ENGINEERING PRACTICE

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SPECIAL FEATURES

Hydrostatic Pressure Testing of Piping

IACPE
INTERNATIONAL ASSOCIATION OF CERTIFIED PRACTICING ENGINEERS

WWW.IACPE.COM

ENGINEERING PRACTICE

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ABOUT

International Association of Certified Practicing Engineers provides a standard of professional competence and ethics. Identifies and recognizes those individuals that have meet the standard. And requires our members to participate in continuing education programs for personal and professional development.

In additional to insuring a professional level of competency and ethics the IACPE focuses on three major areas of development for our members: Personal, Professional, and Networking.

HISTORY

The International Association of Certified Practicing Engineers concept was formulated by the many young professionals and students we meet during our careers working in the field, running training courses, and lecturing at universities.

During question and answer sessions we found the single most common question was: What else can I do to further my career?

We found, depending on the persons avail able time and finances, and very often dependent on the country in which the person was from, the options to further ones career were not equal.

Many times we found the options available to our students in developing countries were too costly and or provided too little of value in an expanding global business environment.

The reality is that most of our founders come from countries that require rigorous academic standards at four year universities in order to achieve an engineering degree. Then, after obtaining this degree, they complete even stricter government and state examinations to obtain their professional licenses in order to join professional organizations. They have been afforded the opportunity to continue their personal and professional development with many affordable schools, programs, and professional organizations. The IACPE did not see those same opportunities for everyone in every country.

So we set out to design and build an association dedicated to supporting those engineers in developing in emerging economies.

The IACPE took input from industry leaders, academic professors, and students from Indonesia, Malaysia, and the Philippines. The goal was to build an organization that would validate a candidates engineering fundamentals, prove their individuals skills, and enhance their networking ability. We wanted to do this in a way that was cost effective, time conscience, and utilized the latest technologies.

MISSION

Based on engineering first principles and practical real world applications our curriculum has been vetted by academic and industry professionals. Through rigorous study and examination, candidates are able to prove their knowledge and experience. This body of certified professionals engineers will become a network of industry professionals leading continuous improvement and education with improved ethics.

VISION

To become a globally recognized association for certification of professional engineers.

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KNOWLEDGE. CERTIFICATION. NETWORKING

LETTER FROM THE PRESIDENT

KARL KOLMETZ

Make 2017 Your Best Year Ever
We have turned a new page, we have a new opportunity to make 2017 our best year ever. What are some thing we can do to make this our best year?

A Good First Step is to Celebrate 2016
What did you accomplish in 2016? What were some special moments in 2016 that you will treasure the rest of your life? Do not get so fast paced in life that you overlook the small treasures. What are you most grateful for? Make a list of good things of 2016.

My grandson was just old enough to begin to understand families. He understood his father, his mother and his brother were his family. His father brought him and his brother to visit me on the farm. My grandson asked me, "Papa K where is your family?" My son answered in less than one second, "Jackson, we are his family." That is a simple small moment that I will treasure the rest of my life.

Bring the Good Forward and Throw Away Yesterday's Ashes
New Year's is a time when people reflect on the past. What worked well and what did not work so well? What should I improve and what should I leave behind? Many of us have a charcoal grill. If we want to have a good fire in the grill today, we must throw away yesterday's ashes. But people bring the hurts forward with them into the new year, in fact some people treasure the ashes for multiple years. You need to throw away yesterday's ashes.

Knowledge
What did you learn in 2016? What are the 3 best lessons you learned in the past 12 months? What path and technique was the most successful? What were your best practices?

Values
You cannot really set goals for the new year until you decide what are your values? Establish the values you wish for your year and life, then set your 2017 goals. Clarity proceeds mastery. Clarify your values and then master your goals.

Be Thankful
Practice a habit of thanking people around you, at work and elsewhere. It is so nice to recognize people and to be recognized.

All the Best in 2017,
Karl

NEWS

Busy End to 2016

IACPE had busy end to 2016. In October IACPE was officially registered in the Ministry of Justice and HAM by deed No. AHU-0076099.AH.01.07, thus our presence in Indonesia has been recognized by the Government of Indonesia.

IACPE President Karl Kolmetz came to Indonesia for MOA and MOU Signing Ceremony in November at Wahid Hasyim University, Semarang, Central Java and continued with a seminar “Career Guidelines” at 17 Agustus 1945 University, Semarang and to gave a CPE I certificate (Certified Practicing Engineer Level I) for fifteen students who finished their IACPE Level Icertification.

Hydrostatic Pressure Testing of Piping

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Introduction

Pressure testing of a new pipe is required prior to commissioning to prove its integrity at operating pressure. Hydrostatic testing is the most common pressure testing method. Testing of an in-service pipeline may be done as part of a preventative program to verify pipe integrity. In-service pipelines may also be tested if operating pressure are to be increased, modifications to the pipe are made or a change in line service planned.

Hydrostatic pressure testing is universally known and accepted as a means of demonstrating the fitness of a pressurized component for pipe service. After a test, a pipe can be expected to safely contain its intended operating pressure. The confidence level that a pipe or pressure vessel is fit for safe service increases as the ratio of test pressure to operating pressure increases. Hydrostatic test reveals weaknesses of pipe by causing ruptures or leaks.

When compared to other equipment in a hydrocarbon processing plant, the piping network is designed to the most stringent standards. Mechanical Engineering codes require a 400% safety factor in the design of these systems. The piping system is normally considered the safest part of the plant. However, even with this level of safety, reviews of catastrophic accidents show that piping system failures represent that largest percentage of equipment failures.

Since these systems are responsible for many catastrophic accidents, operations, design, and maintenance personnel should understand the potential safety concerns. Failure of an operating piping can result in health and safety concerns, damage to property and has the potential for significant environmental impact. Consequently, it is important to ensure that a pipe is free of leaks and is capable of maintaining its integrity at an approved operating pressure in order to limit the risk to the public and the environment.

In some countries, approval from regulatory agencies must be acquired prior to testing. regulatory approvals have been put in place to minimize the risk of unacceptable environmental impact or adverse impacts on other water users as a result of testing activities.

This test has a lot of considerations that must be considered (before or after the test) to obtain maximum result. The purpose of this article is to show how to do the hydrostatic pressure testing in accordance with the steps, procedures and rules.

Definition of Hydrostatic Pressure Test :

Hydrostatic pressure testing involves the filling of a section of pipe to be tested with water, adding additional water to the piping until the desired test pressure is reached and maintaining the pressure in the pipe for a period specified by regulatory authorities.

Many piping standards are using hydrostatic pressure testing in order to prove the integrity of the pipe and welds to the owner company, regulatory authorities and the public. This procedure is conducted on new pipelines as well as on in-service pipelines when a change of service is proposed, an increase in operating pressure is planned or to verify the integrity of the piping.

Hydrostatic pressure testing of new pipe is undertaken following completion of backfilling. Prior to filling the pipe with a water, a cleaning pigs must be run through the test section to remove any debris (e.g. welding litter, dirt) from the pipe. The pipe section to be tested is then filled with test water.

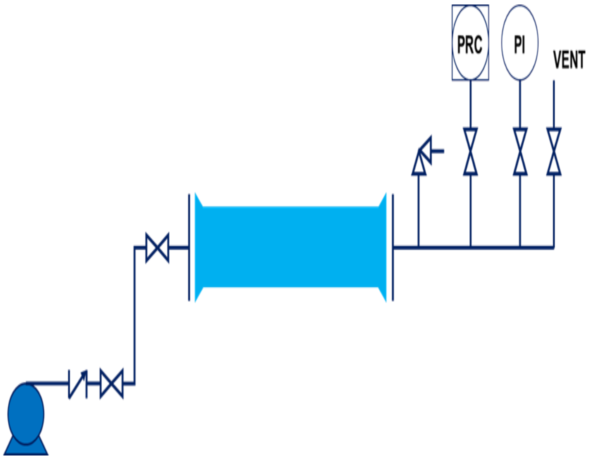


Figure 1 : **Hydrostatic pressure testing of pipe.**

The volume of water required for a test is dependent upon the length of the test section and diameter of the pipe (see table).

Table : Hydrostatic test water volume requirements for standard pipe size.

Outside Diameter		Wall Thickness (*mm*)	Fluid Volume (m^3/*km*)
(*mm*)	(*inch*)		
60.3	2	3.2	2.3
88.9	3	3.2	5.3
114.3	4	3.2	9.1
168.3	6	4.0	20.2
219.1	8	6.4	33.4
273.1	10	6.4	53.2
323.9	12	7.9	74.6
406.4	16	9.5	117.9
508.0	20	12.7	182.9
559.0	22	12.7	223.6
609.6	24	12.7	268.4
762.0	30	12.7	426.1
813.0	34	12.7	487.2
914.4	36	12.7	620.2
1067.0	42	12.7	852.1
1219.0	48	12.7	1118.9

Water sources commonly include rivers, lakes, ponds, dugouts, borrow pits, wells and municipal water supplies. Isolation valves may be used to break long sections of new pipe into smaller test sections that vary in length depending upon the topography traversed and construction season. Alternatively, the pipe may be cut and test heads welded on to allow testing.

Water is reused along a pipeline from one test section to another in order to minimize water requirements. Since the test section of an operating pipe may be downstream from the nearest terminal or fill point, the water may be required to travel along the pipe for a considerable distance prior to reaching the test section.

Water for testing and flushing shall be clean and free from any suspended or dissolved substances which could be harmful to the pipe material or internal coating (where applied) or which could form deposits within the pipeline, or which may be unacceptable at the disposal location.

Care shall be taken to insure the use of clean water for hydrostatic tests and the sea water is prohibited to be used. Hydrostatic testing shall be performed using potable water with a chloride content of max 50 ppm for austenitic steels.

Planning of a hydrostatic test program involves the selection of an appropriate test water source. Ideally the source water should be:

- of high quality
- available in large volumes
- located near the optimum fill location
- accessible with a minimum of disturbance
- within the same drainage basin as the discharge point
- economical

The potential exists during water withdrawal to adversely affect aquatic life, soils and land use. The degree of risk to these environmental components can be influenced by the:

- source water withdrawal rate
- volume withdrawn
- timing
- location and sensitivity of the withdrawal point
- activity needed to prepare, use and abandon the withdrawal site

The main objective of analyzing selection the source water is to confirm that substances that could pose a discharge problem are not being introduced. Surface water or groundwater may be tested for total dissolved solids, salts (electrical conductivity, sodium absorption ratio), pH, trace metals and suspended solids. The selection of a test water source is also dependent upon the ability to obtain approval from regulatory agencies and the landowner.

Tested piping shall be internally cleaned to remove all remaining dust and foreign matter by water flushing or blowing with air. Flushing shall be done with clean water using hydrostatic test water where possible. Water used for special flushing or cleaning of austenitic stainless steel shall not contain more than 50 ppm chlorides, i.e. Where special treatment is required, such as cleaning of compressor suction and lube oil piping, a separate procedure shall be prepared by contractor's Operations/ Commissioning Group.

Equipment of Testing :

In hydrostatic test, all personnel within test are a shall be required to wear eye and hearing Personnel Protective Equipment (PPE). Furthermore, there are some equipment used in hydrostatic pressure test as follows:

- Pressure pump
- Pressure recorder
- Temperature recorder
- Deadweight pressure gauge
- Pressure gauges
- Stroke counter
- Hydrostatic test yield plot
- High pressure hoses
- Check valves
- Fill pipe
- Filter
- Water samples

Preparation of Testing :

All joints, including welds, shall be accessible and left uninstalled, unpainted and exposed for examination during the test. Joints previously tested in accordance with this specification may be insulated or covered.

Piping designed for vapor or gas shall be provided with additional temporary supports, if necessary, to support the weight of the test liquid. Where required, temporary supports shall be specified in the pressure test documents.

Lines which are counterweight supported shall be temporarily blocked during testing in order to sustain the weight of the test fluid. Spring hangers which have been provided with stops for carrying the test load normally do not require additional temporary supports: If this is not the case, temporary support must be provided before filling the system.

Procedures of Testing :

The hydrostatic pressure testing operations shall be carried out by an experienced test engineer who shall have no other duties during the hydrostatic testing operations. The test engineer shall be in full charge of all activities related to the hydrostatic testing operations.

The test engineer shall prepare a procedure for all pressure testing operations, including a statement of the responsibilities of his subordinates. The test procedure shall be submitted to the Principal for approval not less than 3 days prior to commencement of testing or as specified in the scope of work. No part of the hydrostatic pressure testing operations shall commence until the approval is given in writing.

The test procedure shall include, but not be limited to:

- List of nominated personnel who are to supervise the pressure testing operations with their qualifications, tasks, responsibilities and authorities.
- Detailed schedule giving proposed dates of the main activities, tests and mobilization dates of the nominated personnel.
- Details of the selected test sections, including assemblies and pre-test sections.
- Identification of potential safety and environmental hazards, including the necessary measures and emergency plans.
- Details of the line-fill water, including the source, treatment method, discharge/disposal and permits.
- Details of the test equipment, including layouts and size and/or performance.
- Details of the test section preparation, including cleaning, gauging and filling.
- Details of the hydrostatic pressure test preparation, including temperature stabilization period.
- Details of the hydrostatic pressure tests, including pre-test, strength test and leak tightness test.
- Details of the post-testing activities, including depressurizing and documentation.
- Details of the post-testing activities, including depressurizing and documentation.

The normal location for the pressure test gauge is at grade near the pressure test pump and at the highest point of the piping system. Readings may be made at higher points providing the gauge pressure reading plus the static head between grade and the point of measurement does not exceed the maximum test pressure.

Piping systems shall be filled from a low point and filtered with a 10 micron filter. During filling of the system all air or gaseous substance shall be vented from high point to the maximum extent possible. Hydrostatic test water will be discharged to the nearest storm water drains.

Hydrostatic test pressure shall be maintained for a sufficient length of time to visually determine if there are any leaks, but not less than ten (10) minutes. Test pressure shall not be required to be maintained in excess of two hours after notification to contractor.

After completion of the test, pressure shall be released by opening the valve gradually so as not to endanger personnel or damage equipment. As a rule of thumb, pressure releasing rate shall be less than 300 $kg/cm^2/hr$. For piping line for which pressure releasing rate is to be specified, refer to specific job requirements. After completion of the test, the piping and equipment shall be drained completely.

Pressure of Testing :

The minimum of hydrostatic test of piping designed for internal pressure at any point in the system shall be as follows :

- Not less than 1-1/2 times of the design pressure.
- For a design temperature above the test temperature, the minimum test pressure shall be as calculated by the following equation.

$$Pt = \frac{1.5 . P . St}{S} \qquad \text{Eq. 1}$$

Where:

Pt = minimum calculated hydrostatic test pressure (kg/cm^2)
P = internal design pressure (kg/cm^2)
St = allowable stress at test temperature (kg/cm^2)
S = allowable stress at design temperature (kg/cm^2)
See Table I, Appendix A, ASME B31.3
When St and S are equal, test pressure is 1.5 x P.

Where the test pressure as defined minimum yield strength at test temperature, the test above would produce a stress in excess of the specified pressure shall be reduced to a pressure at which the stress will not exceed the specified minimum yield strength at the test temperature.

The maximum test pressure at which the stress produced will not exceed the specified minimum yield strength may be calculated by the following equation:

$$Pm = \frac{2S . E . t}{D} \qquad \text{Eq. 2}$$

Where:

Pm = maximum test pressure (kg/ cm^2)
S = specified minimum yield strength at test temperature (kg/cm^2)
t = specified pipe wall thickness minus mill tolerance (cm)
D = outside diameter (cm)
E = quality factor (see ASME B31.3 table A-1 B)

For hydrostatic testing of piping designed for external pressure as follows:

- Lines in external pressure service shall be subjected to an internal test pressure of 1-1/2 times the external differential design pressure but not less than a gauge pressure of 1.055 kg/m2 (15 psi).
- Jacketed lines, the internal line shall be pressure tested on the basis of the internal or external design pressure, whichever is critical. This test shall be preformed prior to completion of the jacket.
- The jacket shall be pressure tested of the basis to the jacket design conditions.
- Where systems require hydrostatic testing through static equipment, the test pressure shall be selected so as not to exceed vessel test pressure.

Safety of Testing :

The safety related issues including but not limited to the following should be addressed in Job Hazard Analysis (JHA) to be made by contractor for performance of Pressure Testing activities:

- Appointment of contractor's Test Controller who is in attendance and responsible throughout the testing and inspects the welding during testing.
- Appointment of the subcontractor's test controller who will be responsible for ensuring safe testing in accordance to the specification.
- Display of safety warning signs to alert workers in the vicinity of the pressure testing with line, identification.
- Pressure test training and maintenance of a competency register as required by contractor Safety Plan.
- Pressure rating for the test manifold and the test equipment and the required inspection/testing.

Record of Testing :

Records shall be made of each piping system test. These records, when completed, shall be submitted as part of the test and inspection certificates which are required for pre-commissioning. Records shall include:

Date of test
- Identification of piping system and any vessels or equipment tested with the piping system.
- Test medium
- Test pressure and maintaining time
- Minimum ambient temperature

All test records and authorized contractor certifications shall be retained in the Test Package records for turnover to the Owner. Test data base shall be established to monitor status and progress.

Documentation of Testing :

All recorder charts shall be signed by the test engineer when placed on and taken off the recorder. All recorder charts taken during the hydrostatic pressure test shall also be signed by the Company.

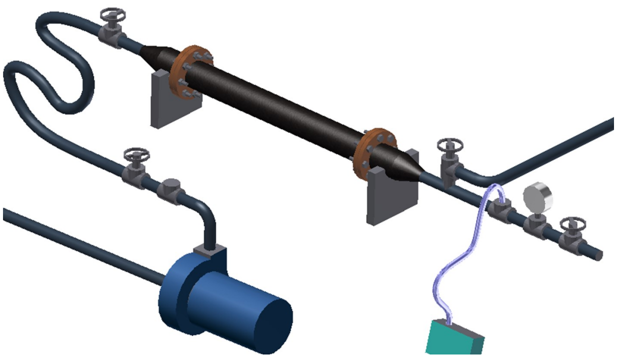


Figure 2 : Record of pressure.

Upon completion of a successful section test a "hydrostatic test certificate" shall be completed and signed by the Contractor and Company. This shall be supported by original hand-written test data of which photocopies shall be handed to the Company. A separate certificate shall be completed for each test, including pre-test and assembly testing.

For each complete pipeline, the Contractor shall compile a "final hydrostatic test report", with a general introduction including all relevant pipeline data, detailing each hydrostatic pressure test. The contents, as a minimum, shall include the following:

- Originals of all “hydrostatic test certificates”
- Originals of all recorder charts
- All pressure readings
- All temperature readings
- Volumes of water added or bled off
- Instrument test certification
- Air content plot and calculation
- Pressure and temperature plots against time
- Pressure/temperature correlation calculation
- Details of line-fill treatment packages
- Pig register of cleaning, gauging and filling
- Photographic record of the used gauging plates
- Leak locating (if carried out)
- Pig register of dewatering (if carried out)
- Details of line-fill water disposal (if carried out)
- Rectification records (if carried out)
- Any special features of the test
- Test procedure

Rectification Requirements of Testing :

If a leak is suspected, pressure reduced to less than 80 % of the test pressure before carrying out a visual examination. If it is not possible to locate the suspected leak by visual examination, use a method which enables the locating of leaks at test pressure without endangering the personnel carrying out the work.

To tie-in or to rectify any defects, it may be necessary to partially or completely dewater the test section. For partial dewatering of test sections containing treated water, the use of bi-directional pigs, remotely controlled inflatable isolation plugs and/or hyperbaric spheres should be considered to isolate the repair or tie-in location from the water-filled test section, instead of total dewatering.

Bi-directional pigs propelled by compressed air should be used for displacement of the line-fill water. Pigging shall be carried out against a back-pressure of hydrostatic head plus 1 bar so that air does not enter into the water-filled section. No attempt should be made to dewater the test section by letting the water run out under the effects of gravity.

The test section shall not be left in the partially or completely dewatered condition longer than one week without any further internal corrosion protection. Depending on the post-dewatering period and the line-fill water quality, it may be necessary to purge the test section with nitrogen or swab it with fresh water and/or inhibition slugs to avoid internal corrosion.

Reference :

“*Guidelines Hydrostatic Test Water Management*”, Canadian Association of Petroleum Producers, Canada, 1996.

“*Process Piping*”, ASME B31.3 Code, 2012.

Control Valves Fundamentals

Introduction

Hundreds or even thousands control loops are networked together in a process system plant to maintain the important process condition; such as pressure, fluid flow and level, temperatures, etc. During the process, each of these loops receives and internally creates disturbances that might affect process conditions

Hence sensors and transmitters are installed to send information about process condition changes to the controller, which can make any needed corrections actual to the desired set point by sending a signal to the final control element. Furthermore, a final control element is needed to provide the power and accuracy to control the flowing medium to the desired service conditions. The most common type of final control element in industrial process control system is control valve. The valve makes the change according to the signal from the controller, completing the loop.

Each type of control valve has a different flow characteristic, and its selection largely based on the type of the application process where it's installed. Some common cases come along with this control valve sizing; an oversized control valve will spend an extra cost and introduce some difficulties in controlling the low flow rates, while an undersized

valve might not be able to handle the maximum capacity of the process flow.

The control valve supports the other devices which work together resulting in an ideal process condition. Hence, it is crucial to make some considerations before deciding the correct control valve sizing and selection. The selected valve has to be reasonable in cost, require minimum maintenance, use less energy, and be compatible with the control loop. Malfunctions in a control valve might cause the process system not to work properly.

Nomenclature

C_f	**Critical flow factor for line-size valve**
C_{fr}	**Critical flow factor for valve between pipe reducers**
C_v	**Capacity coefficient for control valve in fully open position**
C_{vc}	**Calculated coefficient for control valve**
D/d	**Ratio between larger pipe dia. to smaller pipe dia.**
E	**Expansion factor, ρ_{60}/ρ**
k	**Ratio of specific heats**
M	**Molecular weight**
P	**Absolute pressure, psia**
P'	**Absolute pressure, psia**
P_c	**Critical pressure, psia**
ΔP	**Differential pressure, psi**
P_v	**Vapor pressure of liquid at flowing temperature. psia**
Q	**Volume flowrate, gpm**
R	**Correction factor for control valve between pipe reducers**
S	**Specific gravity of liquid, p/ Psow**
S_{60}	**Specific gravity of liquid at 60°F**
T	**Absolute temperature, °R**
v_s,	**Sonic velocity, ft/s**
W	**Weight flowrate, lb/h**
μ	**Viscosity, cp**
p	**Density of fluid at flowing condition, lb/ft3**
ρ_{60}	**Density of fluid at 60°F, lb/ft3**
ρ_{60w}	**Density of water at 60°F, 62.37 lb/ft3**

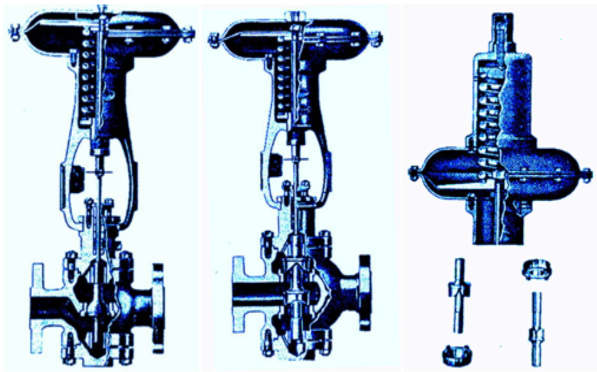
Subscripts

1	**Upstream condition**
2	**Downstream condition**

Major Types of Control Valves

One major group of control valves resembles the globe valve (Fig. 1). In place of a handwheel, an actuator moves the valve stem and plug, thereby opening and closing the valve. The usual actuator is an air-operated device whose housing contains a diaphragm that separates it into two compartments. The diaphragm (and attached valve stem) is balanced in its position by a spring on one side and air pressure on the other. In flow control, the air pressure changes in response to a signal resulting from the measurement of the differential pressure across an orifice or other flow-sensing element.

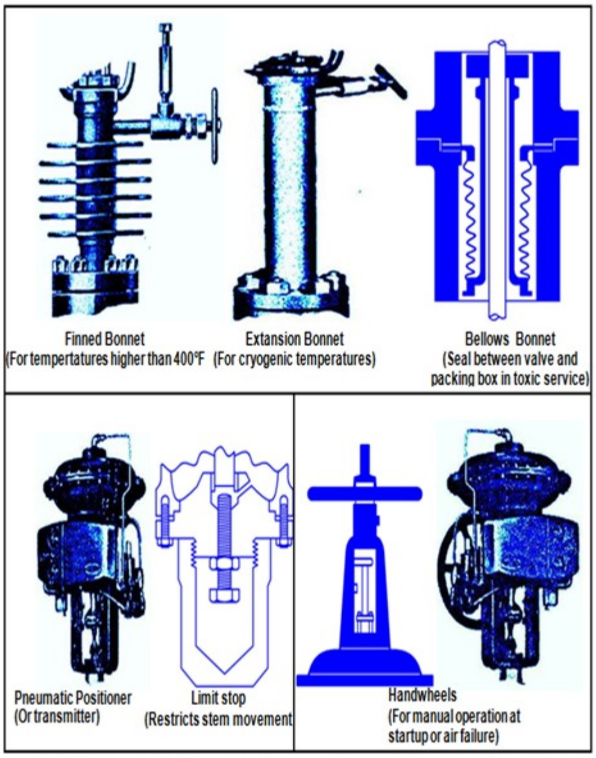
The single-ported control valve (Fig. 1) finds use where tight shutoff is required in addition to flow control. The double-ported control valve (Fig.1) has two seat rings with two plugs on a common stem. This is a higher capacity valve than the single-seated one of the same size. With hard seat rings and high temperatures, the double-seated valve cannot shut off tightly.



Single-Seat (Equal-percentage contoured plug, fails closed) | Double-Seat (Equal-percentage ported plug, fails closed) | Alternative Actuator And plug

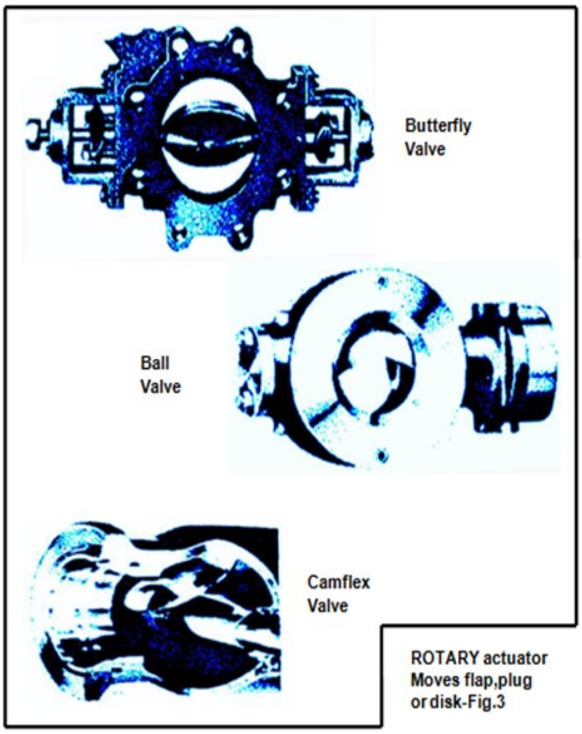
Figure 1

The valve accessories shown in Fig. 2, allow for various operating functions and conditions.



ACCESSORIES extend usefulness of control valves by providing for extreme and unusual conditions-Fig.2

In recent years, a second group of control valves has received wide acceptance. In these types, the actuator rotates a butterfly flap,plug or disk around its axis (Fig.3). Size for size, these valves usually have higher capacities and less flow resistance than the contoured- plug valves. Generally, control valves with rotating axes are suitablefor a wide range of flowcontrol applications.



ROTARY actuator Moves flap,plug ordisk-Fig.3

Characteristics of Valve Plugs

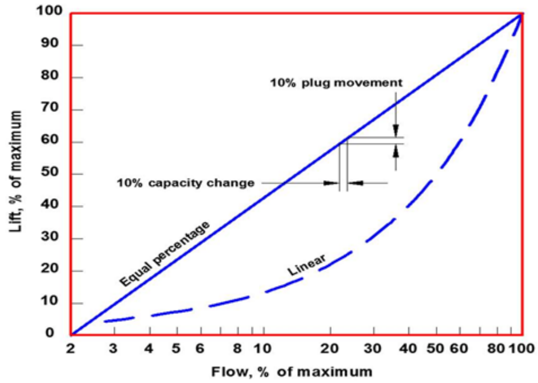
The valve plug can be disk type,solid contoured or ported. Flow-control characteristics depend on the shape or cavities of the plug. The three basic types of plug and their flow characteristics are:

Quick Opening-A single-disk (for high temperatures) or a double-disk (for low temperatures) plug is used for total shutoff or opening. A disk type plug has linear flow characteristics and short stem movement.

Linear Flow-A plug has linear flow characteristics when the flowrate through the valve is proportional to the lift.

Equal Percentage-A plug has equal-percentage characteristics if at any plug position,the same percentage of change in flow takes place for the same amount of plug movement. The percentage of change is related to the flowrate just before the plug is moved, as shown in Fig. 4.

Most plug characteristics are somewhere near or between those described. Manufacturers provide diagrams similar to Fig. 4 for each valve.



Flow characteristics of ported or contoured plugs –Fig.4

A plug having linear-flow characteristics is commonly specified for liquid-level control. The equal-percentage plug is used for pressure or flow control; or where only a small percentage of the overall pressure differential is available; or where pressure drop across the control valve varies greatly.

The modified parabolic-flow characteristic falls between the linear andequal- percent-age characteristics. This type of plug (usually V-port) finds use where the major part of the system pressure drop is available for control.

Actuators (also called operators or valve positioners) lift the valve stem and plug above its seat, or move the plug in the seat cylinder. Butterfly or ball-type control valves have the actuators side-mounted because the actuator stern rotates the valve axle. Plug characteristics can be influenced by the linkage between actuator stem and valve axle.

The valve housing and the operator's yoke are separate pieces. Hence, after a valve is installed, the operator can be rotated around the valve stem or valve axle, relative to the valve body. This enables a convenient position to be chosen for the actuator, in order to provide access to operating points on the valve. Hydraulic, mechanical and piston operators are also available.

Safety Requirements

Without air pressure in the pneumatic actuator, the valve can be in closed or open position.
These alternative positions are accomplished by reversing the seat ring and plug, or by reversing the location of the actuator spring from below to above the diaphragm (Fig. I).

One concern of the designer is to select valves that will fail-safe in the event of instrument-air failure. In principle, a control valve fails safe if temperature

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and pressure of the process system do not increase after the control valve becomes inactive.

For example, fuel-oil control valves to heater burners should fail closed. At the same time, feed to heater tubes (in most cases) should fail open to avoid overheating the furnace tubes. The feed control valve to fractionating columns usually fails closed. Steam supply to reboiler fails closed. Reflux drum vapor outlet and reflux pump discharge valves fail open. Control valves in minimum flow bypass lines at centrifugal pump discharge lines, compressor bypass lines, and reciprocating machine bypass lines fail open.

Reactors are protected under controlled conditions, and usually the feed control valve fails closed. Generally, a designer of flow systems should consult process, instrumentation and equipment engineers when deciding on fail-safe positions for control valves so as to assure orderly shutdown procedures.

Capacity Coefficients of Valves

Valve flow coefficient, C_v, depends on the internal dimensions of the valve and the smoothness of surfaces. Tests made by manufacturers (using water or air at predetermined pressure difference) establish C_v values. Manufacturers give the following definition :

$$C_v = Q(\sqrt{S} / \sqrt{\Delta P})$$

C_v is a capacity index indicating the flow of 60°F water in gpm, which will pass through the completely open valve under a pressure difference of 1 psi between the inlet and outlet flanges. Obviously, if $S = 1$ and $\Delta P = 1$ psi, then $C_v = Q$.

Capacity indexes for the butterfly valve are also given at two throttling positions of the flap, in addition to the fully open position.

Control-valve coefficients for single- and double-seated valves are given in Table I.

Calculated Flow Coefficient, C_{ve} - When sizing control valves, a flow coefficient is calculated with normal design flowrate in gpm from:

$$C_{vc} / C_v = 0.5 to 0.8$$

This is an optimum range for linear and percentage contoured plugs. Some valves have a wider optimum range. All valves will operate below and above these C_{vc}/C_v ratios, but the plug will be closer to the fully open or fully closed position. Under these conditions, we lose the important advantage of having wide flexibility in controllable flow-capacity range, and this may limit operability of the process.

High velocities across the valve orifice can wear out the plug and seat, especially if temperature is also high or when abrasive fluid is present.

Flow Coefficients for Control Valves-Table I

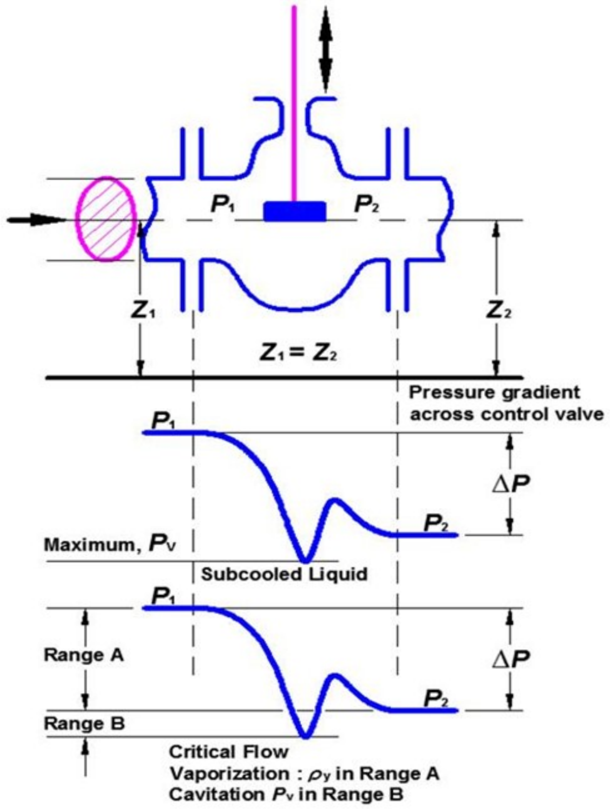
	Flow Coefficient, C_v	
Sizse, In	Single – Seat*	Double-Seat*
¾	---	8
1	9	12
1 ¼	14	18
1 ½	21	28
2	36	48
2 ½	54	72
3	75	110
4	124	195
6	270	450
8	480	750
10	750	1,160
12	1,080	1,620
14	1,470	2,000
16	1,920	2,560

These values have been obtained for Masoneilan 10,000-series (either equal-percentage or V-port) plug valves having full-capacity trim, but also apply to similar valves of other manufacturers [2].

Critical Flow Factor, C_f -The pressure gradient across a control valve is shown in Fig. 5. For liquids, the flow can be considered subcritical if the vapor pressure of the liquid will not get higher than the lowest pressure point across the control valve.(Vapor pressure is the pressure at which the liquid begins to vaporize at its flowing temperature. Tables of thermodynamic properties of liq uids give corresponding saturated-liquid pressures and temperatures.)

If the vapor pressure falls between the ranges of A and 8 (see Fig.5), vaporization or cavitation will occur in the control valve.

If the vapor pressure nears the downstream pressure, P_2, cavitation can be suspected. Cavitation can cause rapid wear of valve plug and seat as well as vibration and noise.If the vapor pressure falls between upstream and downstream pressures, P_1 and P_2, vaporization can occur. In this case, there will be two phase flow in the pipeline after the control valve. If the vapor pressure is higher than the inlet pressure, P_1, the control valve receives two-phase flow; and additional vaporization can be considered across the valve. For this condition, diameter of the downstream pipe will usually be larger than the upstream pipe.



Pressures during liquid flow in a control valve –Fig.5

The criteria for subcritical and critical flows in liquids are,respectively:

$$\Delta P < C_f^2(\Delta P_s) \quad (1)$$

$$\Delta P \geqq C_f^2(\Delta P_s) \quad (2)$$

Where : $$\Delta P_s = P_1 - (0.96 - 0.28\sqrt{P_1 / P_c})P_v \quad (3)$$

And P_c is the critical pressure, psia.

For simplicity: $\Delta P_s = P_1 - P_\upsilon$, provided that $P_\upsilon < 0.5P_1$.

The sizing formula for critical flow is :

$$C_{\upsilon c} = (Q / C_f)(\sqrt{S} / \sqrt{\Delta P_R}) \quad (4)$$

We will use a simplified version of Eq. (4) later in this article.

One example of subcritical flow is that occurring in a control valve located in the discharge line from a centrifugal pump. Critical flow can occur across a pressure reducing valve where the upstream liquid condition is close to the boiling point.

For gases, critical flow is assumed when gas velocity reaches the sonic velocity:

$$\upsilon_s = 68\sqrt{k(P' / \rho)}, ft / s \quad (5)$$

Sonic velocity should be avoided because it can cause noise and vibration.
The criteria for subcritical and critical flows in gases are, respectively:

		Single -Seat		Double-Seat	
Conditioon	**Factor**	**Equal-Percentage**	**V-Port**	**Equal-Percentage**	**V-Port**
Critical Flow Line size control valve	C_f	0.98' or 0.85‡	0.98	0.90	0.98
Critical Flow (Control valve between pipe reducers)	C_{fr}	0.85	0.94	0.86	0.94
Sub Critical flow,D/d =1,5 Sub critical flow, D/d = 2 (Control valve between pipe reducers)	R R	--- -0.96--- --- -0.94--			

$$\Delta P < 0.5C_f^2 P_1 \quad (6)$$

$$\Delta P \geqq 0.5C_f^2 P_1 \quad (7)$$

Critical flow can be avoided by reducing the pressure drop across the valve, by relocating the valve in the flow system, or by choosing a valve with a high C_f value.

The critical flow factor, C_f is a dimensionless number, which depends on the valve type [6].

C_{f}, is the ratio between the control-valve coefficient under critical conditions and the flow coefficient as published in rnanufacturers' literature.

Value Between Pipe Reducers-Flow capacity of a control valve placed between pipe reducers is slightly decreased. In subcritical flow, this is accounted for by a correction factor, *R*. In critical flow, the correction factor is C_{fr}," which replaces C_{f}, in the calculations. *R* and C_{fr} also depend on the ratio between pipe size and valve size. C_f,C_{fr} and R have values smaller than I. Numericaf values for the valves shown in Fig. I are listed in Table II.

Let us now summarize a number of formulas for sizing control valves for liquid and gas services under different flow conditions [I].

Liquid Service

Subcritical Flow-For a liquid flowing well below its saturation temperature in the turbulent zone, with viscosity close to that of water, and sizes of the pipe and control valve identical, the calculated control-valve coefficient is :

$$C_{vc} = Q\sqrt{S}/\sqrt{\Delta P} \quad (8)$$

Where the specific grafity, s, and flowrate Q gpm, are taken at the flowing temperature ; and $\Delta P = P_l - P_2$,

For minimum pressure drop at the fully open plug position, C_v replaces C_{vc}:

$$\Delta P_{(\min)} = (Q/C_v)^2 S, psi \quad (9)$$

If we are initerested in trhe pressure drop at a selected plug position between C_{vc}/C_v = 0.5 to 0.8, a convenient expression is :

$$\Delta P = \left[\frac{Q}{(C_{vc}/C_v)C_v}\right]^2 S, psi \quad (10)$$

Where C_v is taken from the manufacturee's catalog and C_{vc}/C_v is the selected plug position. (The methods of Eq. (9) and (10) can also be adapted to vapor flow.)

The calculated flow coefficient for laminar or viscous flow is :

$$C_{Vc} = 0.072\sqrt[3]{\mu Q/\Delta P)^2} \quad (11)$$

Critical Flow-If the valve and piping are the same size, the simplified calculated control-valve coefficient becomes:

$$C_{vc} = (Q/C_f)(\sqrt{S}/\sqrt{P_1 - P_v} \quad (12)$$

Correction Factor for Control – Valve flow Coefficient Table II

Gas, steam and Vapor service

The calculated control-valve coefficient for subcritical flow will be :

$$C_{vc} = \frac{W}{11.65\sqrt{\Delta P(P_1 + P_2)\rho_1}} \quad (13)$$

Where $\Delta P = P_1 - P_2$, provided that ΔP<0.5 $C_f{}^2 P_1$.

For critical flow when $\Delta P \geqq 0.5$ $C_f{}^2 P_1$:

$$C_{vc} = \frac{W}{10.13 C_f P_1 \sqrt{\rho_1}} \quad (14)$$

If the valve is located between pipe reducers, multiply the righthand side of Eq.(8), (11) and (13) by (1/R); and eq. (9) and (10) by (1/R^2). Replace C_f with C_{fr} in Eq. (12) and (14).

These correction can be neglected if the capacity of the selected control val;ve at normal flow gives a coefficient ratio, C_{vc}/C_v, well within 0.5 to 0.8. The operating position of the valve plug will perhaps not be identical to the calculated position, but this will not change valve or pipe size. Also, in sizing valves for critical flow, make sure that the plug will not operate close to its seat.

Two – Phase Flow

For well-mixed liquid and inert gas in turbulent flow with no additional vaporization, the following applies:

$$C_{vc} = \frac{W}{44.8\sqrt{\Delta P(\rho_1 + \rho_2}} \quad 15)$$

where ρ_l and ρ_2 are the upstream and downstream two phase densities, respectively.

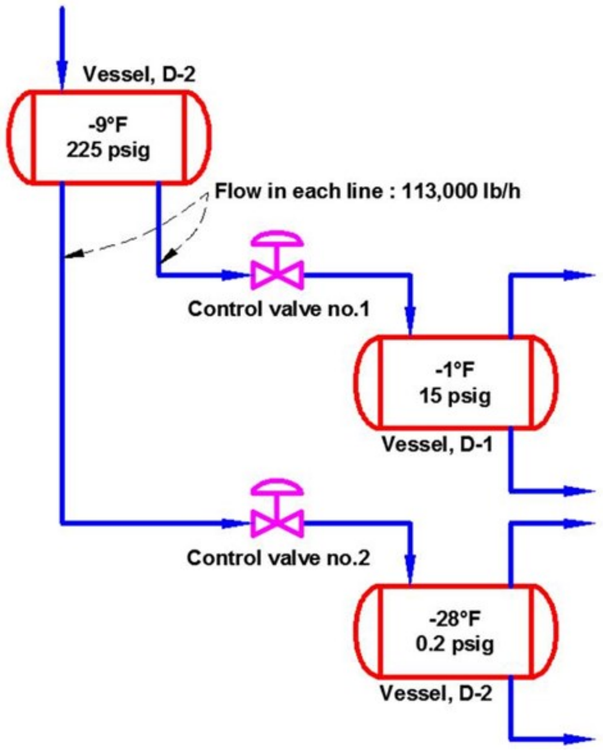
When saturated liquid enters the valve (i.e., $P_l = P_v$), or saturated liquid and its saturated vapor flow concurrently (i.e., $P_v > P_l$), additional vaporization of the liquid can be assumed inside the control valve. For this condition:

$$C_{vc} = \frac{W}{63.3\sqrt{\Delta P \rho_1}} \quad (16)$$

Where the maximum $\Delta P = 0.5 C_f^{\,2} P_1$. (For calculating the densities in two-phase flow, see Part I of this series, *Chem.Eng.,Dec.23, 1974,pp.60-61*)

Example Illustrated Computations

Let us size the control valves for handling a flow of 113,000 lb/h (348 gpm) of liquid ammonia in each of..



Flow relations for sizing control valves – Fig.8

Two lines for the system sketched in Fig.6. The three vessels are. The three vessels are located side by side (i.e., all are at the same elevation). Physical property data for liquid ammonia are: S_{60} = 0.615, E = 0.92, and M =17. Therefore, S =0.615 X 0.92 = 0.566. Pressure and temperature in each vessel, and corresponding thermodynamic properties, are

Vessel	D-1	D-2	D-3
Temperature, °F	-1	-9	-28
Pressure, psig	15	225	0.2
Heat content, btu/lb	41.8	33.2	12.8
Latent heat, btu/lb	569.6	575.6	589.3

Control Valve no.1 – As liquid ammonia flows from vessel D-2 to D-1, its heat content increases, and liquid in the pipeline after the control valve is subcooled. (Pipelines before and after the control valve can be sized for liquid flow).

Due to the large pressure difference between vessels D-2 and D-1, cavitation is possible in the control valve. Hence, we can consider the liquid to be in critical flow, and estimate the maximum vapor pressure in the valve from $P_v = 0.5P_l$, or :

$$P_v = 0.5(225 + 14.6) = 119.8\,psia$$

We will assume that a single set valve having a $C_f = 0.98$ (see Table II) will prove adequate by substituting the appropriate values into Eq.(12) to find:

$$C_{vc} = (348/0.98)(\sqrt{0.566}/\sqrt{239.6 - 119.8}) = 24.4$$

From table I, we establish that a 2-in single-seat V-port control valve having a flow coefficient of 36 may be adequate. We then check the ratio $C_{vc}/C_v = 24.4/36 = 0.68$, which falls well within the desired range of 0.5 to 0.8.

The 2-in lines (before and after the control valve) are relatively short, and when handling 348 gpm will have a small pressure loss. Consequently, pipe resistance will have practically no effect on size of the control valve.

Control Valve No.2 - As liquid ammonia flows from vessels D-2 to d-3, its heat content decreasses. Heat is released in the liquid, and as the liquid flows across the valve, vaporization will occur. The actual pressure drop, as determined from $\Delta P = 0.5 C_f^{\,2} P_1$, or:

$$\Delta P = 0.5(0.98)^2(239.7) = 115\,psia$$

Substituting the appropriate values into Eq. (16) yields:

$$C_{vc} = \frac{113.000}{63.3\sqrt{115(35.2)}} = 28.0$$

We may choose a 2½ -in single-seated control valve whose C_v is 54 (Table I). The coefficient ratio will be 28.0/54 = 0.52, which is acceptable. (Note: a low C_{vc}/C_v value was aimed for in this control valve

because C_{vc} was calculated with liquid density. It is not unreasonable to use a two-phase downstream density taken at the outlet of the control valve. Flashing and vapor density are then calculated with the critical downflow pressure. A much higher C_{vc} will result, and a high C_{vc}/C_v can be accepted.) Because of the subzero temperatures, an extension bonnet can be specified as an eccessory to the control valve (see Fig.2)
In flowing from vessel D-2 to D-3, the liquid releases 33.2 – 12.8 = 20.4 Btu/lb of liquid, or a total of 20.4 (113,000) = 2.3 X 10^6 Btu. The amount of vapor flashed with this heat is 2,300,000/582.5 = 3,950 lb/h. This leaves 113,000 – 3,950 = 109,050 lb/h liquid. These quantities can be used for calculating the two-phase flow resistance of downstream pipe.

Operating Conditions
Control valves are ussually the same size or one size smaller than the upstream pipe size, never larger. Control valves are much smaller than line size when high pressure differentials have to be absorted.

Control valves can accommodate a wide range of capacities and pressure differentials. Flowrates and process conditions are usually well determined for piping and components sizing. When sizing control valves, verify alternative capacities, periodically changing capacities and the related pressure differentials. Control over an extremely wide capacity range might require two control valves in parallel, one for the high flowrates, the other for the low ones. In borderline cases, or for a future increase in capacity, a larger valve body with reduced trim might be desirable.

In most instances, pressure differentials are part of the entire resistance of the piping system. Where an overall pressure differential is determined (for example, between two process vessels), one-third of the overall pressure drop can be attributed to the control valve , and two-thirds to friction loses in piping and equipment. At high pressure differentials, most of the pressure drop will be absorbed by the control valve. When pressure differentials must be minimized, the control valve should be line size, such as in steam feedlines to turbines.

Butterfly valves operate with very little pressure drop (decimals of 1 psi). They are usually suitable in comressor – discharge lines and cooling-water supply lines. However, under throttling conditions, the butterfly valve's coefficient decreases considerably. The coefficient is 50% at 72°position compared with the fully open (90°) position, and 33% at 60° setting.

At centrifugal pumps, pipe resistance in the discharge line (including that of any equipment in this stream) is usually known. An additional 25 to 50% of the discharge pipe resistance can be added for the control valve. With two control valves in series, there is double the amount of additional resistance can be added for the control valve. With two control valves in series, there is double the amount of additional resistance. For a control valve installed in long discharge line or in a system with high resistance and relatively small flow changes, the pressure drop across the valve can be 15 to 25% of total system resistance.

A control valve (except butterfly) can only regulate flow by aborsing and giving up pressure drop to the system. Economy in operation of control valves dictates lower pressure drops. However, the valve's capacity and range of control decrease rapidly with lower available pressure differentials.

Flow coefficient for Hand-Operated Throttling Valves -Table III

No.576P No.556 No.1040 No.1042 No.1046

Bronze Globe valves (Threaded)		
Size In	Flow Coefficient, C_v For Valves No. 546P-150 Psi No. 556P-200 Psi No. 576P-300 Psi	Flow Coefficient, C_v For Valves No. 556-200 Psi No. 576-300 Psi
¼	0.9	1.2
½	2	4.2
¾	5	8.6
1	10	14.5
1 ½	24	29.5
2	41	49

Steel Globe Valves (Flanged)		
Size In	Flow Coefficient, C_v For Valve No. 1040-150 Psi	Flow Coefficient, C_v For Valves No. 1042-300 Psi No. 1046-600 Psi
2	46	55
2 ½	72	90
3	105	130
4	200	235
6	400	400
8	720	720

Note : Flow coefficients have been obtained for valves manufactured

Changes in the specific gravity, or inaccurate density estimates, will have minor effect on valve capacity. These are small values – square-root function of the calculated flow coefficient.

When critical flow occurs in the liquid, the piping after the control valve (and bypass valve) should be carefully sized. Vaporization increases pipe resistance considerably. To stay within reasonable velocities when vaporization occurs across the control valve, the downstream piping and block valve will often be larger in size than the upstream pipe size.

In some cases of saturated liquid flow, vaporization in and after the control valve can be avoided by providing a static head of liquid upstream of the valve. This should be noted on the engineering flow diagram.

At high pressures, high temperatures, or large pressure differentials, the control valve should not operate close to its seat. High velocities can wear the plug and seat. This causes inaccurate flow control, and leakage when the valve shuts off.

Bypassing the Control Valve

A bypass is usually provided for control valves smaller than 2 in., in high-viscosity services, in handling liquids containing abrasive solids, in boiler feed water services, and in high (over 100 psi) pressure reducing steam service.

For consitency in piping design, the flow coefficient for the bypass valve should be about the same as that for the control valve. Table III lists the flow coefficients for some of one manufacturer's globe valves. Because of various seat –and-plug designs, valve coefficients are not the same for comparable globe valves made by different manufacturers.

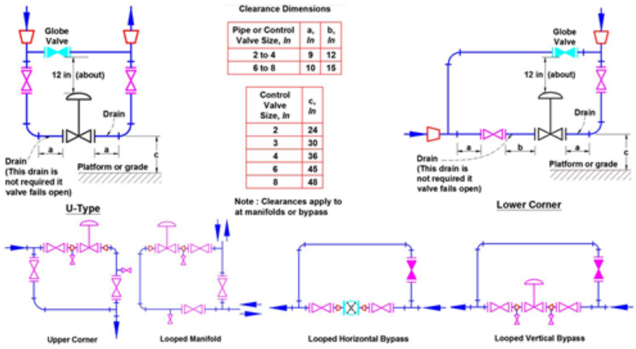
We find by comparing the data in Table III for globe valves with the flow coefficient for double-seated control valves in table I that the bypass valve and control valve can be the same size. For single-seated control valves, the bypass globe valve can be one size smaller than the control valve. We can size btpass globe valves or manually operated throttling valves in the same way as control valves provided that flow coefficient are available.

Piping the Control Valve

The best position for a control valve is with the stem vertically upward. A control valve will operate in angular, horizontal or vertically downward position. Neither piping designers nor operators accept these position. Large angle-control valves are an exception; a horizontal position for them can be most practical.

A single control valve without block valves and bypass is usually sufficient in clean-fluid service; or where paralel equipment containing control valves is installed with block valves located at pipe headers. Where dirty fluid or solid particles can be occasionally expected, a temporary or permanent strainer is installed, upstream of the control valve. Single control valves have handwheel operators.

Most piping specifications call for control valves to be located above grade or platform elevation, and at the edge of accessways, except for those valves that have to be located in self-draining pipelines. For example, a control valve placed in an overhead grafity-flow slurry line.



MANIFOLDS and bypasses for installing control valves into the process piping require proper clearances and drains-Fig.7

For inplace maintenance, clearance space is required below and above the valve for removing the seat, plug, actuator cover, spring and yoke. Estimated clearance requirements are shown in Fig.7. Dimensions of control valves are given by manufacturers [2,4]

If flow conditions permit, manifolds for the control valve that are smaller in size than the main piping will prove economical. Typical standard manifolds are shown in Fig.7[10]. The U-type is chosen when the inlet and outlet flows approach the control valve from an elevation higher than that of the valve. The corner type is used when flow is from a high point to a low point, or the reserve. The looped-bypass type serves horizontal flows near grade. A looped-corner bypass can bring a control valve over the operating platform. For economical support, control -valve manifolds should be located near structural columns.

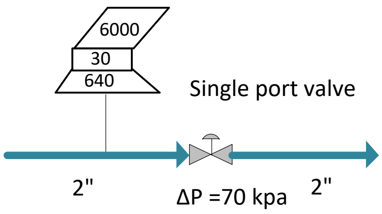
For pressure-relieving and draining a control-valve manifold, provide drain valves or plugs at low points. One drain point is required if the control valve fails open. Drains on each side of the conotrol valve are needed if it fails closed. In saturated-steam flow, one or two steam traps are advisable at the low points of a pocketed control-valve manifold.

The automatic control valve is part of an instrumentation system. Sensing points for flow, pressure, temeperature and level should be close to the control valve, as should the transmitter. Instrument wioring and tubing connect these elements. Air lines run from the transmitter to the diaphragm housing, and from the transmitter to the instrument air header.

Level controllers usually have gage glass companions. It is convenient for the plant operator to see the gage glasses from the control-valve manifold when operating the control valve handwheel or the bypass globe valve.

Example Case 1: Sizing a Control Valve in Liquid Hydrocarbon Application

A process utilizes a 2 inch pipe for the fluid flow of 6000 kg/h of liquid butane. Flow temperature is 30 °C, inlet pressure is 640 kPa and the expected ΔP is 70 kPa. Design a control valve for the stream.



Solution:

Step 1

Specify process variables:

T = 30 °C
P_1 = 640 kPa
W = 6000 kg/h
ΔP = 70 kPa
P_{vap} = 282.59 kPa
$P_{critical}$ = 550.7 kPa
μ = 0.0001307 Pa.s
S = 0.5701
D_1 = 2 inch = 50 mm
D_2 = 2 inch = 50 mm

Based on the pipe size given, we can assume a nominal valve size for the purpose of calculation. In this case, we shall try a nominal valve size of 1.5 inch.

d = 1.5 inch = 37.5 mm

Step 2

Determine the equation constants N:

Mass flow rate is specified and the unit of measure is in metric units. Thus, from Table 2,

N_2 = 0.00214
N_6 = 2.73

Step 3

Calculate miscellaneous valve/ fitting equations

Resistance Coefficient for inlet fitting

$$K_1 = 0.5\left(1-\frac{d^2}{D_1^2}\right)^2 = 0.5\left(1-\frac{37.5^2}{50^2}\right)^2 = 0.0957$$

Resistance Coefficient for outlet fitting

$$K_2 = \left(1-\frac{d^2}{D_2^2}\right)^2 = \left(1-\frac{37.5^2}{50^2}\right)^2 = 0.1914$$

In this example, the inlet and outlet piping has the same size. Observe that the Bernoulli Coefficient will have the same value and will cancel each other later when calculating the pipe geometry factor F_P.

- Upstream Bernoulli Coefficient

$$K_{B1} = 1-\left(\frac{d}{D_1}\right)^4 = 1-\left(\frac{37.5}{50}\right)^4 = 0.6836$$

- Downstream Bernoulli Coefficient

$$K_{B2} = 1-\left(\frac{d}{D_2}\right)^4 = 1-\left(\frac{37.5}{50}\right)^4 = 0.6836$$

- Inlet Head Loss Coefficient

$$\begin{aligned} K_i &= K_1 + K_{B1} \\ &= 0.0957 + 0.6836 \\ &= 0.7793 \end{aligned}$$

Step 4—Calculate Preliminary C_v:

$$\begin{aligned} C_v &= \frac{W}{N_6\sqrt{\Delta P \rho}} \\ &= \frac{6000}{2.73\sqrt{70x0.5701x1000}} \\ &= 11 \end{aligned}$$

Step 5—Calculate Combined Liquid Pressure Recovery and Piping Geometry Factor:

$$\begin{aligned} F_{LP} &= \left(\frac{1}{F_L^2} + \frac{K_i C_v^2}{N_2 d^4}\right)^{-1/2} \\ &= \left(\frac{1}{0.9^2} + \frac{0.7793\,\boldsymbol{x}\,11^2}{0.00214\,\boldsymbol{x}\,37.5^4}\right)^{-1/2} \\ &= 0.8920 \end{aligned}$$

Step 6—Calculate Valve/Fitting Geometry Factor:

$$\begin{aligned} F_P &= \left[\frac{(K_1 + K_2 + K_{B1} - K_{B2})C_v^2}{N_2 d^4} + 1\right]^{-1/2} \\ &= \left[\frac{(0.0957 + 0.1914 + 0.6836 - 0.6836)11^2}{0.00214\,\boldsymbol{x}\,37.5^4} + 1\right]^{-1/2} \\ &= 0.9959 \end{aligned}$$

Step 7—Determine the maximum allowable sizing pressure drop (ΔP_s):

Calculate Critical Pressure Ratio Factor (r_c)

$$\begin{aligned} r_c &= 0.96 - 0.28\sqrt{\frac{P_{vap}}{P_c}} \\ &= 0.96 - 0.28\sqrt{\frac{282.59}{550.7}} \\ &= 0.7594 \end{aligned}$$

- Calculate Vena-Contracta Pressure Drop (ΔP_{vc})

$$\begin{aligned} \Delta P_{vc} &= P_1 - r_c P_{vap} \\ &= 640 - 0.7594\,\boldsymbol{x}\,282.59 \\ &= 425.39\ \boldsymbol{kPa} \end{aligned}$$

- Calculate Critical "Choked" Pressure Drop (ΔP_{crit})

$$\begin{aligned} \Delta P_{crit} &= \left(\frac{F_{LP}}{F_P}\right)^2 \Delta P_{vc} \\ &= \left(\frac{0.8920}{0.9959}\right)^2 (425.39) \\ &= 341.25\ kPa \end{aligned}$$

The F_L term in the ΔP_{crit} calculation is replaced by F_{LP}/F_P as the valve is attached to fittings. The calculated critical pressure drop has a value of 341.25 kPa, which is well above the process specified value (70 kPa). This means that the liquid butane will not have any flashing or cavitation when it flows through the control valve. Thus,

$$\Delta P_s = 70\ kPa$$

Step 8—Compute Reynolds Number (N_{Re}):

$$\begin{aligned} N_{Re} &= \frac{34.6 F_d W}{\mu_{cp}\sqrt{F_L C_v}}\left[\frac{F_L^2 C_v^2}{890 d^4} + 1\right]^{1/4} \\ &= \frac{34.6x1.0x6000}{0.0001307x\sqrt{0.9x11}}\left[\frac{0.9^2 x11^2}{890x37.5^4} + 1\right]^{1/4} \\ &= 504{,}817{,}231 \end{aligned}$$

The F_d used in calculation above is 1.0 as we are using a single ported valve. Computed Reynolds number is well above 40,000 hence, F_R = 1.0, this formula is for British unit.

Step 9—Compute Final C_v:

$$\begin{aligned} C_v &= \frac{W}{N_6 F_P F_R\sqrt{\Delta P_s \rho}} \\ &= \frac{6000}{2.73x0.9959x1.0x\sqrt{70x0.5701x1000}} \\ &= 11.05 \end{aligned}$$

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