CONTROL VALVES

STD-09-051 Rev. 0, Sep-03

September 15,2003

CONTENTS

1. SCOPE

(1) This standard practice provides the guidelines for the preparation of basic design information concerning control valves in refineries, petrochemical plants and similar plants.
Selection and sizing method of control valves given in Appendixes is only for reference.

(2) Specific instructions given in the Project Specifications take precedence over requirements given in this standard practice.

2. WORK PROCEDURE

2.1 Input to the Design

The base data required to design the control valve and major source documents of them are as follows :

2.1.1 Source Documents

(1) Basic Engineering Design Data (BEDD)
(2) Process Flow Diagram (PFD)
(3) Piping & Instrument Diagram (P&ID)
(4) Process Simulation Output
(5) Hydraulic Calculation Sheets

2.1.2 Base Data

(1) General
- Tag number
- Service name
- Regulation

(2) Operating conditions
- Fluid name
- Flow rate – normal, maximum and minimum
- Inlet pressure at normal or maximum flow rate
- Pressure drop across valve at normal or maximum flow rate
- Operating temperature
- Physical properties at control valve inlet for single phase, and at inlet and outlet for flashing service and mixed phase
  Liquid : Specific gravity, Viscosity, Vapor pressure, Critical pressure and solid%
  Vapor : Molecular weight, Viscosity, Specific heat ratio (k) and Compressibility factor (Z) and solid%

Flash %(wt base) at inlet and outlet should be specified for flashing service and mixed phase.

(3) Construction data
- Design pressure
- Design temperature
- Fail-safe position
- Seat tightness, if specifically required
- Maximum shut-off pressure
- Line size, inlet and outlet
- Line class, inlet and outlet
- Allowable maximum selected CV-value, if necessary

2.2 Output from the Design

Instrument engineer conduct the control valve design based on the data shown in section 2.1.2 and the following are determined :

(1) Calculated CV
(2) Selected CV
(3) Control Valve Type and Body Size
(4) Predicted Noise Level

According to the selected control valve, the following information should be indicated on P&IDs :

(1) Control Valve Type and Body Size
(2) Block Valve Size
(3) By-pass Line and Valve Size
(4) Noise Protection, if required

2.3 Work Steps

The design of control valve will be conducted according to the following work steps, as a guideline :

(1) PFD : The Philosophy of all process control is indicated on PFD.
(2) P&ID : All required facilities including the accessories of control valve for all process controls shown on PFD are indicated on P&ID.
(3) Hydraulic Calculation
(4) Preparation of Control Valve Data Sheets

3. DESIGN

This section indicates guidelines to design the control valve and to obtain required information for the control valve design.

3.1 Flow Rate

(1) Flow rate and its relevant pressure drop across the control valve should be provided on data sheets. The maximum and minimum flow conditions should be specified.

(2) In case of 3-way control valve, the flow rate and the pressure drop for each flow path should be provided. Especially, where the 3-way valve is applied for the duty control of the heat exchanger, the maximum flow to the exchanger path and zero flow to the by-pass should be specified in addition to the normal flow conditions.
The 3-way control valve should not be applied to the service where tight shut is required due to large leakage.

(3) Abnormal operating condition including start-up, shutdown, regeneration, etc. should be also considered in preparation of the control valve data sheets in addition to the normal operation.

3.2 Pressure Drop

The pressure drop across the control valve based on the flow rate during normal operation can be obtained as follows :

3.2.1 Centrifugal Pump Discharge

(1)	$\Sigma\Delta P_{fric}$ 5.0 kg/cm^2	$\Delta P_{CV} = 0.5 \text{ x } \Sigma\Delta P_{fric}$
(2)	5.0 kg/cm^2 < $\Sigma\Delta P_{fric}$ 6.25 kg/cm^2	$\Delta P_{CV} = 2.5$ Kg/cm^2
(3)	6.25 kg/cm^2 < $\Sigma\Delta P_{fric}$ 10.0 kg/cm^2	$\Delta P_{CV} = 0.4 \text{ x } \Sigma\Delta P_{fric}$
(4)	10.0 kg/cm^2 < $\Sigma\Delta P_{fric}$ 13.4 kg/cm^2	$\Delta P_{CV} = 4.0$ Kg/cm^2
(5)	13.4 kg/cm^2 < $\Sigma\Delta P_{fric}$	$\Delta P_{CV} = 0.3 \text{ x } \Sigma\Delta P_{fric}$

Where :

ΔP_{CV}	=	pressure drop across the control valve	(kg/cm^2)
ΔP_{fric}	=	friction losses of lines, equipment, instruments, piping parts, etc.	(kg/cm^2)

The above criteria should not be applied to the control valve located on such special high head pump discharge as pipe line pump.

When the above criteria results in uneconomical or unpractical pump head, ΔP_{CV} should be determined in consultation with Lead Process Engineer.

It is recommended that the size of control valve should be estimated to confirm whether the calculated pressure drop is proper or not, because the proper pressure drop will lead to one or two size smaller control valve than line size. The procedure for calculating the size of control valve is described in Appendix-3.

For the calculation of SDPfric and line sizing, refer to STD-09-043 Rev. 0, Sep-03 “Hydraulic Design”.

3.2.2 Minimum Pressure Drop

The minimum pressure drop should be kept as follows :

(1) Liquid service (mainly pump discharge) ΔP_{CV} = 0.7Kg/cm^2
(2) Vapor service ΔP_{CV} = 0.2Kg/cm^2

In case of available pressure drop of the control valve is less than above values, discuss the control valve selection with vendor under the assist of instrument engineer.

Note : Minimum pressure drop criteria should be reviewed by project to project basis.

3.2.3 Variation of Static Pressure

Where the variation of operating pressure either in the fluid source or destination is expected, the provision for such variation should be considered in determining the pressure drop across the control valve.

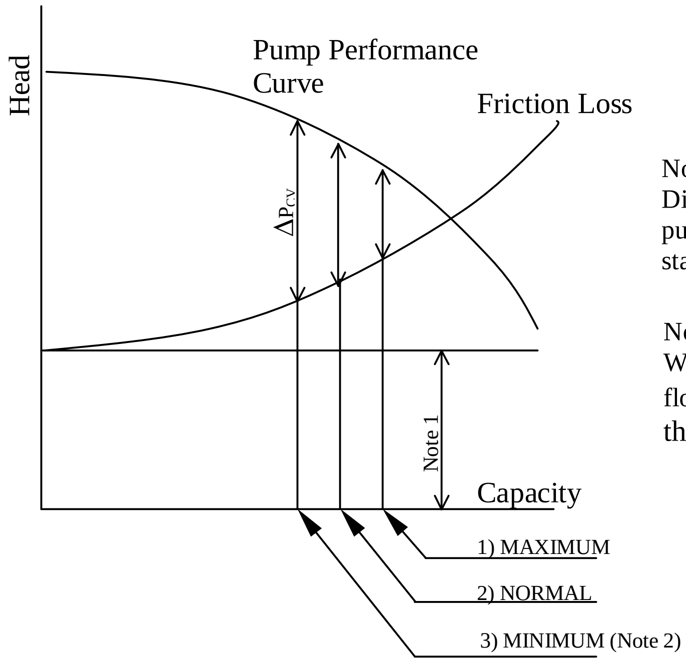
3.2.4 Control Valve in Reactor Circuit

In case where the pressure drop at EOR is provided for the design, it should be also indicated on control valve data sheets that the opening of the control valve should be 85~90% to avoid over design. Because the control valve in Reactor Circuit is usually designed so that its opening is as follow according to the running status :

	Opening
(1) SOR (Start of Run)	60~70%
(2) MOR(Middle of Run)	60~70%
(3) EOR (End of Run)	85~90%

3.2.5 Pump Performance Curve

It is required to confirm that ΔP_{CV} for following three cases : 1) flow rate at maximum, 2) flow rate at normal , 3) flow rate at minimum, is suitable to flexible operation., when pump performance curve becomes available



Note 1
Difference of operating pressure between pump suction and discharge side (including static head)

Note 2
When actual flow rate is less than pump minimum flow, ΔP_{CV} should be determined based on the pump head at pump minimum flow.

3.2.6 Example Calculation

Various illustrations for determining the pressure drop across the control valve are attached to Appendix-6.

3.3 Seat Leakage

ANSI/FCI 70-2 provides the seat leakage specification and classes of the control valves with a rated CV greater than 0.1. The seat leakage is classified into the following 6 classes according to the maximum allowable seat leakage :

Seat Leakage Class

Leakage Class	Maximum Seat Leakage	Description
Class I	No test is required.	A modification of any Class II, III or IV valve where design intent is the same as the basic class, but by agreement between user and supplier, no test is required.
Class II	0.5% of rated valve capacity	This class establishes the maximum permissible leakage generally associated with commercial double-port, double-seat control valves or balanced single-port control valves with a piston ring seal and metal-to-metal seats.
Class III	0.1% of rated valve capacity	This class establishes the maximum permissible leakage generally associated with Class II, but with a higher degree of seat and seal tightness.
Class IV	0.01% of rated valve capacity	This class establishes the maximum permissible leakage generally associated with commercial unbalanced single-port, single-seat control valves and balanced single-port control valves with extra tight piston rings or other sealing means and metal-to-metal seats.
Class V	5×10^{-12} m^3 / sec-water / mm-orifice diameter / bar-differential	This class is usually specified for critical applications where the control valve may be required to be closed, without a blocking valve, for long periods of time with high differential pressure across the seating surfaces. This class is generally associated with metal seat, unbalanced single-port, single seat control valves or balanced single port designs with exceptional seat and seal tightness.
Class VI	Bubble-tight, as per specified in FCI 70-2	This class establishes the maximum permissible seat leakage generally associated with resilient seating control valves either unbalanced or balanced single port with "O" rings or similar gap-less seals.

On P&IDs, TSO (Tight Shutoff) should be indicated to the control valves whose seat leakage should be minimized from a safety viewpoint. TSO is required for the following control valves preferably :

(1) Control valves incorporated in emergency shutoff system (for this application, client’s approval is required)

(2) Control valves on fuel gas / fuel oil supply lines to furnace even if the dedicated emergency shutoff valves are provided.

Seat leakage class selection is joint work of process and instrument engineer. Normally Class II or III is assigned to the control valves and Class V or VI is assigned to those with TSO.

3.4 Shutoff Pressure

Actuator is designed based on the shutoff pressure (ΔP_{shut}) which is obtained by the following equations :

$\Delta P_{shut} = P_{up}$ (+ 1.0 : when the downstream pressure is vacuum) (kg/cm^2) ? ? ? ? ? (1)

$\Delta P_{shut} = P_{up} - P_{down}$ (kg/cm^2) ? ? ? ? ? (2)

Where :

P_{up} = Design pressure of upstream line:* (kg/cm^2G)

P_{down} = Min. normal operating pressure of downstream line (kg/cm^2G)

The above equation (1) is usually applied to obtain shutoff pressure for general services. The above equation (2) is applied to control valves whose downstream pressure is maintained at a constant pressure at all times.

To ensure correct application of either equation (1) or equation (2), various examples for typical flow in the refinery are illustrated in Appendix-7.

*For the determination of design pressure of upstream line, refer to STD-09-019 Rev. 0, Sep-03 "Determination of Design Conditions" .

3.5 Shutoff Speed

The shutoff speed of the control valves should be evaluated for the purpose of ensuring the safety after emergency trips. The evaluated shutoff speed should be specified on the control valve data sheets, as required.

(1) Standard shutoff speed is 10 seconds for valve sizes of 4 inch and smaller, or 15 seconds for valve sizes of 6 inch and larger. If the faster shutoff speed is required, it should be specified on the data sheet. If very fast shutoff action (less than 2.0 seconds), the shutoff speed should be evaluated and specified using the dynamic simulation.
(2) For anti-surge control valve of centrifugal compressor, the requirements on opening speed should be designed by dynamic surge studies in cooperation with compressor vendor and anti-surge controller vendor.
(3) For shutoff service in long liquid pipelines, the shutoff speed requirements should be evaluated to prevent pressure surging . DYNALIQ is applied to surge analysis, unless otherwise client does not have requirement on surge analysis method.

3.6 Noise and Vibration

Check of noise level and vibration caused by high noise is control valve vendor's work. The maximum allowable noise level is specified in project specification. Unless otherwise specified, 85 dB(A) can be applied. Guidelines for noise level estimation are provided in Appendix-4.

3.7 Accessories

The accessories assist and monitor the working the control valve and the following accessories are available. The accessories which need to be installed should be indicated on P&IDs :
(1) Solenoid valve and manual reset
To add the same on-off action as that of emergency shutoff valve to the control valve
(2) Hand wheel
To adjust valve opening manually, hand wheel is usually provided for the control valve without by-pass valve (Hand wheel should not be provided when interlock is provided.)
(3) Minimum / Maximum stopper
To limit the range of closing / opening
(4) Limit switch
To indicate the on-off status of valve

3.8 Others

3.8.1 Weld Type Control Valve

Weld type control valve should be used for special services such as high temperature and high pressure service or highly toxic service as per the project specification.

3.8.2 Fire Resistance

The emergency shut-off valve at vessel outlet should be considered the requirements of fire resistance as per the project specification.

3.8.3 Failure Action
One of the following actions should be specified in the case of instrument air failure :
- Failure close (FC)
- Failure open (FO)
- Failure locked close (FLC)
- Failure locked open (FLO)

For the selection of failure action, refer to section 6.2 "Failure Action of Control Valve" of STD-09-056 Rev. 0, Sep-03.

3.8.4 Block and By-pass Valve
Refer to sect. 5.6 " Control Valve Assembly" of STD-09-056 Rev. 0, Sep-03.

3.8.5 Gas Break Through
Refer to sect. 6.3 "Inadvertent Control Valve Opening" of STD-09-057 Rev. 0, Sep-03.

4. RELATED DOCUMENTS

The following publications will constitute a part of this standard practice. Unless otherwise specified, refer to the latest edition. Applicable publications for the Project will be specified in each of the Project Specifications.

PDIL Standard Practice

STD-09-019 Rev. 0, Sep-03	Determination of Design Conditions
STD-09-043 Rev. 0, Sep-03	Hydraulic Design
STD-09-057 Rev. 0, Sep-03	Pressure Relieving Design
STD-09-056 Rev. 0, Sep-03	Design Criteria for P&ID Preparation

ANSI / FCI (Fluid Control Institute)

Std 70-2	Quality Control Standard for Control Valve Seat Leakage

IEC (International Electrotechnical Commission)

534	Industrial-Process Control Valve

API (American Petroleum Institute)

RP 551	Process Measurement Instrumentation

ISO (International Organization for Standardization)

5208	Industrial Valves - Pressure Testing for Valves

BS (British Standards Institution)

1655	Specification for Flanged Automatic Control Valves for the Process Control Industry (face-to-face dimension)
6739	Code of Practice for Instrumentation in Process Control Systems : Installation Design and Practice

ISA (Instrument Society of America)

S75.01	Handbook of Control Valve - Flow Equations for Sizing Control Valves

5. DEFINITIONS

Flow Rate - Normal : the normal flow rate or maximum flow rate in the normal operation (normal maximum flow rate) defined in the process specifications, such as in the heat and material balance for the process.

Flow Rate - Design : the normal (or normal-maximum) flow rate plus extra margin added to the normal flow rate, in order to enable the rate to be controlled around the normal flow rate. If the normal flow rate is not specified such as in the case of normally no flow lines, this flow rate shall be determined based on the process specifications or operational studies.

Flow Rate - System Limit : the highest flow rate, where the control valves will be fully opened or all head losses will be equal to the system driving force. This flow rate shall be equal to or higher than the design flow rate.

Flow Rate - Minimum : the minimum controllable flow rate or specified lower operating range of the process unit.

Turndown : the ratio of the minimum flow rate to the normal flow rate.

Rangeability : the ratio of the maximum flow rate at which the control valve will provide safe, stable control to the minimum flow rate at which the control valve will provide safe, stable control.

Choked flow : is that condition at constant inlet pressure when no increase in flow rate is achieved for a decrease in downstream pressure.

Vena contracta : is that point downstream of the flow restriction where the flow stream reaches its minimum cross sectional area and thus its maximum velocity and minimum pressure.

Cavitation : is a two-stage phenomenon, the first stage of which is the formation of vapor bubbles within the liquid system. The second stage is the collapse or implosion of those bubbles back into the all-liquid state. Valves with incremental pressure reduction may be one of the solutions to minimize or prevent cavitation.

Flashing : is that condition where the cavitation vapor persists downstream of the region where bubble collapse normally occurs, ie, the cavitation process stops before the completion of the second stage defined in the above "Cavitation".

6. ABBREVIATIONS

Cv	=	valve flow coefficient	(-)
C	=	critical flow factor	(-)
C_f	=	valve - reducer critical flow factor	(-)
d	=	valve size	(mm)
D	=	line size	(mm)
G	=	gas specific gravity (air = 1.0)	(-)
G_f	=	specific gravity at flowing temperature	(-)
L	=	trim travel or lift	(unit of length)
P_1	=	upstream pressure	(kg/cm^2A)
P_2	=	downstream pressure	(kg/cm^2A)
P_c	=	critical pressure	(kg/cm^2A)
P_{down}	=	minimum probable downstream pressure	(kg/cm^2G)
P_{up}	=	maximum upstream pressure with the valve fully closed	(kg/cm^2G)
P_v	=	vapor pressure of liquid at flowing temperature	(kg/cm^2A)
P_{VC}	=	pressure at vena contracta	(kg/cm^2A)
q	=	liquid flow rate	(m^3/h)
r_0	=	valve pressure drop ratio	(-)
R	=	sub - critical flow capacity correction factor	(-)
T	=	flowing temperature	(°K)
T_{th}	=	steam superheat	(°C)
W	=	flow rate	(1000 kg/h)
Z	=	compressibility factor	(-)
ΔP	=	actual pressure drop $P_1 - P_2$	(kg/cm^2)
ΔP_{CV}	=	control valve pressure drop	(kg/cm^2)
ΔP_{elev}	=	static heads	(kg/cm^2)
ΔP_{fric}	=	friction losses	(kg/cm^2)
ΔP_{head}	=	total static heads	(kg/cm^2)
ΔP_s	=	critical pressure drop for cavitation	(kg/cm^2)
ΔP_{shut}	=	maximum shut-off pressure for actuator sizing	(kg/cm^2)
ΔP_{total}	=	total pressure drop between source and destination	(kg/cm^2)
α	=	design margin	(-)
ϕ	=	Flow rate ratio to maximum flow rate	(-)
γ	=	rangeability = Cvmax / Cvmin	(-)
μ	=	viscosity	(cP)
σ	=	lift ratio = L / Lmax	(-)
ω	=	Cv ratio = Cv / Cvmax	(-)

7. REFERENCES

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LITERATURE

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APPENDIX-1 FLOW CHARACTERISTICS

A1.1 Inherent Flow Characteristics

(1) ***Quick opening.*** This trim design provides a large opening as the plug is first lifted from the seat, with lesser flow increases as the plug opens further. This type is most commonly used where the valve will be either open or closed with no throttling of flow required. The flow characteristics can be calculated as follows:

$$\omega = \left(1 - \frac{1}{\gamma}\right)\sqrt{\sigma} + \frac{1}{\gamma} \quad \text{or} \quad \sigma = \left[\left(\omega - \frac{1}{\gamma}\right) \Big/ \left(1 - \frac{1}{\gamma}\right)\right]^2 \qquad \text{(A1-1)}$$

Where : ω = Cv ratio = Cv / Cv_{max} (-)
σ = lift ratio = L / L_{max} (-)
γ = rangeability = Cv_{max} / Cv_{min} (-)
Cv= flow coefficient (-)
L = trim travel or lift (unit of length)

(2) ***Linear.*** Linear trim provides equal increases in Cv for equal increases in stem travel. Thus the Cv increase is linear with plug position throughout its travel. The flow characteristics can be calculated as follows:

$$\omega = \left(1 - \frac{1}{\gamma}\right)\sigma + \frac{1}{\gamma} \quad \text{or} \quad \sigma = \left(\omega - \frac{1}{\gamma}\right) \Big/ \left(1 - \frac{1}{\gamma}\right) \qquad \text{(A1-2)}$$

(3) ***Butterfly.*** Butterfly trim provides second power increases in Cv for equal increases in stem travel. The flow characteristics can be calculated as follows:

$$\omega = \left(1 - \frac{1}{\gamma}\right)\sigma^2 + \frac{1}{\gamma} \quad \text{or} \quad \sigma = \left[\left(\omega - \frac{1}{\gamma}\right) \Big/ \left(1 - \frac{1}{\gamma}\right)\right]^{1/2} \qquad \text{(A1-3)}$$

(4) ***Equal percentage.*** Equal percentage trim provides equal percentage increases in Cv for equal increases of stem travel. This is accomplished by providing a very small opening for plug travel near the seat and very large increases toward the more open position. As a result, a wide rangeability of Cv is achieved. The flow characteristics can be calculated as follows:

$$\omega = \gamma^{\sigma - 1} \quad \text{or} \quad \sigma = \frac{\log\omega}{\log\gamma} + 1 \qquad \text{(A1-4)}$$

This equation will give a straight line on a semi-logarithmic graph.

The flow characteristics described above are illustrated in Fig. A1-1 and A1-2, assuming the rangeability of the valve, γ = 40.

Fig. A1-1 Inherent flow characteristics

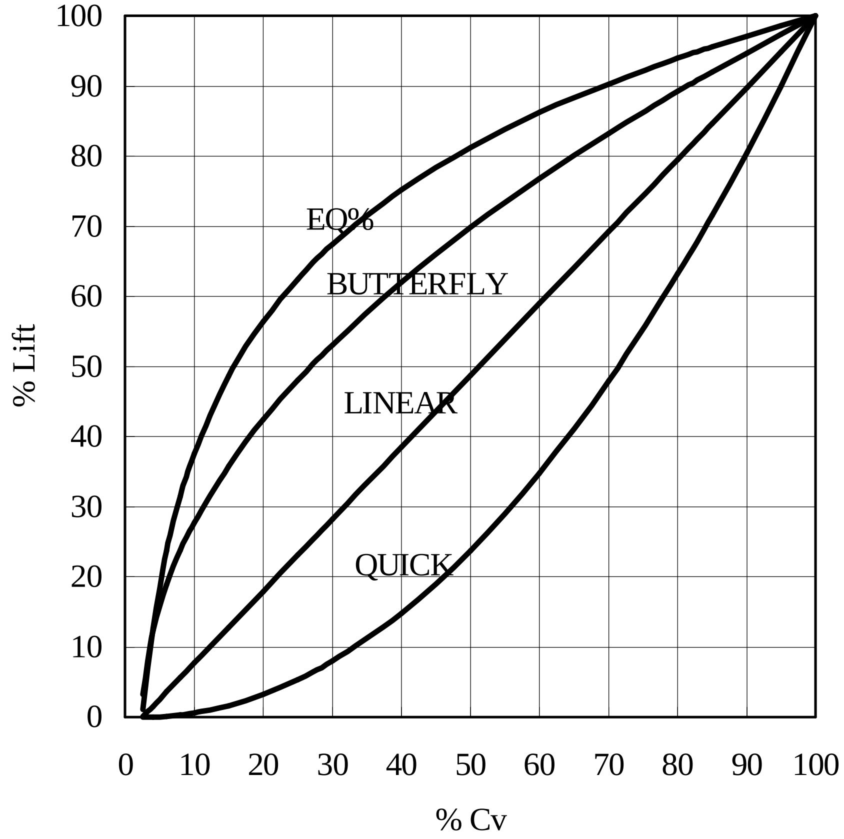
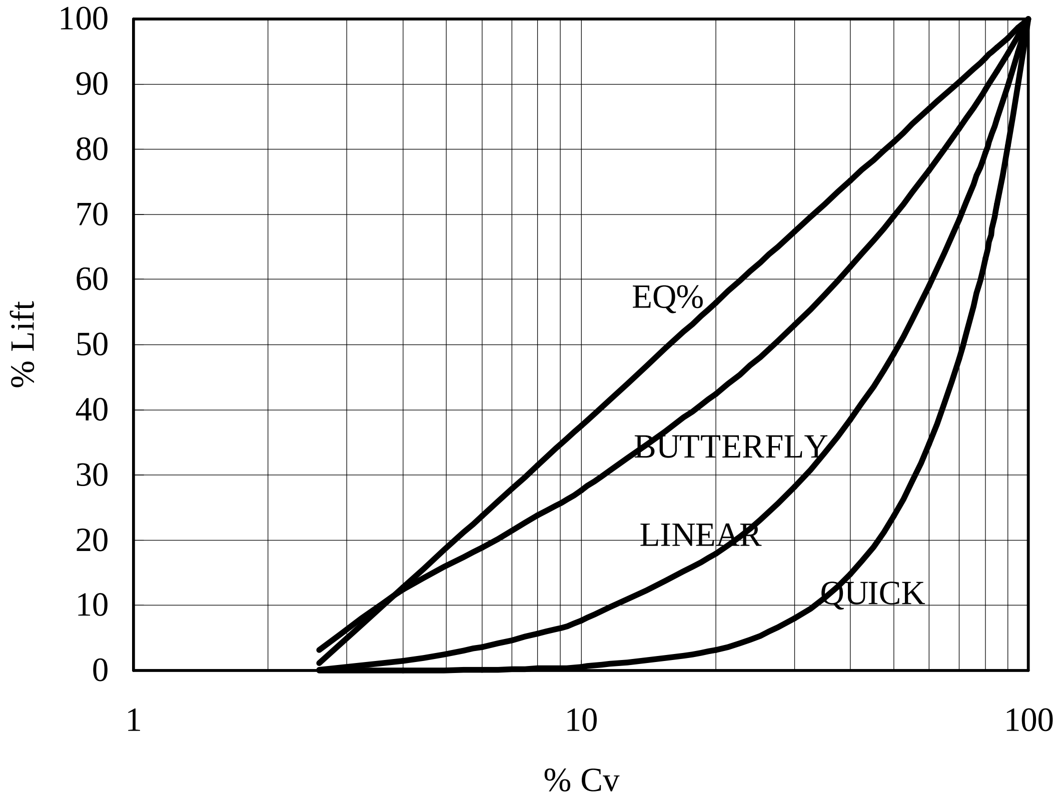


Fig. A1-2 Inherent flow characteristics (semi-log)



A1.2 Installed Flow Characteristics

(1) The pressure difference across the valve often varies with flow due to pump performance and friction losses in the system. This results in an "installed characteristic", which will differ from the inherent characteristic.

(2) Assuming total pressure drop, i.e. the sum of friction losses and control valve pressure drop, is maintained at constant for any flow rate, the Cv ratio, ω , can be calculated as follows :

$$\omega = \frac{1}{\sqrt{1-\frac{1}{r_0}\left(1-\frac{1}{\phi^2}\right)}} \qquad \text{(A1-5)}$$

Where :

r_0 = valve pressure drop ratio = (ΔPcv at maximum flow rate)/(Total pressure drop)

ϕ = (Flow rate)/(Maximum flow rate)

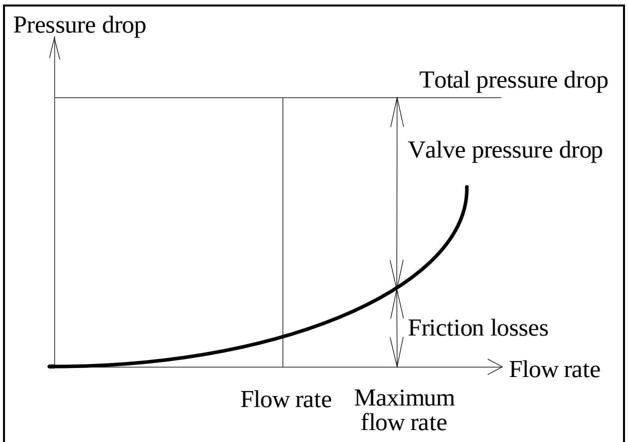


Fig. A1-3 Pressure drop in the system

(3) The installed rangeability can be calculated with previous equations (A1-1) to (A1-4) and (A1-5), making the valve pressure drop ratio, r_0, a parameter. The calculation result for linear and equal-percentage characteristics are illustrated in Fig. A1-4 and A1-5, respectively.
Note that the particular installation can have a substantial effect on both flow characteristics and rangeability. Equal-percentage characteristics will be distorted toward linear or even quick-opening characteristics. Installed rangeability will be decreased down to 10:1 to 15:1 from the inherent rangeability of 40:1.

Fig. A1-4 Installed characteristic (Linear)

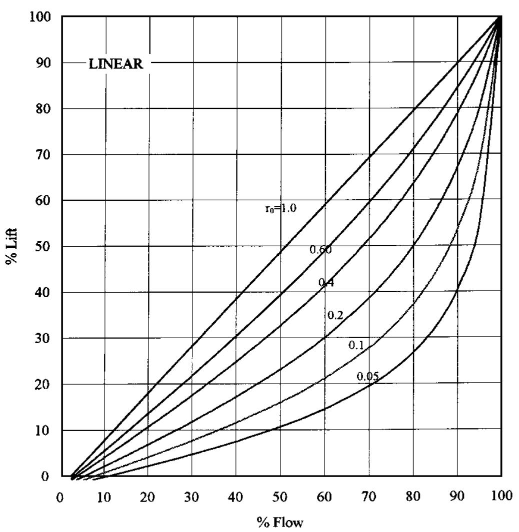
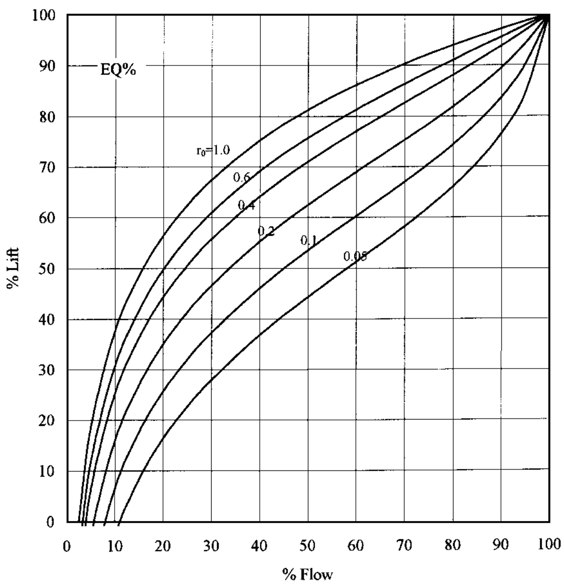


Fig. A1-5 Installed characteristic (Equal-percentage)



A1.3 Selection Guide for Flow Characteristics

(1) The flow characteristic of the control valve will be "Linear", except for the following items (2) and (3).
(2) "Equal percentage (denoted as EQ%)" will be applied for the following services:

- For temperature control
- For the services where the pressure drop across control valve or flowrate varies significantly
- For service with relatively small γ_o (Fig. A1-4)

(3) "Quick opening (denoted as On-off)" will be applied for the following services:

- Self-actuated pressure regulator
- On-off control service
- For service with relatively large friction, such as product rundown line.

APPENDIX-2 TYPE OF CONTROL VALVES

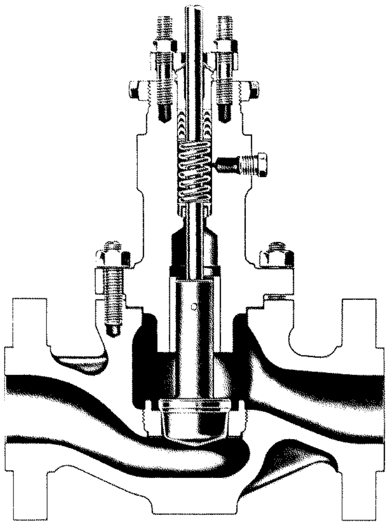
(1) Although valve type selection is described herein, due to the many proprietary valve designs to vendors' own criteria, selection of valve type for a particular service will be conducted by the instrument engineer.
(2) General features and typical application for control valves are summarized as follows:

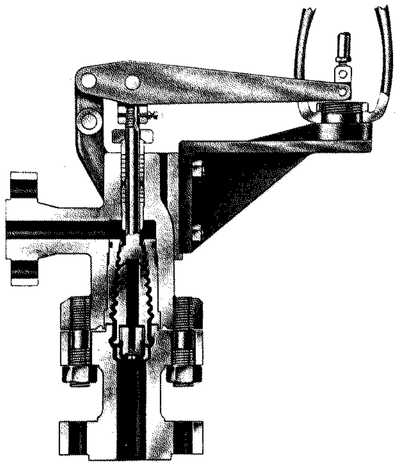
Table A2-1 Type of control valves

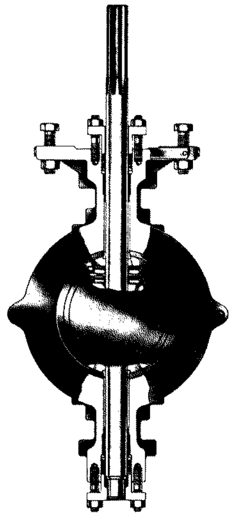
Type	General	Application
Globe body valve	Flow characteristics : Optional Rangeability (effective) : 10 Leak (% of rated capa.) : 0.01 to 0.5%	Most common, provided in the middle of straight line. 1½" and smaller : Single seat valve 2" and larger : Double seat valve Single seat valve can be used for tight shut off service.
Angle body valve	Flow characteristics : Optional Rangeability (effective) : 10 Leak (% of rated capa.) : 0.01 to 0.5%	Applicable to flashing, erosive, high ΔP, or slurry services. Available single seat valve only Solid contained less than 0.05 kg/m^3
Butterfly valve	Flow characteristics : Optional Rangeability (effective) : 15 Leak (% of rated capa.) : 3 to 5%	Applicable to high flow rate and low pressure drop services. Simple and economical, use for 4" and larger. Gas service with low pressure drop High viscous, slurry service Maximum opening is usually limited to 60 degrees for throttling. Do not use with opening of 10% or less as a control valve.
Ball valve	Flow characteristics : essentially EQ% Rangeability (effective) : 50 Leak (% of rated capa.) : 0.001	Applicable to high flow rate and high shut-off pressure services. Low resistance at full open Suitable as a shut-off valve Solid contained service High rangeability
Eccentric plug valve (Camflex)	Flow characteristics : globe/ball Rangeability (effective) : 100 Leak (% of rated capa.) : 0.001	Advantages over butterfly valves
Three-way valve	Flow characteristics : Rangeability (effective) : 10 Leak (% of rated capa.) : 0.01 to 0.5%	Splitting into two lines or mixing of two streams. For splitting flow : double seat valve For mixing flows : single seat valve Economical rather than two 2-way valves. Applicable for 4" and smaller.
Saunders valve	Flow characteristics : Special Rangeability (effective) : 10 Leak (% of rated capa.) : 0.001	Corrosive, slurry, high viscous service Bodies can be fully lined with corrosion resistant materials. Self cleaning ability Suitable as a shut-off valve

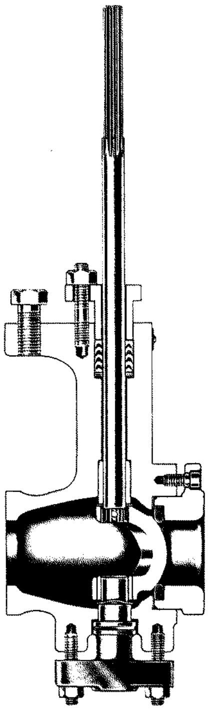
(3) The control valves of "globe type", "eccentric plug type (camflex type)" or "cage type" are used for the general throttling service, except for the following items (a) to (d).
   (a) Angle body control valves may be applied for the following applications:
      - Slurry service
      - High viscous service
      - Service that requires valve flushing to prevent coking or polymerization in the valve
   (b) Butterfly control valves may be applied for:
      - Large size piping application but to allow excessive leakage and limit of pressure drop.
   (c) Eccentric butterfly valve with soft seat may be applied for :
      - On-off service
   (d) Ball control valves may be applied for:
      - On-off shut-off service or for slurry service
   (e) Saunders (diaphragm) control valves may be applied for the following applications:
      - Corrosive service
      - Slurry service
      - High viscous service
      - Service that requires no stagnation in the valve body, such as sanitary service

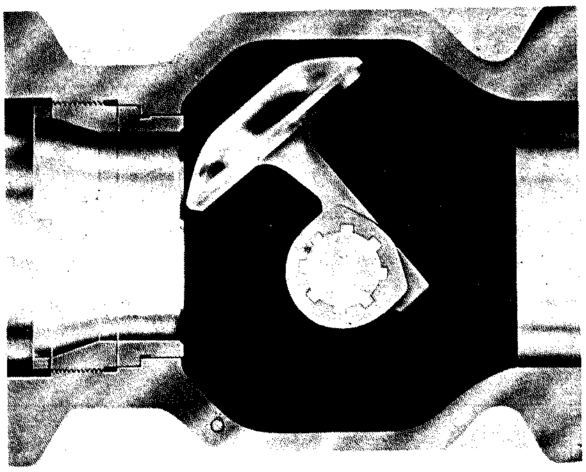
Fig. A2-1 Type of control valves

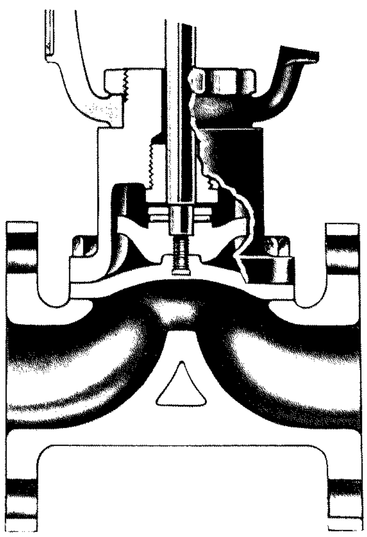

Globe Body Valve, Top Guided

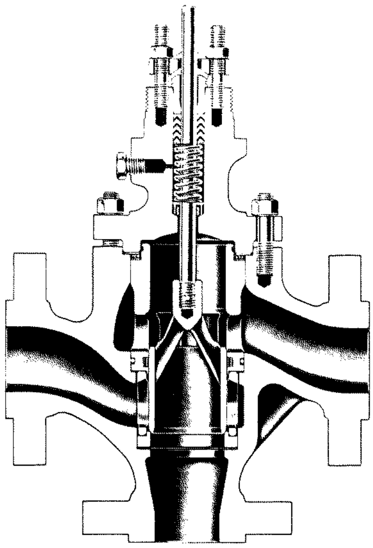

Angle Body Valve, Low Noise Trim


Butterfly Valve


Ball Valve


Eccentric Plug Valve


Saunders Valve


Three-Way Valve

APPENDIX-3 ESTIMATION OF CONTROL VALVE SIZE

Although estimation of control valve size is described herein, due to the many proprietary valve designs to vendors' own calculation method, valve sizing for a particular service will be conducted by the instrument engineer.

A3.1 Calculation of Flow Coefficient, Cv

The equations provided in this Appendix are Masoneilan's method that is based on ISA S75.01 "Control Valve Sizing Equations".

A3.1.1 Critical Flow Factor

(1) Assume valve type and select the critical flow factor, C_f, from Table A3-1 below.
If valve type is not determined, assume as $C_f = 0.85$.

(2) When valves mounted between pipe reducers, C_{fr}, R, C_{fr}/R will also be necessary in later calculations.

Table A3-1 Critical Flow Factor At Full Opening

Valve Type	Maisoneilan type	Flow To	C_f	C_{fr} (Note-1)	R (Note-1)	C_{fr}/R (Note-1)	K_c
Single seat, globe	21000 Series	Close	0.85	0.81	0.96	0.84	0.58
		Open	0.90	0.86	0.96	0.89	0.65
Double seat, globe	10000 Series	Contoured	0.90	0.86	0.96	0.89	0.70
		V-Port	0.98	0.91	0.96	0.95	0.80
Camflex	35002 Series	Close	0.68	0.65	0.95	0.68	0.35
		Open	0.85	0.80	0.95	0.84	0.60
Ball	36002 Series	Open	0.60	0.55	0.87	0.63	0.24
Butterfly	37002 Series	Either	0.65	0.60	0.81	0.74	0.32
LO-DB(Note-2)	41000 Series	Either	0.94	0.87	0.94	0.91	0.80

(Note-1) Reducers correction for D/d = 1.5. Refer to vendor's instruction for other than D/d = 1.5.
(Note-2) Low Noise Valve (Low Decibel)

A3.1.2 Liquid Service

(1) Turbulent liquid flow (Including flashing service)

	Liquid	
Check of Critical Flow	$\Delta P = P_1 - P_2$ $\Delta P_s = P_1 - \left(0.96 - 0.28\sqrt{\frac{P_v}{P_c}}\right)P_v$ If $\Delta P < C_f^2(\Delta P_s)$, flow is sub-critical. If not, flow is critical. If reducers are provided, check critical flow with $(C_{fr}/R)^2(\Delta P_s)$ instead of $C_f^2(\Delta P_s)$.	
Region	Sub-critical Flow	Critical Flow
Cv without Reducer	$C_v = 1.17q\sqrt{\frac{G_f}{\Delta P}}$	$C_v = \frac{1.17q}{C_f}\sqrt{\frac{G_f}{\Delta P_s}}$
Reducer Correction	$C_v = \frac{C_{v(\text{without reducers})}}{R}$	$C_v = \frac{C_f}{C_{fr}} \cdot C_{v(\text{without reducers})}$

(2) Laminar flow

Calculate turbulent and laminar Cv, use larger value as required Cv.

Turbulent flow	Laminar flow
$C_v = 1.17q\sqrt{\dfrac{G_f}{\Delta P}}$	$C_v = 0.0324 \cdot \left(\dfrac{\mu q}{\Delta P}\right)^{2/3}$

(3) Cavitation check

If $\Delta P < K_c \Delta P_s$, cavitation will not occur.

Where :

ΔP	=	actual pressure drop P_1 - P_2	(kg/cm^2)
P_1	=	upstream pressure	(kg/cm^2A)
P_2	=	downstream pressure	(kg/cm^2A)
ΔP_s	=	critical pressure drop for cavitation	(kg/cm^2)
P_c	=	critical pressure	(kg/cm^2A)
P_v	=	vapor pressure of liquid at flowing temperature	(kg/cm^2A)
C_f	=	critical flow factor, see Table A3-1	(-)
C_{fr}	=	critical flow factor, see Table A3-1	(-)
R	=	sub-critical flow capacity correction factor, see Table A3-1	(-)
C_V	=	valve flow coefficient	(-)
q	=	liquid flow rate	(m^3/h)
G_f	=	specific gravity at flowing temperature	(water = 1.0 at 15°C)
μ	=	viscosity	(cP)
K_c	=	coefficient of incipient cavitation, see Table A3-1	(-)
d	=	valve size	(mm)
D	=	line size	(mm)

Sample Calculation A3-1. : Liquid with specific gravity of 0.461 is flowing through a control valve at the rate of 32.5 m^3/h, inlet pressure of 33.1 kg/cm^2G and outlet pressure of 31.5 kg/cm^2G. Calculate the flow coefficient, assuming vapor pressure of 20.2 kg/cm^2A and critical pressure of 43.3 kg/cm^2A.

Solution : Assume single seat globe valve with reducer; C_{fr} = 0.81, R = 0.96, C_{fr}/R = 0.84, K_c = 0.58.

ΔP = (33.1+1.03) – (31.5+1.03) = 1.6 kg/cm^2

ΔP_s = (33.1+1.03) – (0.96–0.28 $\sqrt{20.2 / 43.3}$) (20.2) = 18.60 kg/cm^2

Flow is sub-critical, since ΔP =1.6 kg/cm^2 < $(C_{fr}/R)^2 \Delta P_s$ = $(0.84)^2$ (18.60) = 13.12 kg/cm^2

Cv = (1.17) (32.5) $\sqrt{0.461 / 1.6}$ = 20.4 without reducer correction.

Hence, Cv =20.4 / 0.96 = 21.3 with reducer correction.

A3.1.3 Gas and Steam Service with Conventional Valve

<table>
<tr><th></th><th colspan="2">Gas and Steam</th></tr>
<tr><td>Check of Critical Flow</td><td colspan="2">$\Delta P = P_1 - P_2$
If reducers are provided, check with C_{fr} instead of C_f .
If $\Delta P < 0.5 C_f^2 P_1$, flow is sub-critical. If not, flow is critical.
For sub-critical flow : $y = \frac{y_0}{C_f}\sqrt{\frac{\Delta P}{P_1}}$ ($\leq$ 1.5), y_0 = 1.40 for LO-DB valve
y_0 = 1.63 for other types.</td></tr>
<tr><td>Region</td><td>Sub-critical Flow</td><td>Critical Flow</td></tr>
<tr><td>Gas Cv without Reducer</td><td>$C_v = \frac{54.5W\sqrt{Z}}{C_f P_1 \sqrt{G_f}(y - 0.148y^3)}$</td><td>$C_v = \frac{54.5W\sqrt{Z}}{C_f P_1 \sqrt{G_f}}$</td></tr>
<tr><td>Steam Cv without Reducer</td><td>$C_v = \frac{83.7(1 + 0.00126T_{sh})W}{C_f P_1 (y - 0.148y^3)}$</td><td>$C_v = \frac{83.7(1 + 0.00126T_{sh})W}{C_f P_1}$</td></tr>
<tr><td>Reducer Correction</td><td>$C_v = \frac{C_{v(\text{without reducers})}}{R}$</td><td>$C_v = \frac{C_f}{C_{fr}} \cdot C_{v(\text{without reducers})}$</td></tr>
</table>

Where :

ΔP	=	actual pressure drop $P_1 - P_2$	(kg/cm^2)
P_1	=	upstream pressure	(kg/cm^2A)
P_2	=	downstream pressure	(kg/cm^2A)
C_f	=	critical flow factor, see Table A3-1	(-)
C_V	=	valve flow coefficient	(-)
W	=	flow rate	(1000 kg/h)
Z	=	compressibility factor	(-)
G_f	=	specific gravity at flowing temperature = G $\times$ 288 / (T+273)	(air = 1.0)
G	=	gas specific gravity	(air = 1.0)
T	=	flowing temperature	(°C)
T_{sh}	=	steam superheat	(°C)
C_{fr}	=	critical flow factor, see Table A3-1	(-)
R	=	sub-critical flow capacity correction factor, see Table A3-1	(-)

A3.2 Valve Size Selection

(1) Cv required, Cv_{req} , will generally be twice the calculated Cv for the normal flow, Cv_{cal} , i.e.:

$Cv_{req} = 2.0 \times Cv_{cal}$, for normal flow Cv

The Cv for the selected valve, Cv_{sel} , will be next larger Cv-value than the required Cv, Cv_{req} .

(2) Lift ratio of the control valve, which is described in Appendix-1, will preferably be between 50% - 70% for the normal flow conditions. (50% $\leq$ Lift% $\leq$ 70%)

(3) It is better to maintain the lift ratio over 10% (Lift% $\geq$ 10%) for the minimum flow conditions, taking higher pressure drop than those in normal conditions into account.

Table A3-2 Typical control valve flow coefficient, Cv (SAMPLE)

Single seat, globe, full or high capacity

Valve Size (in.)	Orifice Dia. (in.)	Cv
¾	0.812	12
1	0.812	12
1½	1.250	25
2	1.625	46
3	2.625	110
4	3.500	195
6	5.000	400

Double seat, globe, full area

Valve Size (in.)	Orifice Dia. (in.)	Cv
¾	0.875	8
1	1.063	12
1½	1.500	28
2	2.000	48
3	2.625	110
4	3.500	195
6	5.250	450
8	7.000	750
10	8.750	1160
12	10.500	1620
14	12.251	2000
16	14.000	2560

Camflex, full area

Valve Size (in.)	Orifice Dia. (in.)	Cv
1	0.719	14
1½	1.125	30
2	1.437	50
3	2.312	135
4	3.000	230
6	4.500	500
8	6.000	850
10	7.500	1300
12	9.250	1750

Ball

Valve Size (in.)	Orifice Dia. (in.)	Cv
-	-	-
-	-	-
2	1.66	105
3	2.50	250
4	3.33	525
6	5.00	1050
8	6.67	1950
10	8.33	3000
12	10.00	4400

Butterfly (MiniTork)

Valve Size (in.)	Orifice Dia. (in.)	Cv
2	-	54
3	-	180
4	-	390
6	-	1150
8	-	2050
10	-	3200
12	-	4600
14	-	5600
16	-	7400
18	-	9500
20	-	11800
24	-	17200

APPENDIX-4 NOISE ESTIMATION

A4.1 General

Major sources of noise generated in a control valve are :

(1) Mechanical vibration

Mechanical vibration is induced by the pulsation of the fluid (liquid or gas) passing through the valve, and can lead to resonance of the valve trim and fatigue failure of the valve stem, plug post, or other parts. Possible cures for this type of noise include reduction of guide clearances, larger stem size, change in plug mass, or sometimes reversal of flow direction. There is presently no reliable method for predicting noise generated by mechanical vibration in a control valve.

(2) Aerodynamic noise

Aerodynamic noise is the result of turbulent flow, and is propagated to the downstream and, to some degree, the upstream piping. As a result of the propagation, this noise is accompanied by the damaging effects of vibration and creation of potentially dangerous sound pressure levels in the surrounding atmosphere. The problem can be reduced by the use of one or a combination of the following:

- Specially designed valves that have curved paths or multiple orifices to reduce velocities.
- Several valves or valves and orifices in series.
- In-line silencers upstream and downstream.
- Heavy-walled piping.
- Acoustic insulation covering the affected piping.

Although the above last item can effect ambient noise level reduction, the potential for damage to the piping will not be reduced and the noise will still be propagated in the piping beyond the insulation.

(3) Hydrodynamic noise

Cavitation noise and flashing noise can be generated by the flow of liquid through a valve and piping. Cavitation is more serious than the other noise problems, since it will strongly limit the life of valve components and downstream piping.

A4.2 Aerodynamic Noise

Prediction of acoustic energy generated within a control valve is based on Jet noise theory. Basically, for subcritical conditions where the jet velocity at the vena contracta is below sonic, control valve noise is generated by the intense turbulence created in the shear layer downstream of the vena contracta. For critical conditions, additional noise is induced by the interaction between the turbulence and the shock waves developed by the critical flow velocity.

The aggregate quantity of sound energy generated, denoted by the sound power, can be given as follows:

$$PWL = k_1 \log \left[C_v \, C_f \, P_1 \eta \, T^{1/2} \right]$$

Where,
- PWL = Sound power level (dB)
- C_v = Actual required flow coefficient
- C_f = Critical flow factor at actual lift
- P_1 = Upstream pressure (Pa-a)
- η = Acoustical efficiency
- T = Absolute temperature (K)
- k_1 = Constant for determination of sound power level

This acoustic energy is transmitted into the downstream piping, and then re-radiated into surrounding environment. The sound intensity is reduced dramatically through this process. The reduction in sound intensity provided by the pipe walls is given as follows:

$$TL = -k_2 \log [P_2 D^3 T^{1/2} t^3]$$

Where, TL = Sound transmission loss (dB)
P_2 = Downstream pressure (Pa-a)
D = Downstream pipe diameter (m)
t = Pipe wall thickness (m)
k_2 = Constant for determination of transmission loss

Valve vendors have determined, based on their shop tests, k_1, k_2 and η for individual type and size of their valves. Thus the sound pressure level is unknown until process data are fixed and the valve vendor is selected.

Acoustic Engineering Team of HSE Systems Department keeps computer program for noise prediction of various control valves and can provide noise level data in the early phase of the engineering.

Process engineer should consult a noise control engineer when the process data indicate the excess of the following criterion:

$$L_C(\text{from Fig.A4.3}) + 10 \log(0.5\ m\ C^2) \geqq \text{Screening Criterion} = 85\ \text{dB(A)}$$

Where, m = Mass flow through the valve (kg/s)
C = Speed of sound in the gas at the valve (m/s)
= 91.2 (kT / MW)0.5
k : Ratio of the specific heats in the gas
T : Gas temperature (K)
MW : Molecular weight(kg/kgmol)

A4.3 Hydrodynamic Noise

No noise and/or vibration problems may occur in liquid service lines when the valve is selected properly. Contrary, when the valve is used under severe conditions, e.g. the downstream pressure goes down to vapor pressure of the liquid, significant noise and vibration will be generated with associated flashing and/or cavitation.

For the purpose of screening the following criterion may be convenient:

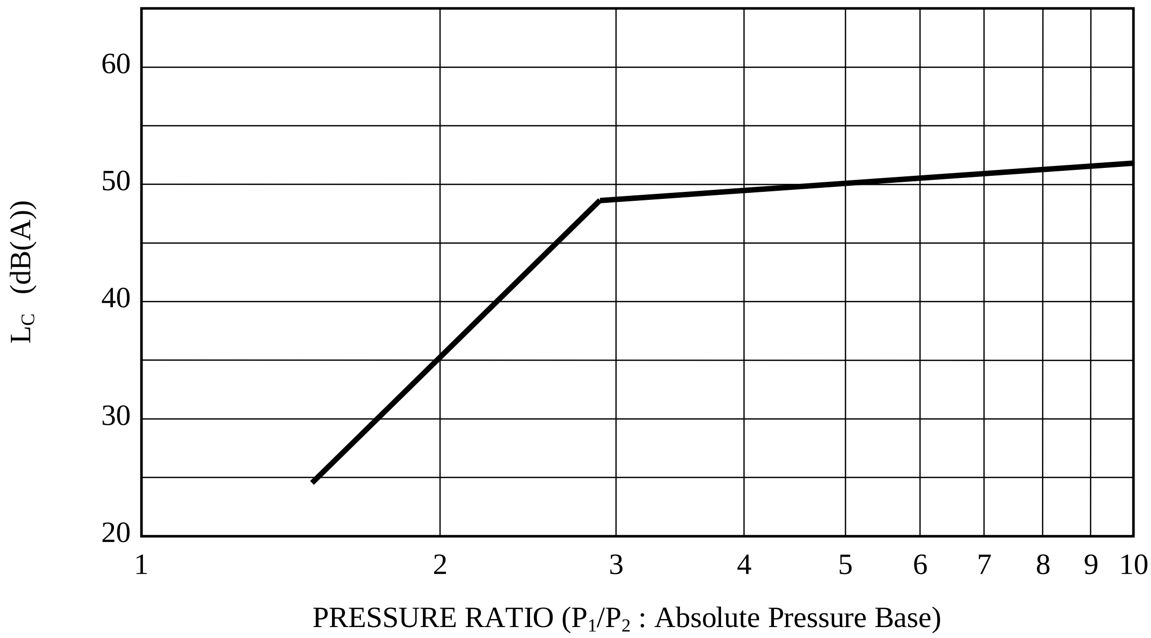
$$P_1 - 1.43 (P_1 - P_2) \geq P_v$$

Where, P_1 and P_2 are upstream and downstream pressure of the fluid (Pa-a),
P_v is vapor pressure, in Pa-a, of the liquid downstream of the valve.

Notes: The above equation assumes 0.7 as liquid pressure recovery factor (FL).
The actual value of FL depends on lift and type of valve.

When the left side of the above equation fails to reach P_v process engineer should consult and ask a valve vendor for precise study to avoid cavitation.

Fig. A4.3 Noise Intensity at 1m downstream of Control Valve and 1m from Pipe Surface



APPENDIX-5 TYPE OF ACTUATORS

(1) Although actuator type selection is described herein, due to the many proprietary designs to vendors' own criteria, selection of actuator type for a particular service will be conducted by the instrument engineer.

(2) Advantages and disadvantages for actuators of control valves are summarized as follows:

Table A5-1 Type of actuators

Type	Advantages	Disadvantages
Diaphragm type	Low cost Simplicity Inherent fail-safe action Low supply-pressure requirement Adjustability Maintainability Ability to throttle without positioner Fast stroking speeds possible	Limited torque availability Limited temperature range Inflexibility for changing service conditions Limited stroke
Cylinder type	High capability (large thrust, long stroke, etc.) Fast stroking-speed possible Relatively high actuator stiffness	Fail-safe requires higher cost Positioner required for throttling Higher cost High supply-pressure requirement
Electric motor type	Compact Suitable for remote applications Long stroke Large torque	High cost / torque ratio Lack of fail-safe action Limited throttling ability Slow stroking-speed Lack of adjustability
Hydraulic or Electro-hydraulic	High torque Very high actuator stiffness Excellent throttling stiffness Fast stroking-speed	High cost Complexity Large size and weight Fail-safe action requires accessories

(3) The diaphragm actuator is commonly used, due to its dependability and its simplicity of design.

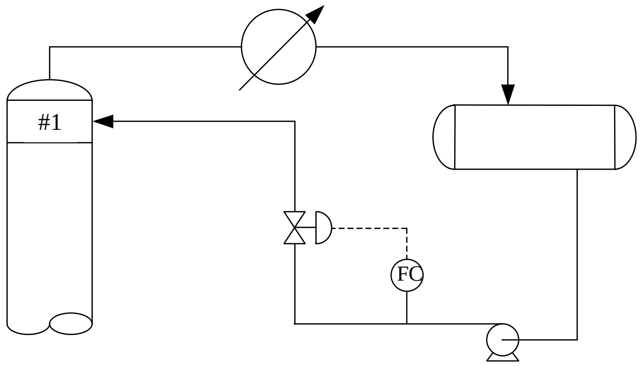
(4) The following guidelines are applicable to select the actuators, and will be detailed by the valve vendor.

(a) Power supply
The available power source of actuator at the location of the valve limits the type selection of actuator. Although compressed air is available in normal plant , other driving force for actuator should be used in local remote control plant without operator such as well head and pipeline valve station.

(b) Fail-safe action
Although the overall reliability of power sources is high, many processes require specific valve actions if the power source fails. Many actuators, such as the diaphragm type actuators, can incorporate required failure action without extra cost.

APPENDIX-6 EXAMPLE CALCULATION OF PRESS. DROP ACROSS CONTROL VALVE

A6.1 Reflux Line



Friction Loss(kg/cm^2)

Lines (Pump to tower)	0.5
Flow Meter	0.2
$\Sigma\Delta P_{fric}$ =	0.7

$\Sigma\Delta P_{fric} \le 5.0$ kg/cm^2

Therefore, $\Delta P_{CV} = 0.5\times\Sigma\Delta P_{fric}$

$= 0.5\times0.7$

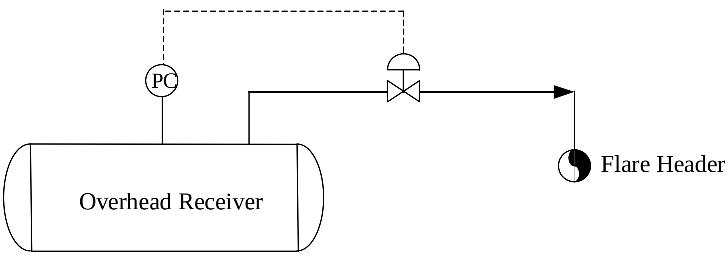
$= 0.35$ kg/cm^2

Accordingly, from minimum pressure drop for liquid service,

$\Delta P_{CV} = 0.7$ kg/cm^2

A6.2 Operating Pressure at Overhead Receiver

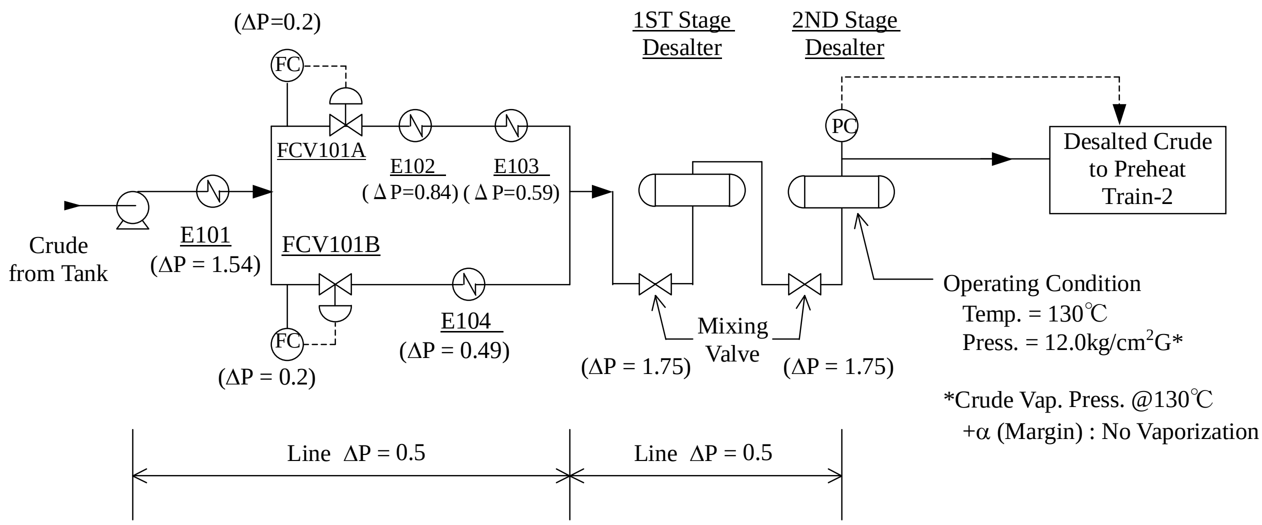
The procedure for determining the operating pressure in overhead receiver operated under the low pressure close to the atmospheric level are as follows :



Operating Pressure at Flare Header	0.3 kg/cm^{2}G
Line Pressure Drop (Receiver to Flare Header)	0.1 kg/cm^2
PCV (from minimum pressure drop)	0.2 kg/cm^2
Operating Pressure at Receiver	0.6 kg/cm^2 G

A6.3 Crude Supply System (Crude Distillation Unit)

A6.3.1 Preheat Train-1



(1) The route of FCV101A is governing, because the total friction loss of this is greater than the other.

Friction Loss (kg/cm^2)		
	E101, 102 & 103	2.97
	Lines	1.0
	Flow Meter	0.2
	Mix. Valve	3.5
	$\Sigma\Delta P_{fric}$ =	7.67

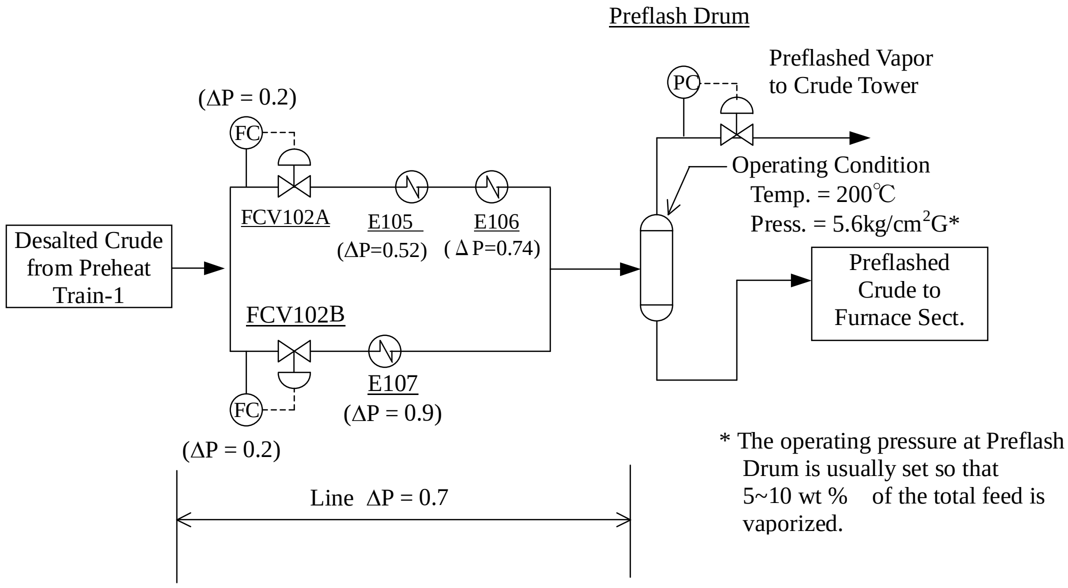
$6.25 kg/cm^2 < \Sigma\Delta P_{fric}$ $10.0 kg/cm^2$

Therefore, $\Delta P_{CV} = 0.4 \times \Sigma\Delta P_{fric}$
$= 0.4 \times 7.67$
$= 3.1 kg/cm^2$

(2) FCV101B $\Delta P_{CV} = 3.1 + (0.84 + 0.59 - 0.49)$
$= 4.04 kg/cm^2$

Note : In case of the multi-paths as shown in the above illustration and if the pressure drop among control valves is significantly different, reduction of the sizes of the lines whose paths are not governing should be recosidered.

A6.3.2 Preheat Train-2

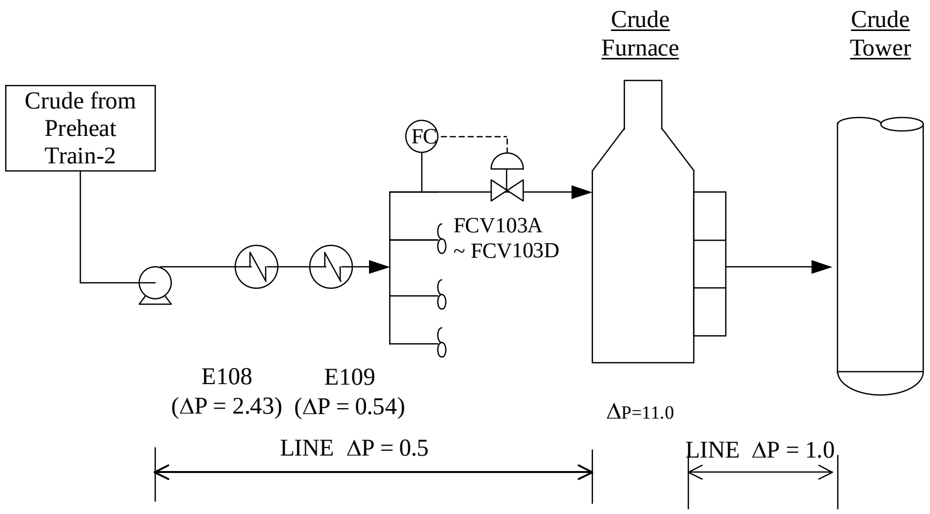


The pressure drops across FCV102A & 102B are obtained according to the pressure balance, since the operating pressures at inlet (2nd Stage Desalter) and outlet (Preflash Drum) are fixed.

(1) FCV102A 12.0 - (0.52 + 0.74 + 0.2 + 0.7) - 5.6 = $4.24 kg/cm^2$
(2) FCV102B 12.0 - (0.9 + 0.2 + 0.7) - 5.6 = $4.6 kg/cm^2$

Note : In case of the multi-paths as shown in the above illustration and if the pressure drop among the control valves is significantly different, reduction of the sizes of lines whose paths have greater ΔP_{CV} should be reconsidered.

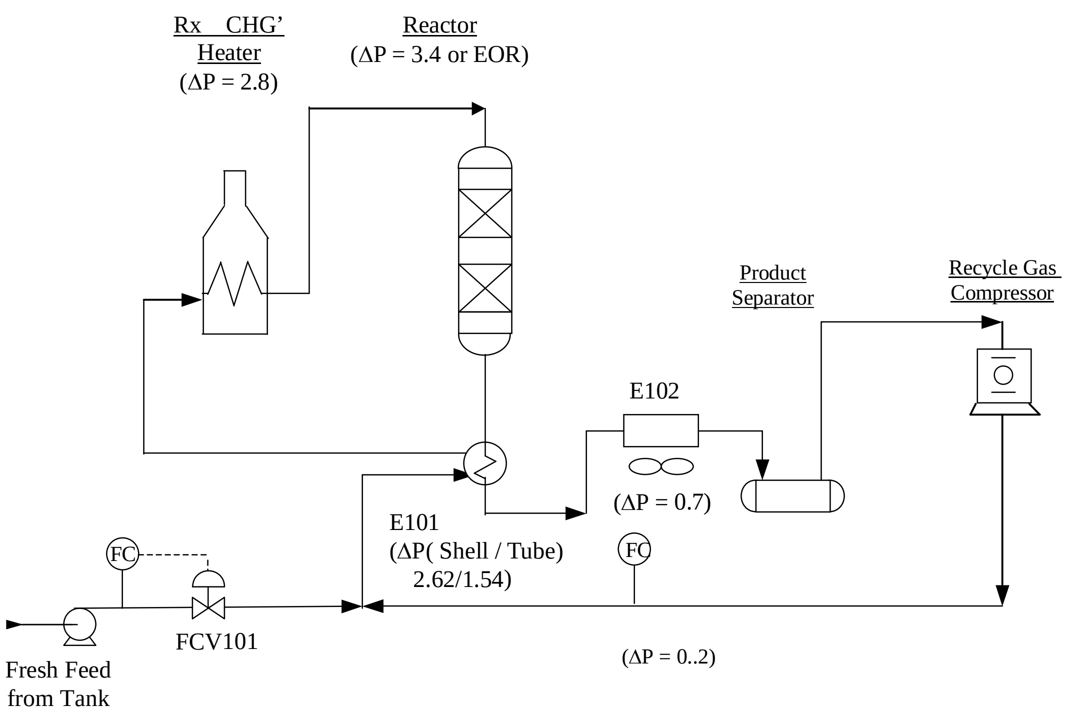
A6.3.3 Crude Furnace Section



Friction Loss(kg/cm^2)	E108 & 109	2.97
	Furnace	11.0
	Lines	1.5
	Flow Meter	0.2
	$\Sigma\Delta P_{fric}$ =	15.67

$13.4 kg/cm^2 < \Sigma\Delta P_{fric}$
Therefore, ΔP_{CV} for FCV103A ~ D = $0.3 \times \Sigma\Delta P_{fric}$ = $4.70 kg/cm^2$

A6.4 HDS Reactor Circuit



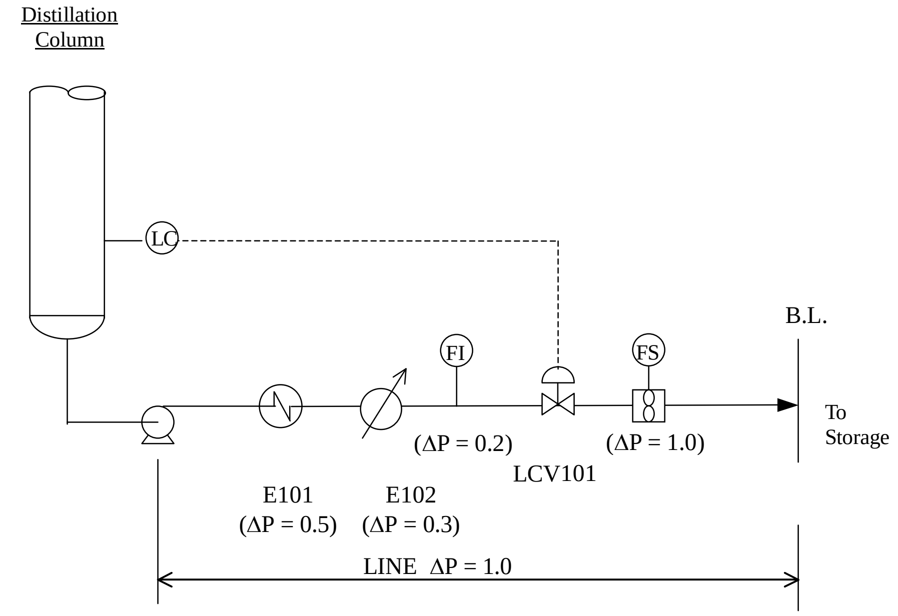
Friction Loss in Rx Circuit (kg/cm^2)

E101 & 102	4.86
Reactor	3.4
Heater	2.8
Flow Meter	0.2
Lines	2.5
$\Sigma\Delta P_{fric}$ =	13.76

$13.4 kg/cm^2 < \Sigma\Delta P_{fric}$
Therefore, ΔP_{CV}(FCV101) = $0.3 \times \Sigma\Delta P_{fric}$ = $4.13 kg/cm^2$

Notes : In case where the pressure drop at EOR status is specified, indicate that the control valve should be designed based on the opening of 85 ~ 90% to avoid oversizing.

A6.5 Product Rundown



Friction Loss (kg/cm^2)

E101 & 102	0.8
Flow Meter	0.2
Turbine Meter	1.0
Lines	1.0
$\Sigma\Delta P_{fric}$ =	3.0

$\Sigma\Delta P_{fric}$ 5.0 kg/cm^2

Therefore, ΔP_{CV} for LCV101 $= 0.5\times\Sigma\Delta P_{fric}$

$= 0.5\times3.0$

$= 1.5$kg/cm^2

Notes : In case where B.L. pressure is given and includes enough margin, indicate that the control valve should be designed based on the opening of 85 ~ 90% to avoid oversizing.

APPENDIX-7 EXAMPLE OF SHUTOFF PRESSURE

The equations (1),(2), P_{down} and MNOP quoted in the illustrations are as follows :

$\Delta P_{shut} = P_{up}$ (+ 1.0 : when the downstream pressure is vacuum) (kg/cm^2) ? ? ? ? ? (1)

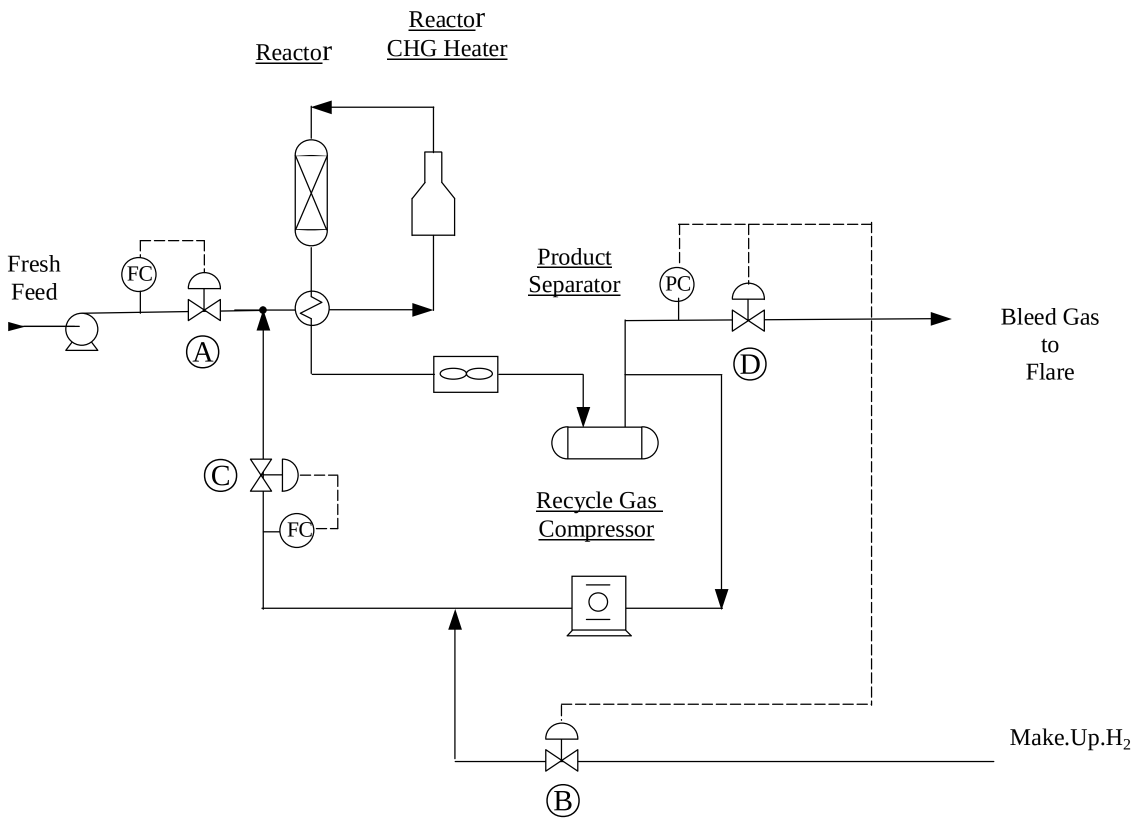
$\Delta P_{shut} = P_{up} - P_{down}$ (kg/cm^2) ? ? ? ? ? (2)

Where :

ΔP_{shut}	=	Shutoff pressure	(kg/cm^2)
P_{up}	=	Design pressure of upstream line	(kg/cm^2G)
P_{down}	=	Minimum normal operating pressure of downstream line	(kg/cm^2G)

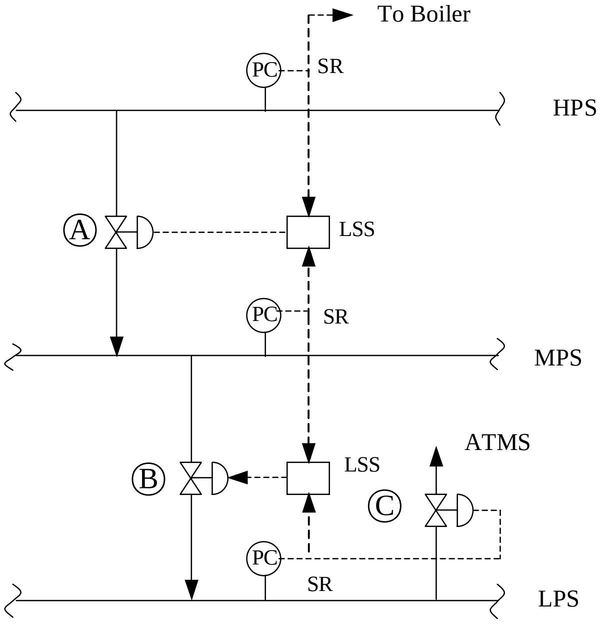
MNOP(Minimum normal operating pressure) : minimum operating pressure during normal operation.

A7.1 High Pressure Reactor Circuit



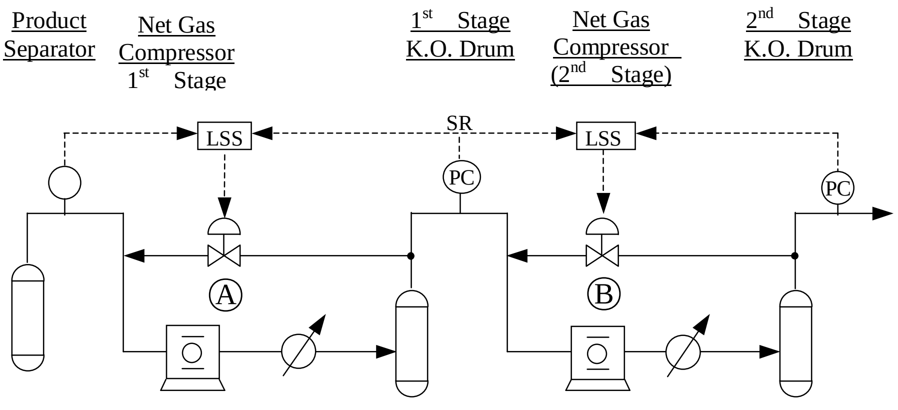
CV	Service	Equation	P_{down}
Ⓐ	Fresh Feed	(2)	MNOP at Prod. Sep.
Ⓑ	Make Up H_2	(2)	MNOP at Prod. Sep.
Ⓒ	Recycle Gas	(2)	MNOP at Prod. Sep.
Ⓓ	Bleed Gas	(1)	—

A7.2 Steam Let-Down Valve



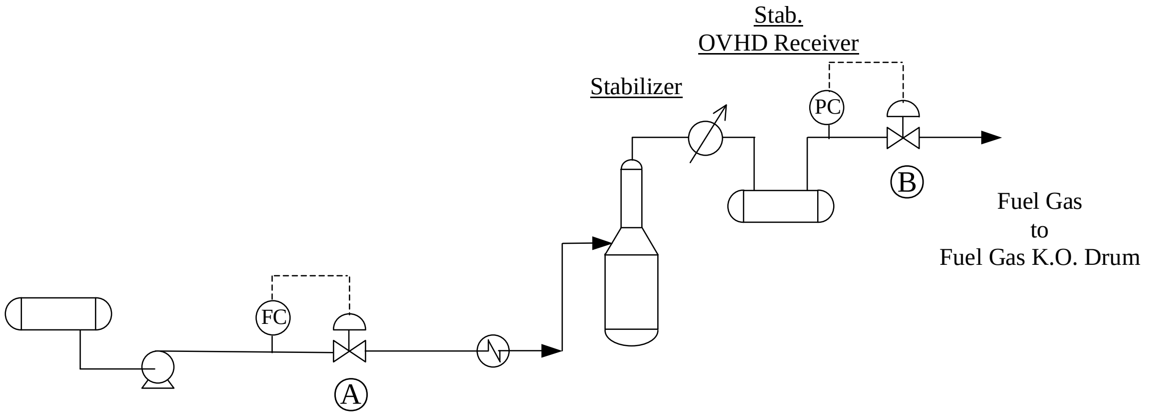
CV	Service	Equation	P_{down}
Ⓐ	HPS to MPS	(2)	MNOP of MPS
Ⓑ	MPS to LPS	(2)	MNOP of LPS
Ⓒ	LPS to ATMS	(1)	—

A7.3 Compressor Spill Back



CV	Service	Equation	P_{down}
Ⓐ	1ST Stage Spill Back	(2)	MNOP at Prod. Sep.
Ⓑ	2ND Stage Spill Back	(2)	MNOP at 1^{ST} Stage K.O. Drum

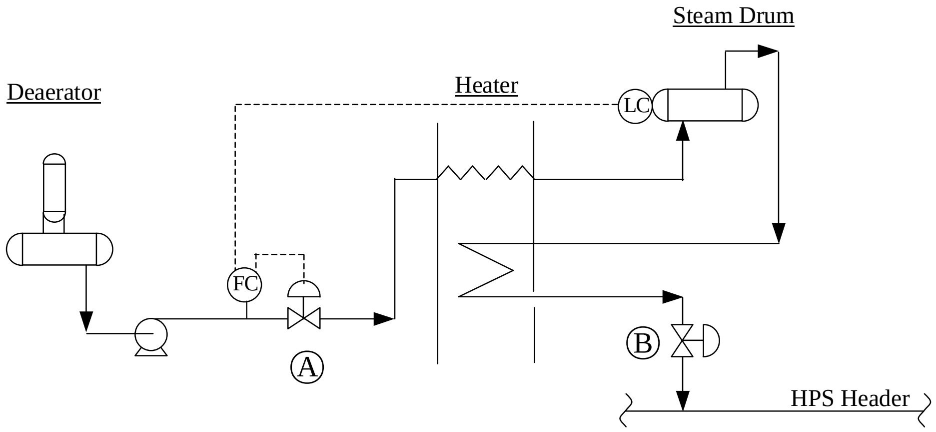
A7.4 High Pressure Tower Feed (EX. Stabilizer)



CV	Service	Equation	P_{down}
Ⓐ	Feed	(2)	MNOP at Stab. OVHD Receiver
Ⓑ	Fuel Gas	(2)	MNOP at Stab. OVHD Receiver

The same procedure as above is applicable to the followings :
(1) Lean Amine to Absorber
(2) Sponge Absorber Feed
(3) LPG Extractor

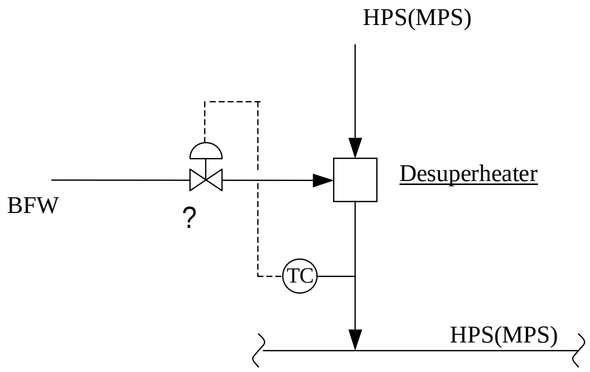
A7.5 BFW Injection to Boiler & WHB



CV	Service	Equation	P_{down}
Ⓐ	BFW	(2)	MNOP at Steam Drum
Ⓑ	HPS	(2)	MNOP of HPS

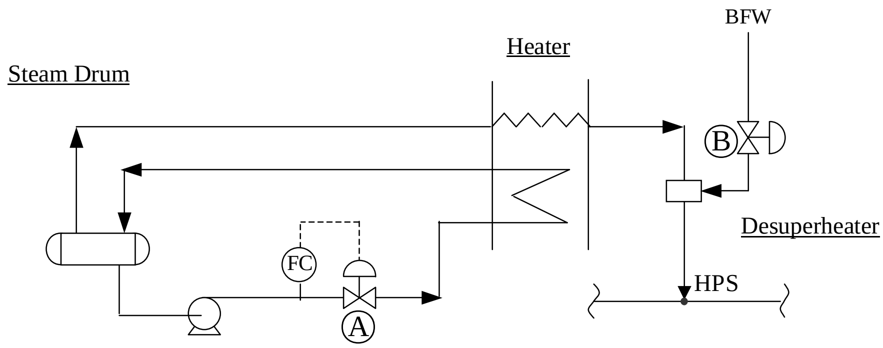
* If Steam Drum pressure is controlled.

A7.6 BFW to Desuperheater



CV	Service	Equation	P_{down}
Ⓐ	BFW	(2)	MNOP of HPS (MPS)

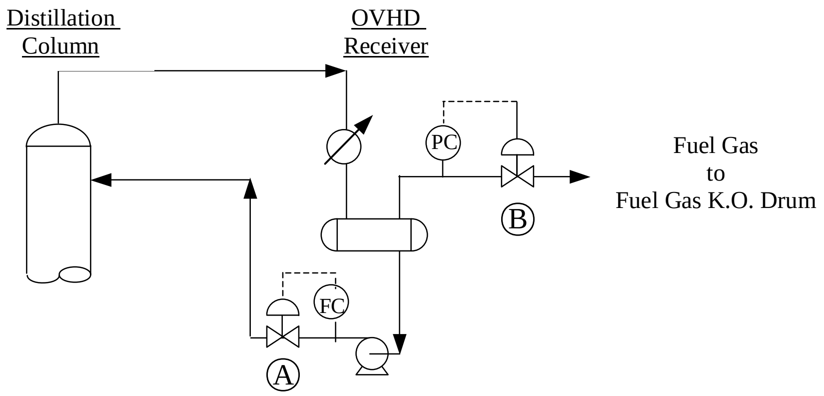
A7.7 BFW Circulation on Steam Generation



CV	Service	Equation	P_{down}
Ⓐ	BFW Circulation	(2)	MNOP at Steam Drum
Ⓑ	BFW	(2)	MNOP of HPS

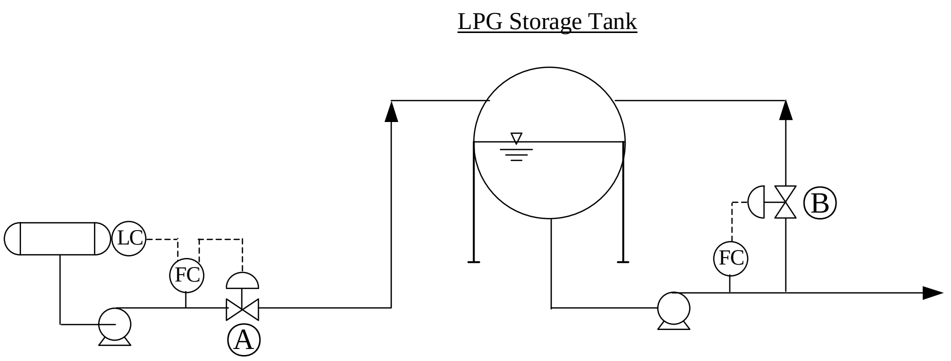
* If Steam Drum pressure is controlled.

A7.8 Reflux Line



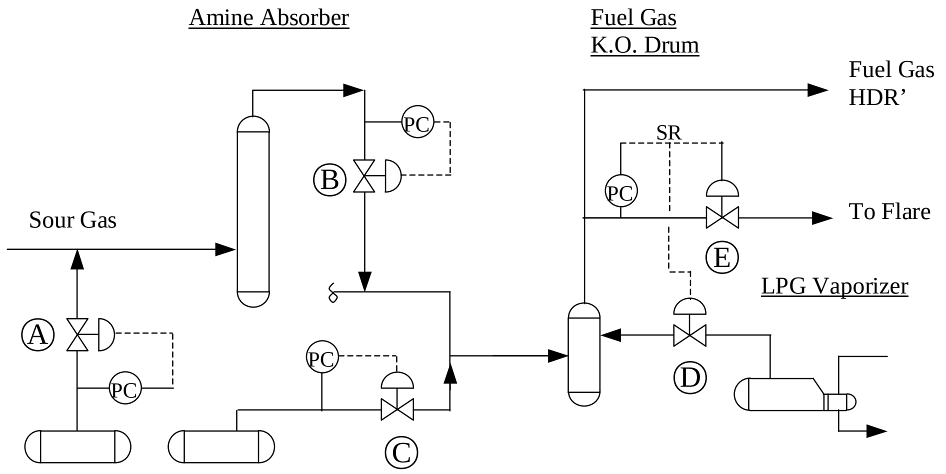
CV	Service	Equation	P_{down}
Ⓐ	Reflux	(2)	MNOP at OVHD Receiver
Ⓑ	Fuel Gas	(2)	MNOP at Fuel Gas K.O. Drum

A7.9 Pressurized Tank



CV	Service	Equation	P_{down}
Ⓐ	LPG Prod.	(2)	LPG vapor pressure at minimum ambient temperature
Ⓑ	Pump Min. Flow	(2)	LPG vapor pressure at minimum ambient temperature

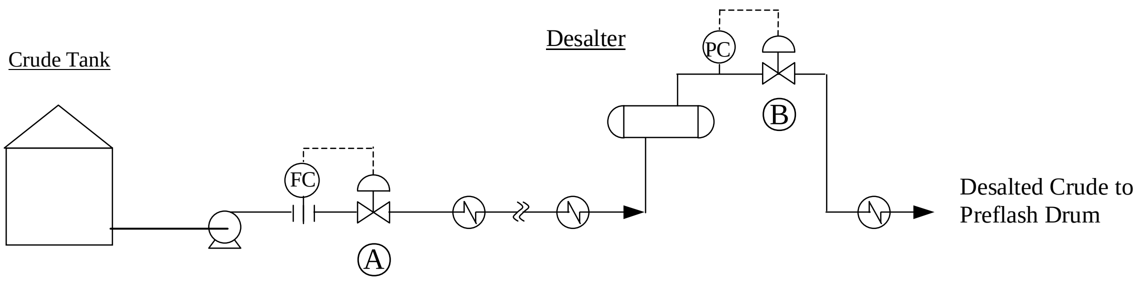
A7.10 Off Gas to Fuel Gas System



CV	Service	Equation	P_{down}
Ⓐ	Sour Gas	(2)	MNOP at Amine Absorber
Ⓑ	Treated Gas	(2)	MNOP at Fuel Gas K.O. Drum
Ⓒ	Fuel Gas	(2)	MNOP at Fuel Gas K.O. Drum
Ⓓ	LPG Vapor	(2)	MNOP at Fuel Gas K.O. Drum
Ⓔ	Bleed Gas	(1)	—

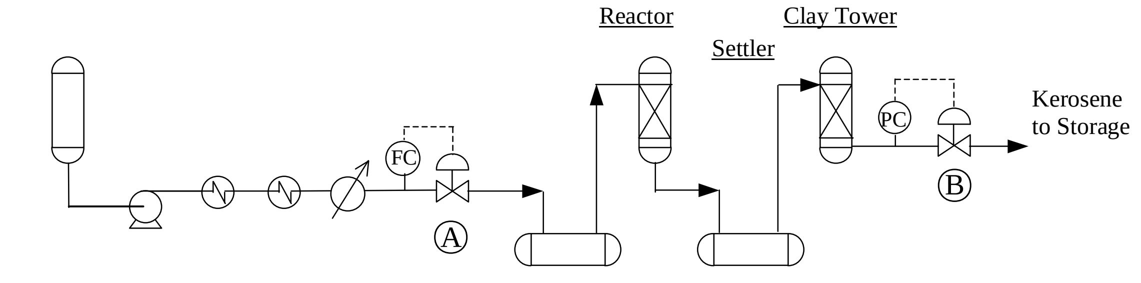
A7.11 Full Liquid Service with PC

(1) Desalter



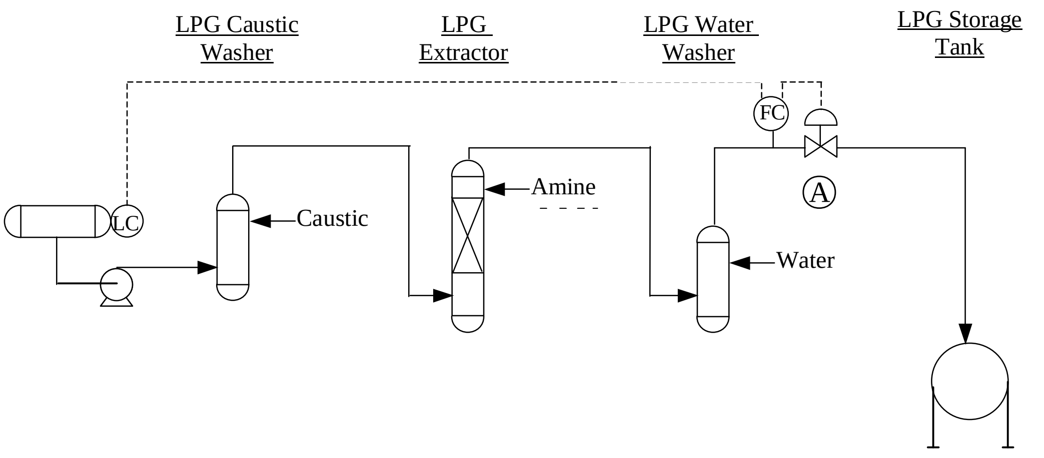
CV	Service	Equation	P_{down}
Ⓐ	Crude	(2)	MNOP at Desalter
Ⓑ	Desalted Crude	(2)	MNOP at Preflash Drum

(2) Kerosene MEROX



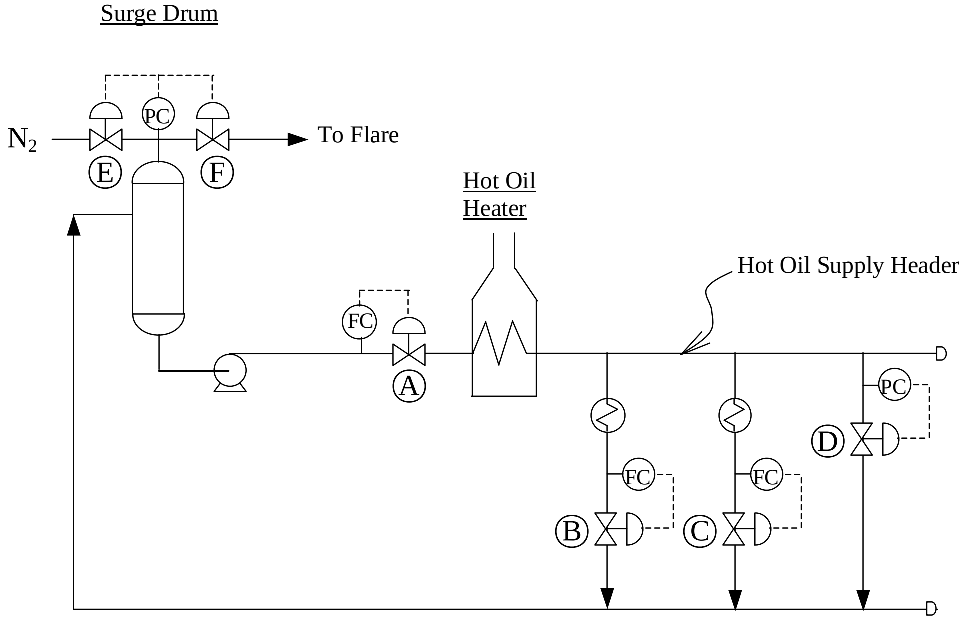
CV	Service	Equation	P_{down}
Ⓐ	Kerosene	(2)	MNOP at Clay Tower
Ⓑ	Treated Kerosene	(1)	—

(3) LPG Treating



CV	Service	Equation	P_{down}
Ⓐ	Treated LPG	(2)	LPG vapor pressure at minimum ambient temperature

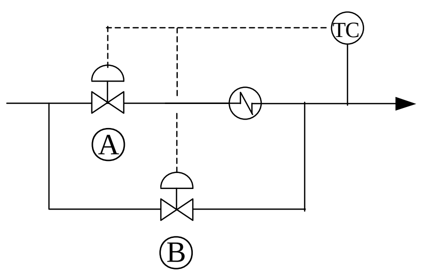
A7.12 Hot Oil Circuit with Pressurized Surge Drum



CV	Service	Equation	P_{down}
Ⓐ	Hot Oil Supply	(2)	MNOP at Supply Header
Ⓑ	Hot Oil Return	(2)	MNOP at Surge Drum
Ⓒ	Hot Oil Return	(2)	MNOP at Surge Drum
Ⓓ	Hot Oil Circulation	(2)	MNOP at Surge Drum
Ⓔ	N_2 Seal	(2)	MNOP at Surge Drum
Ⓕ	Bleed Gas	(1)	—

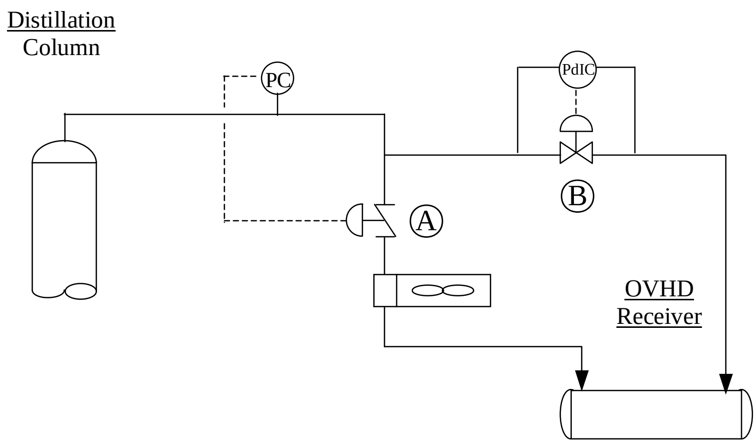
A7.13 Special System

(1) Heat Exchanger By-Pass



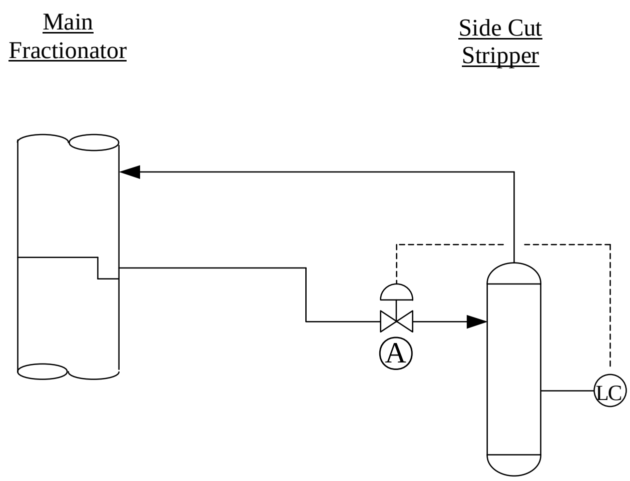
CV	Service	Equation	P_{down}
Ⓐ	To H/E	(2)	MNOP at downstream equipment
Ⓑ	H/E By-Pass	(2)	MNOP at downstream equipment

(2) Hot By-Pass of Total Condenser



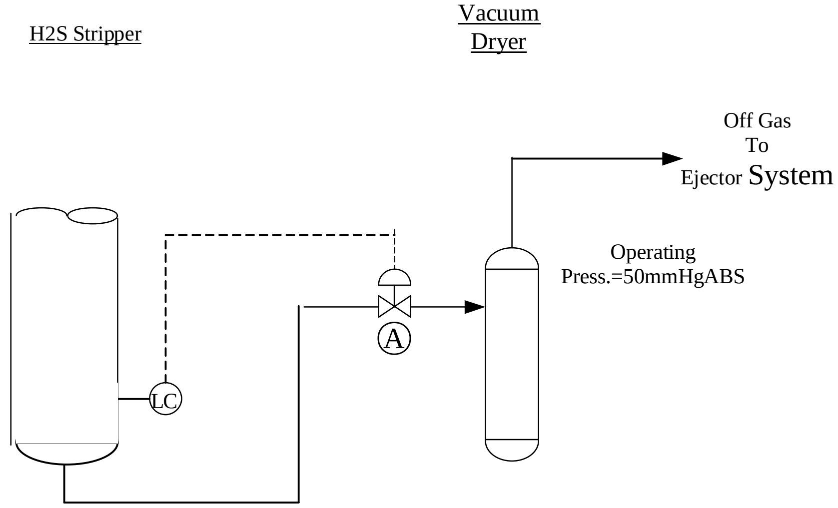
CV	Service	Equation	P_{down}
Ⓐ	OVHD Vapor	(2)	MNOP at OVHD Receiver
Ⓑ	By-Pass	(2)	MNOP at OVHD Receiver

(3) Side Cut Stripper Feed



Shutoff pressure of Ⓐ should be 2.0kg/cm^2 (maximum difference of static head (1.0Kg/cm2)+ margin (1.0Kg/cm2)).

(4) Vacuum Dryer



CV	Service	Equation	P_{shut}
Ⓐ	Gas Oil Product	(1)	P_{up}+1.0