upflow lines, where the problems associated with slug flow tend to be more severe. In vertical upflow lines, slug flow develops when the line rises vertically so as to create a pocket where liquid can settle and cause a liquid plug, which is later picked up and pushed through the line by the vapor phase. Slug flow will not occur in a gravity-flow line that has no pockets.

An objective of process design should be the mitigation of slug flow in lines with two-phase flow. Slug flow causes pressure fluctuations in piping, which can cause vibrations of piping and equipment, upset process conditions, and inconsistent instrument sensing. The potential for slug flow is increased if a line in two-phase service is sized for low pressure drop. Therefore, the most common method of eliminating slug flow is to reduce line sizes to the minimum permitted by available pressure differential. By minimizing line size, fluid velocities are increased which results in less liquid slippage and holdup thereby reducing the likelihood that slug flow will be initiated. Other ways for eliminating slug flow include:

- Arranging the piping configuration to protect against slug flow (i.e., elimination of pockets)

- Relocating a control valve so as to eliminate or minimize downstream two-phase piping (by placing the valve next to the downstream vessel)

- Increasing the vapor to liquid flow ratios (i.e., by adding more heat for increased vaporization or by injecting with a gas stream)

- Size lines to exceed 10 feet per second velocity.

- Installing parallel piping runs to accommodate normal and turndown flow rates (i.e., large-diameter piping for the normal flow; small-diameter piping for the turndown case)

- Installing liquid knockout pots to catch and separate the liquid and vapor phases by collecting and draining the liquid portion of the stream

In the event that a satisfactory solution to eliminating slug flow cannot be found, the piping should be braced for possible slug flow. The P&IDs should be marked accordingly.

9.2 Mixture Density

In vertical two-phase flow lines, the static head can be a significant portion of the total pressure drop. The process engineer should not use the homogeneous mixture density. The assumption of homogenous mixing, i.e. a slip ratio of 1.0, almost always results in an underestimated value for the two-phase mixture density (with a resulting low value for the static head). This error increases with an increase in the density ratio for the two phases. Since liquid slippage decreases with an increase in overall mass velocity, the error in calculating the combined mixture density decreases. Nevertheless, regardless of overall mass velocity, the assumption of no-slip flow in the calculation of mixture density should not be made. The API 14E guidelines for mixture density are shown below.

$$\rho_m = \frac{12409 s_l P + 2.7 R s_g P}{198.7 P + RTZ}$$

P = pressure in psia
S_l = liquid specific gravity at standard conditions (Use average gravity for water hydrocarbon mixtures)
T = operating temperature in ºR
S_g = gas specific gravity at standard conditions
R = gas/liquid ratio in ft^3/barrel at standard conditions
Z = gas compressibility factor

9.3 Line Sizing Criteria

The guideline to avoid slug flow and erosional velocity are the primary criteria in the sizing of lines in two-phase flow service. Line noise could be a secondary criterion. Refer to previous sections for line-sizing criteria based upon the particular service application.

9.4 Calculation

Due to the complexity of two-phase flow calculations, using a computer program for these calculations is recommended. Use the Beggs-Brill-Moody correlation for horizontal pipe runs. For vertical runs use Dukler-Eaton-Flanigan. For column risers use twice the vertical distance in the calculation.