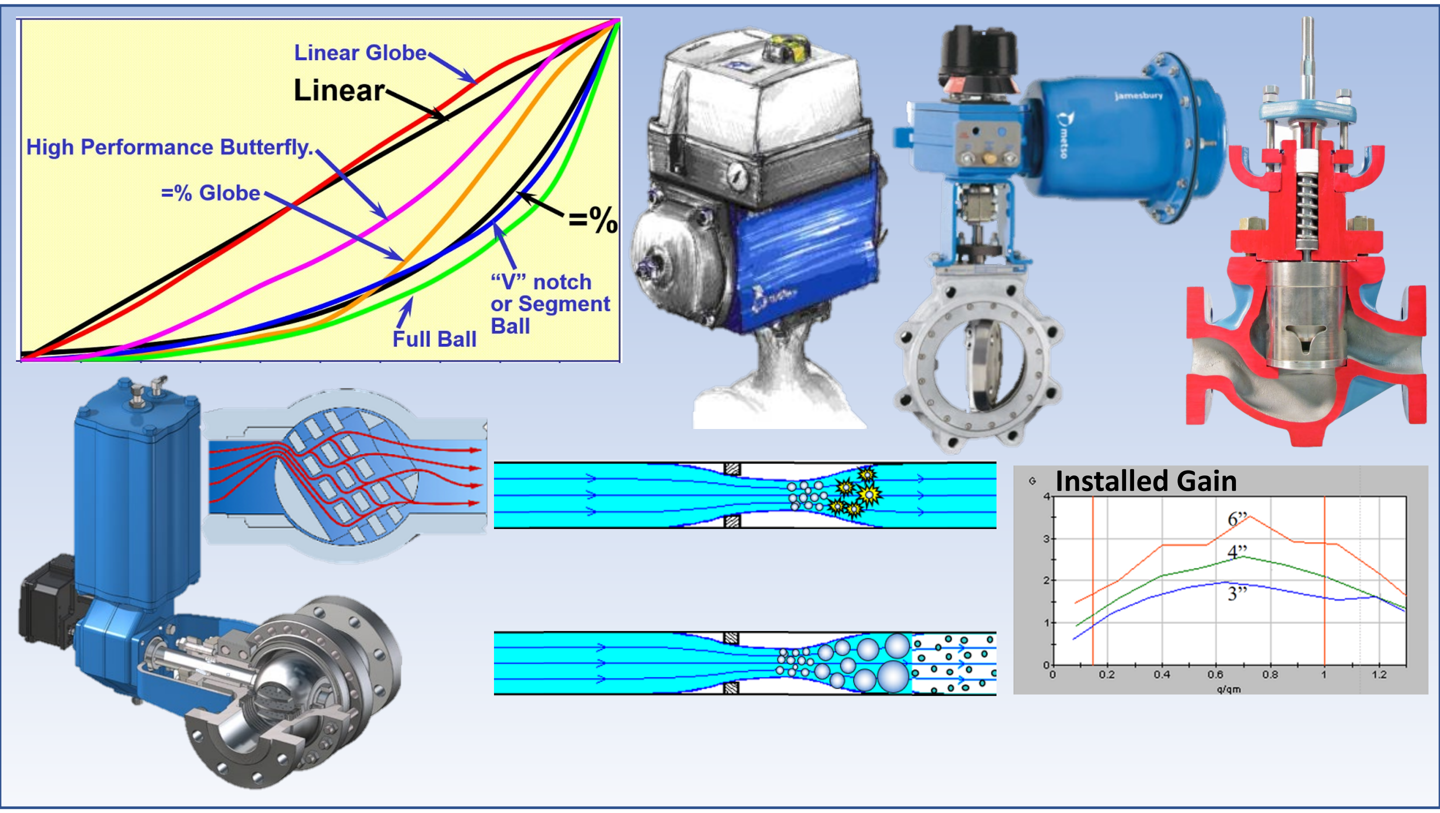
Professional Development Workshop

Final Control Elements



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Final Control Elements

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Introduction to Control Valves

Topics

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  - **Seat leakage specification**
- **Major control valve types**
- **Selection criteria**
  - **Operating temperature and pressure**
  - **Cost**
  - **Weight**
  - **Flow characteristic**
  - **Cavitation potential**
  - **Stem sealing**

These are some of the topics we will cover.

Flow Coefficient, Cv

$$Cv = Q\sqrt{\frac{G}{\Delta P}}$$

C_V: The flow in Gallons Per Minute of 60 degree water when there is a pressure differential of 1 PSI across the valve

C_V: A number that tells you relatively how much "STUFF" will go through a control valve.

For example, if a valve with a **C_V** of 100 is flowing 250 gpm, under the same process conditions a valve with a

CV of	will flow
200	500 gpm
300	750 gpm

The bigger the C_V, the bigger the flow

Typically, doubling the size of the valve will increase the C_V about three to four times..

The definition of Cv "the flow in gallons per minute of 60 degree water when there is a pressure differential of 1 PSI across the valve" although correct is not very useful, since the pressure drop is rarely 1 psi, but rather something else.

Notice that the pressure drop (delta p) is under the square root sign, so in order to double the flow, the pressure drop has to increase by four times.

The important thing to remember about Cv is that it is a number that is assigned to a control valve (determined by testing the vale) that gives you a relative idea as to how much flow that valve can handle. For the same process conditions, a valve with twice the Cv can handle twice the flow, and so on.

Seat Leakage Specification

ANSI/FCI 70-2-2006

Class I	By agreement between user and supplier
Class II	0.5% of rated valve capacity
Class III	0.1% of rated valve capacity
Class IV	0.01% of rated valve capacity
Class V	$5X10^{-4}$ ml per minute water/inch seat dia./psi of pressure differential
Class VI	Small number of bubbles per minute depending on seat diameter..

Most users and manufacturers of control valves specify allowable seat leakage using the six classes of seat leakage defined in the "Control Valve Seat Leakage" standard sponsored by the Fluid Controls Institute, FCI 70-2.

Class 1 refers to valves that are of designs that would be applicable to Classes 2, 3 or 4, but by agreement between user and supplier, no test is required. Class 1 is rarely specified.

Classes 2, 3 and 4 specify seat leakage as a percentage of the valve's fully open Cv.

Class 2 is the level of leakage equivalent to a Cv of a half percent of the valve's fully open Cv, and we will see in a moment that Class 2 can represent a fairly large amount of leakage.

Class 3 at a tenth of a percent of the valve's fully open Cv is 5 times tighter than Class 2, but can still represent quite a bit of leakage.

Class 4, at one hundredth of a percent of the valve's fully open Cv is a significant improvement. Most users consider Class 4 to be a good seat tightness for metal seated control valves.

Class 5 changes the rules from a percentage of the valve's rated fully open Cv, to a very small amount of leakage based on the valve's seat diameter, and the pressure differential that the valve is shutting off against. We will see in a moment that Class 5 only allows an extremely small amount of seat leakage.

Classes 2 through 5 are usually done with water as the test medium, but the standard also allows air.

Class 6 is an air test and only allows a very few bubbles of air coming through a layer of water. Class 6 represents a valve with extremely tight shutoff. Although some manufacturers claim that some of their metal seated valves can achieve Class 6 shutoff, it is most often only specified for soft seated valves.

Seat Leakage Comparison

ANSI/FCI 70-2

4 inch globe valve. Water, 60 psi pressure drop.
(Wide open flow = 1,500 gpm)

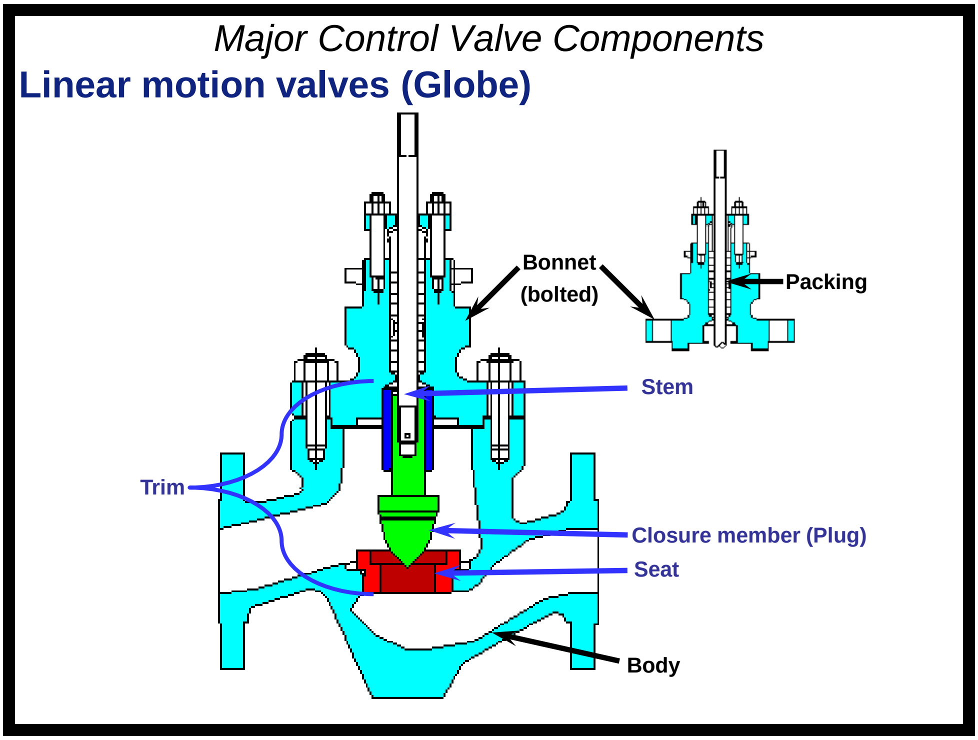
Class II	7.5 gpm
Class III	1.5 gpm
Class IV	0.15 gpm
Class V	3×10^{-5} gpm (0.12 ml/min) (50 ml in ≈ 7 hrs.)

- Class VI is specified only with an air test
- Comparison of air tests with Class V and Class VI:
  - Class VI about 1/10th the leakage of Class V…

Here is a comparison of the leakage through a 4 inch globe valve with a pressure differential of 60 psi for classes 2 through 5.

Note that Class 5 is very tight! It would take this valve's class 5 leakage almost seven hours to fill one of those little 50 ml vodka bottles you get on the airplane!

Since Class 6 leakage is specified for air only, we can't compare it with the water leakage rates shown. When comparing air leakage rates between Class V and Class VI valves, the leakage through a Class 6 valve will be about one tenth the leakage through a similar Class 5 valve.



Here we see the major components of a globe control valve. Globe control valves are sometimes referred to as linear motion control valves.

The body of a valve is defined as the part of the valve which is the main pressure boundary and that provides the pipe connecting ends.

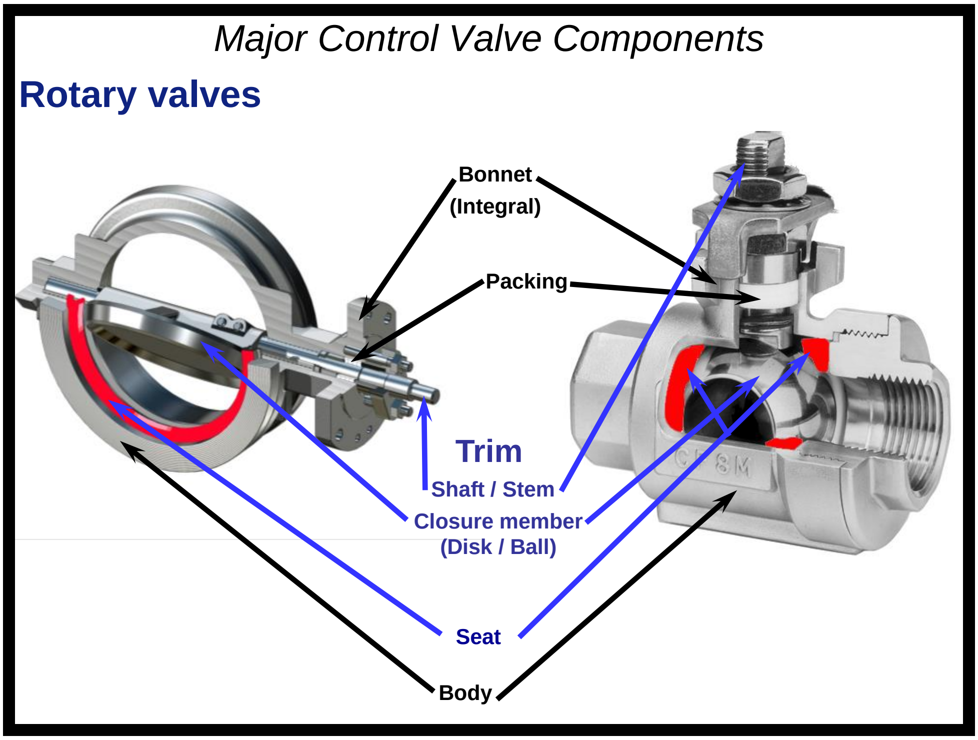
The Bonnet is defined as the portion of the pressure retaining boundary that contains the stem seal packing. In the case of globe valves, the bonnet is bolted to the valve body and provides the opening to the body for the assembly of internal parts. To differentiate this type of bonnet from the integrally cast bonnets for most rotary valves, they are often designated as “bolted bonnets.”

A valve’s trim consists of the internal parts that are in flowing contact with the controlled fluid.

The Closure member is the moveable part that is positioned in the flow path to modify the rate of flow through the valve. In the case of globe valves it is called the Plug.

The seat is fixed in place in the valve body and along with the closure member creates the orifice that controls the rate of flow through the valve. It also provides the surface for the plug to contact for the purpose of shutting off the flow.

The stem connects the actuator to the closure member.



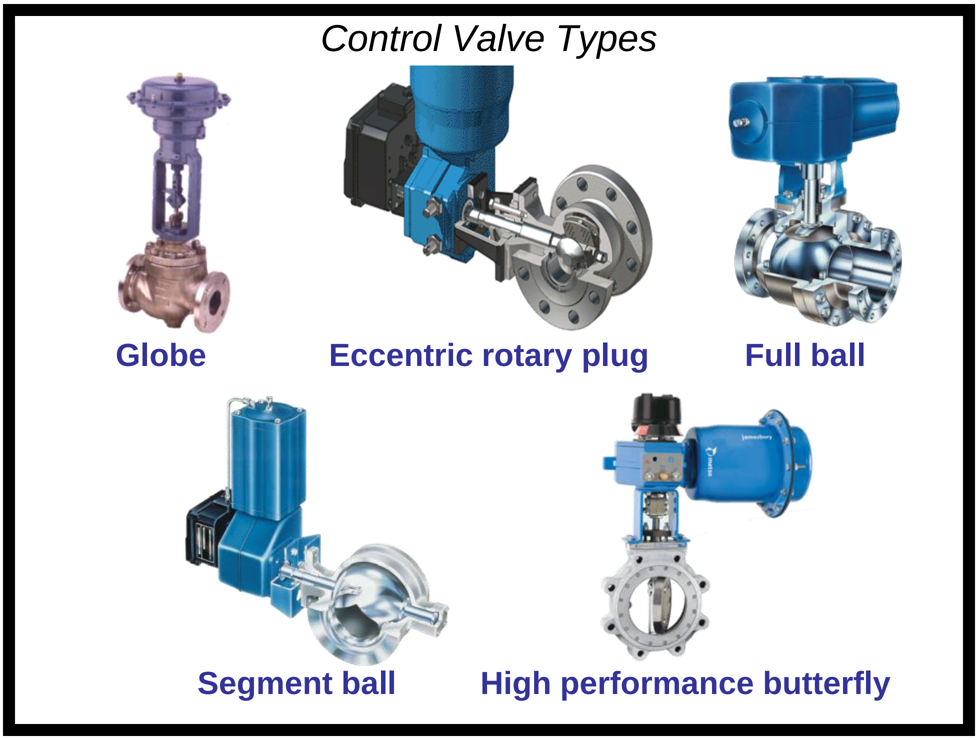
The definitions of the major components of the rotary control valves, which are also known as quarter turn control valves, are the same as those for the globe valves.

The bonnets of nearly all rotary valves are integrally cast as part of the body casting, rather than being bolted to the body like the bonnets of the globe valves. To differentiate this type of bonnet from the bolted bonnets of the globe valves, they are often designated as “integral bonnets.”

The component that connects the actuator to the closure member is called the stem by some manufacturers, and the shaft by others.

The closure member in a butterfly valve is called the disk, and the ball in in a ball valve.

In the pictures of the butterfly valve and the ball valve, we have colored the seats red to make it easier to see where they are. The two valves pictured here are Jamesbury valves and the seats are normally made of PTFE or Jamesbury’s proprietary Xtreme® both of which are actually white.



Next we will give a brief overview of the more popular styles of control valves.

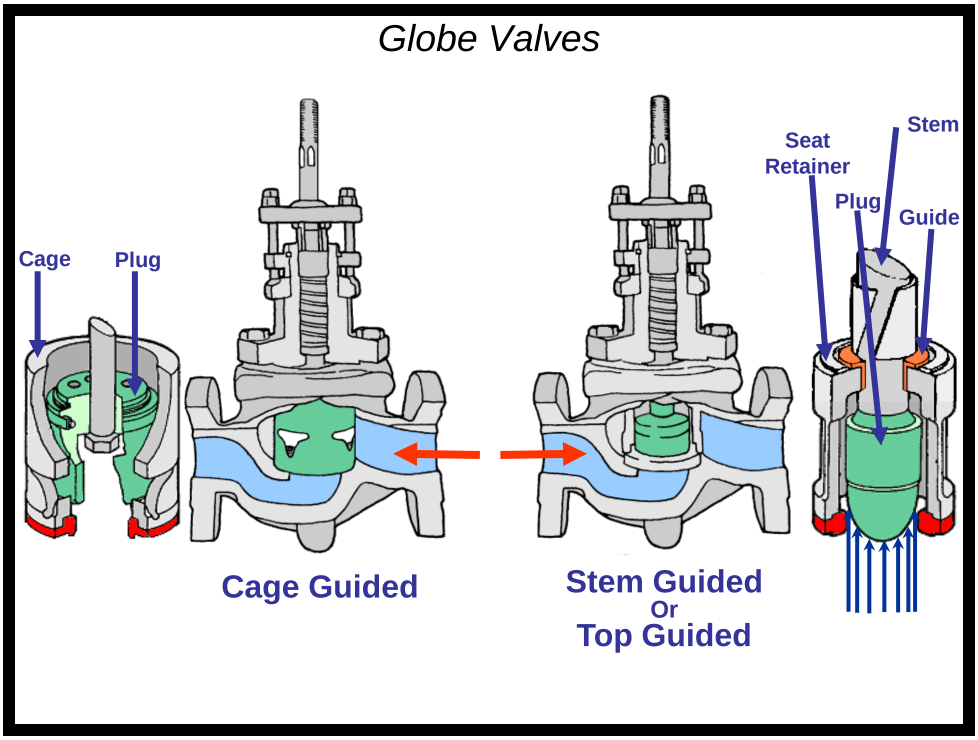
We will start with the most versatile (and most expensive), the globe valves. From there we will work our way down through the other types shown here, in order of decreasing versatility and cost until we get to the high performance butterfly valves which are the least versatile and the least expensive.

Globe Valves

The "Globe" valve name comes from the spherical body of some older designs of manual globe valves

Globe valves get their name from the shape of the body of some of the older designs, like the one pictured above. If you ignore the flanges and the bonnet, the body has a spherical, or globe shape.

Modern globe valves have body shapes that lend themselves to high speed automated machining and don't have much of a globe shape.



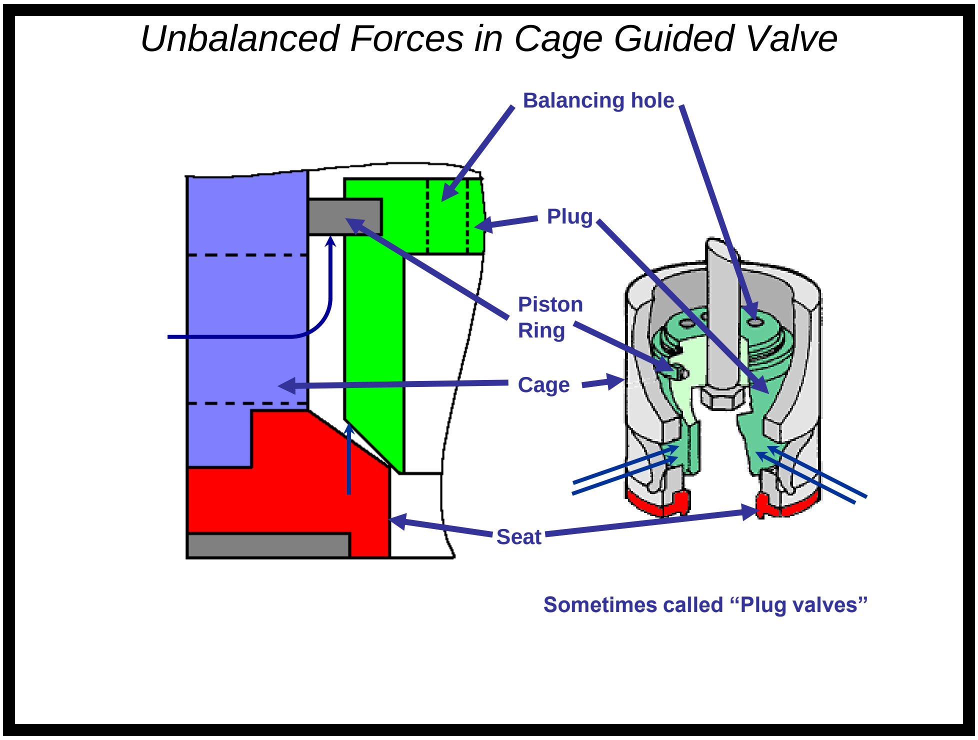
The globe valve is the most mature of all the control valves and is therefore available with the greatest number of options. Most globe valves can be fitted with full area trim (the greatest Cv that size valve is capable of) and also one and sometimes two or more sizes reduced trim. Modern globe valves come in two basic configurations. Stem guided (also referred to as top guided) and cage guided. The two designs get their names from the way the plug is held in place and made to move up and down in a straight line. With the cage guided valve, the plug is held in place by the cage, sort of like the way a piston in your car engine rides up and down in the cylinder. With the stem guided valve, the plug is held in place and guided by a guide busing in the bonnet.

The main strength of the cage guided design is that it is pressure balanced. The process pressure only pushes upward on a small portion of the plug, so only relatively small actuators are required. The cage guided valve's weakness is that it is not good for dirty service. Dirt particles are likely to lodge in the close tolerances between the plug and cage and cause the valve to stick.

The strength of the stem guided design is that there are no tight tolerances in the flow path. Dirt particles will sweep past the plug and the seat without getting caught anywhere that they would cause sticking or binding, making the design good for dirty service. The stem guided valve also has fewer parts, making it easier and less expensive to rebuild. The weakness of the stem guided valve is that the process pressure pushes upward on the entire surface area of the bottom of the plug. In large valves and at high pressures it may not be possible to find a large enough actuator. Notice that with the stem guided valve the flow is coming from below the plug.

The way that the cage guided valve controls the flow is that as the plug moves up and down inside the cage more or less of the triangular windows in the side of the cage are open to allow flow to go through.

The way the stem guided valve controls flow is that as the contoured plug moves up and down the open area between the plug and the seat becomes greater or less.



This illustrates the main cause of unbalanced forces in a cage guided globe valve. On the right is a drawing of a cage guided valve’s plug, cage, seat and piston ring. On the left is a cross sectional drawing of the left hand portion of the plug, cage, seat and piston ring. The tolerance between the cage ID and the plug OD has been exaggerated in the drawing.

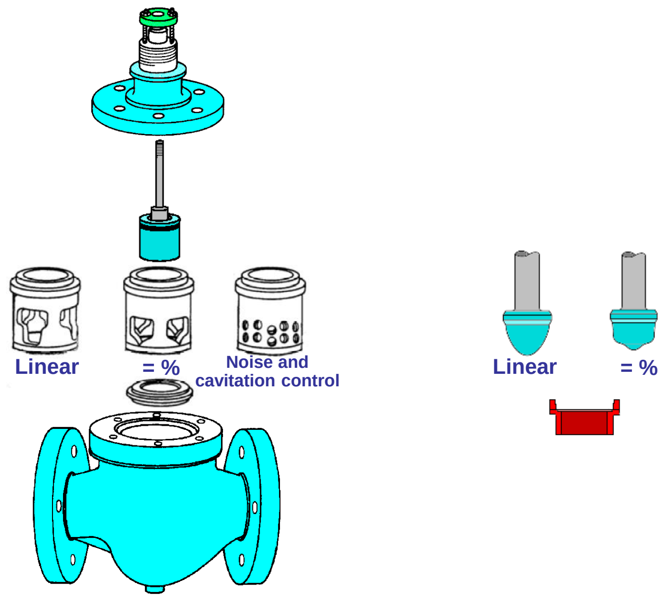
When the valve is closed, the process pressure comes through the triangular window on the left side of the drawing and pushes upward on the piston ring and on the portion of the plug that is not contacting the seat due to different angles on the plug and the seat. The process pressure force acting on this small area is the only force trying to open the valve. There cannot be any difference in pressure between the top of the plug and the bottom of the hollow portion of the plug because of the balancing holes that are drilled through the top of the plug.

Rotary "Plug" Valve

Some people refer to the stem or post guided valve as a "plug" valve because of the controlling member which is called a plug and looks like a plug.

This is OK, but keep in mind that there is a rotary valve called a plug valve as shown in the figure, so you need to be sure that others understand what you are talking about.

Globe Valve Trim Versatility

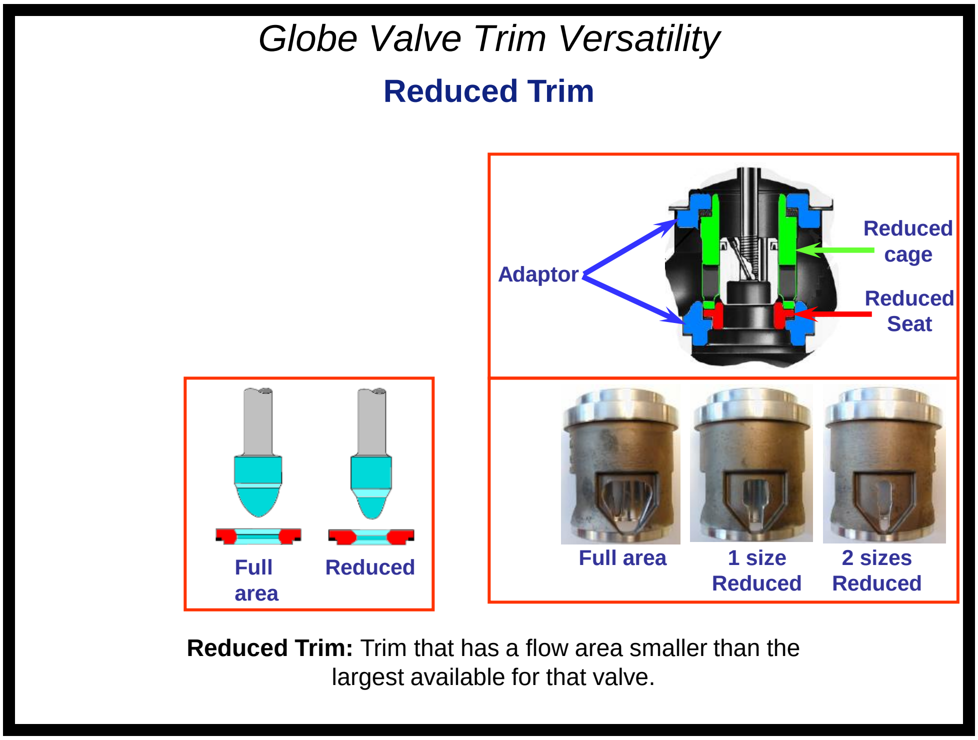


Trim: The internal components that control the flow.

This illustrates the versatility of globe valves. The trim size, material and characteristic can be easily changed. There are a number of options for noise and cavitation reduction.

With the valves of some manufacturers, you can even change a cage guided valve to a stem guided valve and vice versa.

Although many plants will not allow in-line repair, the globe valve can be repaired and the trim changed without removing it from the line since the trim is accessed from the top after removing the bonnet.



Most globe valves offer the option of reduced trim, that is trim with a Cv rating of a smaller sized valve.

“One size reduced” means that the valve with the reduced trim will have a Cv of the next size smaller valve.

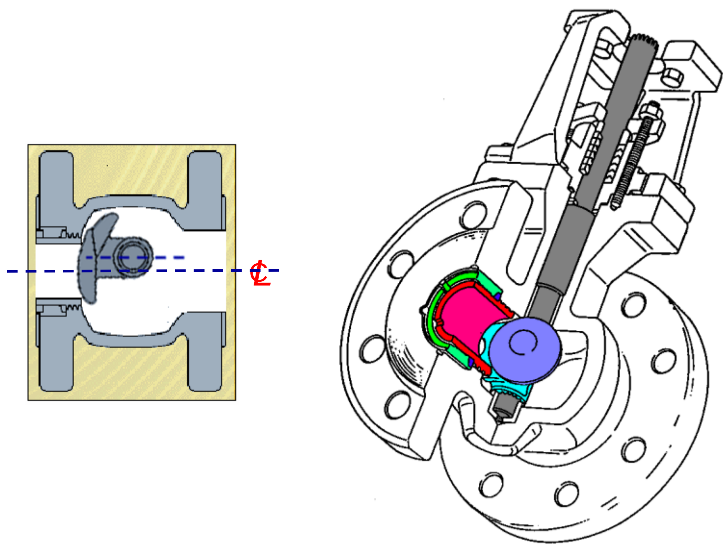
It is typical for stem guided valves to be available with full area, one size reduced and two sizes reduced trim. Stem guided valves in two or three inch (depending on the manufacturer) and smaller are available with many reductions to be able to handle a wide variety of low flows.

For cage valves there are two ways of making reduced trim..

The upper option uses the cage from the next smaller size valve with adaptor rings above and below. It is only practical to use trim that is one size reduced with this method. In order to change trim sizes you have to replace the cage, the seat and the plug. The actuator may have the wrong stroke length and also have to be replaced.

The lower option uses the same size cage, but with the windows in the reduced trim cages being smaller. With this method it is practical to make both one and two sizes reduced trim. In order to change trim sizes you only need to replace the cage. The seat and plug size do not change.

Eccentric Rotary Plug Valve

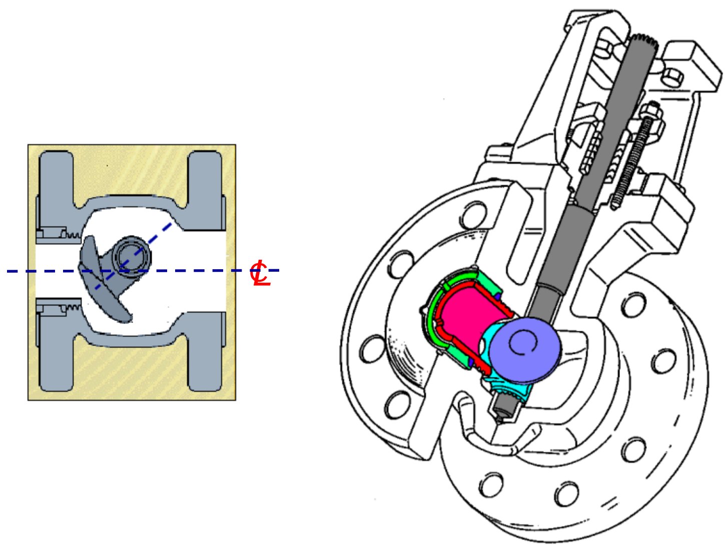


The eccentric rotary valve combines much of the ruggedness of globe valves with the compactness and long stem seal life of rotary valves.

The name “Eccentric Plug” comes from the fact that the valve shaft has a double offset. The shaft is offset behind the seating surface and it is also offset from the centerline of the valve.

Once the valve shaft and plug start to rotate, the plug cams away from the seat so that once the valve has started to open there is no rubbing between the plug and seat. (See the illustration on the next page.)

Eccentric Rotary Plug Valve



Here the plug has cammed away from the seat.

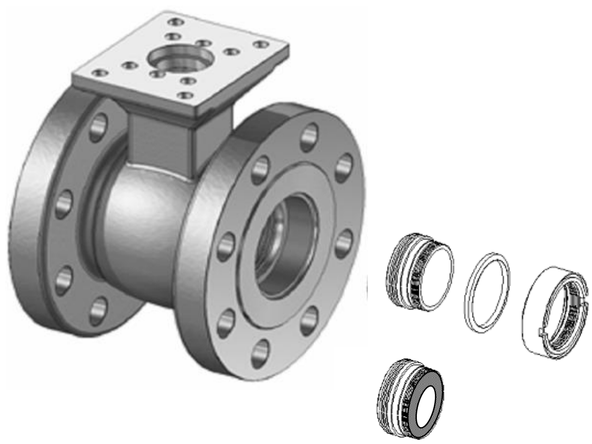
Eccentric Rotary Plug Valve

Noise and Cavitation Control Trim

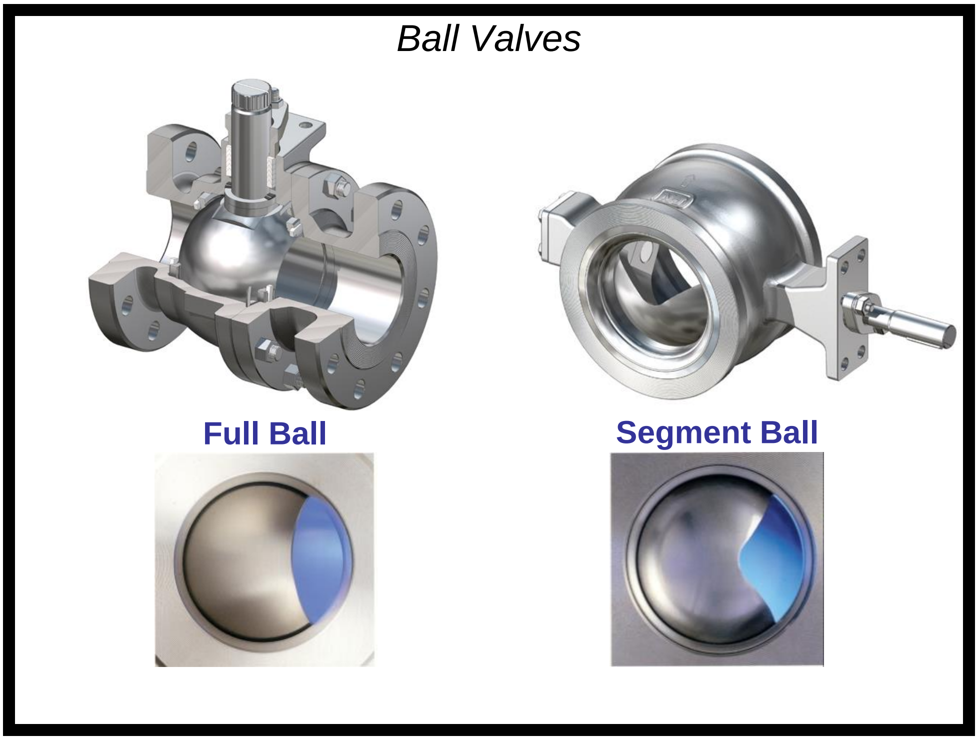
Although the eccentric rotary plug valve does not have as many options as globe valves, they are available with one design of noise and cavitation reduction trim.

Eccentric Rotary Plug Valve

Reduced Trim (one reduction only)



Typically, only full area trim and one sized reduced trim are available. In the Neles Finetrol valve, the reduced trim consists of a seat ring that has a smaller ID than that of the full area trim. The same plug is used with either seat ring.

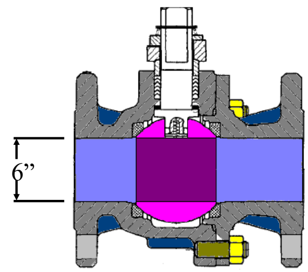


Ball valves come in two basic configurations. (1) full ball (round ball) and (2) segment ball.

Full Ball

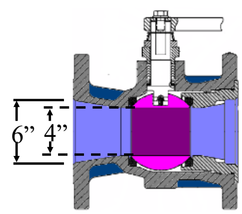
Ball Valve Port Size

- Full port
  - Port in ball is same size a valve size
  - No restriction
    - Better for erosive media
    - Less likely to plug with thick media
    - No press. drop in on-off applications
  - More expensive than standard port (About 40% more).



Full Port

- Standard (or reduced) port
  - Port in ball is one size smaller than valve size
  - Minimal press. loss in on-off applications
  - satisfactory for most control and on-off applications
  - Less expensive than full port valves
  - Specified more often than full port valves…



Standard Port
(or Reduced Port)

Full ball valves come in both full port and standard (or reduced) port. With full port ball valves the hole in the ball is the same size as the nominal size of the valve. For example, a 6 inch full port ball valve would have a ball with a 6 inch hole in it. With standard port ball valves, the hole in the ball is usually the same as that of the next size smaller full port valve. For example a 6 inch standard port ball valve would have a ball with a 4 inch hole in it.

Because there is no restriction in the full port valve, when used in on-off service, velocity is kept to a minimum inside the valve and the ball, which can be a benefit with erosive media. Also, in on- off service, the full port valve reduces the potential for a thick media (high consistency pulp is one example), that can bridge across a restriction to plug the valve. Because the ball in a full port valve is larger, and the body must be larger to contain it, full port valves are more expensive than standard port valves costing around 40% more.

In on-off applications, when fully open, full port valves don’t add any pressure loss to the system. Standard port valves do add some pressure loss to the system, but in most cases it is minimal.

Standard port valves turn out to work fine in most on-off and control applications, are less expensive than full port valves and are specified more often that full port valves.

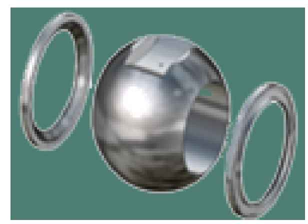
Full Ball

Ball Valve Seat Materials

- Soft seats are the most common (
  - PTFE, glass filled PTFE, Jamesbury *Xtreme*
  - Drop tight shut-off
  - Temperature < 400 to 450° F
    - To 500° F with Jamesbury *Xtreme*

Soft Seats

- Metal seats for severe service
  - High temperature
  - Abrasive media
  - Significantly more expensive than soft seated valves. (About twice as much)
  - Very good shutoff, but not drop tight…



Metal Seats

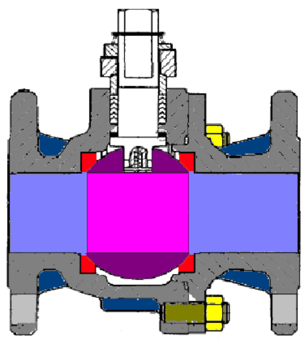
Ball valves are available with both soft seats and metal seats. For most applications where temperatures are moderate and the media is not very abrasive, soft seats are the best choice. The most common soft seat materials are pure PTFE (often called virgin PTFE) and glass fiber filled PTFE. They give drop tight shut-off and are significantly less expensive than metal seated valves. Temperatures are limited to about 400 degrees F at low pressure for pure PTFE and about 450 degrees F at low pressure for glass filled PTFE. Jamesbury has a proprietary seat material named XTREME® that has similar characteristics to PTFE but is rated to 500 degrees F at pressures up to 400 psi.

For higher temperatures and very abrasive media, metal seats are required. The metal seated valves are significantly more expensive than soft seated valves for a number of reasons. Metal seats are more expensive to machine than PTFE seats. After machining, the metal seating surfaces have to be hard faced through a welding overlay process. The balls are more expensive to make because perfect roundness is more critical, and they must be hard chrome plated. The friction between a metal ball and metal seats is much higher than the friction between a ball and PTFE seats, so the valve stem needs to be made larger and from more expensive materials than the stems needed for the soft seated vales that have lower torque requirements. Good quality metal seated ball valves give very good, but not drop tight, shutoff.

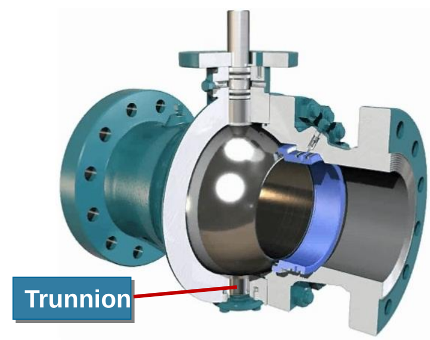
Full Ball

Ball Valve Styles

Seat supported (floating) ball
Trunnion-mounted ball



Seat Supported Ball



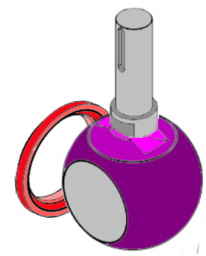
Trunnion Mounted Ball

There are two styles of ball valve designs: the seat supported, also called the floating ball design and the trunnion mounted ball design.

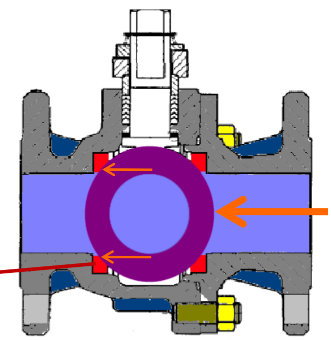
Full Ball

Seat Supported Ball Valves

- Ball held in place by the seats
- When closed, line pressure pushes the ball into the down stream seat
  - Slotted ball and blade shaft allow the ball to float downstream creating a tight seal
  - High shut off pressures and large balls can result in high ball-to-seat friction and high torque
  - Relatively smaller sizes, < 8” to 10
- Down stream sealing only
- Bi-directional shut-off
- Simpler design than trunion mounted ball valve
  - Less expensive…



Down stream Seal



The ball in the seat supported ball valve is held in place by the seats. The illustrations on this slide give a clear view of how the seat supported valve works. The top of the ball has a slot in it which is engaged by the stem which has flat sides. When the valve is closed and the line pressure pushes it in the down stream direction the slot allows the ball to move (or float) a few thousandths of an inch and press against the down stream seat resulting in a tight seal.

In larger valves at high pressures, the process pressure pushing the ball into the seat can result in high ball to seat friction and excessively high torque required to operate the valve. For this reason, most manufacturers limit the size of their seat supported valves to less than 8 to 10 inches.

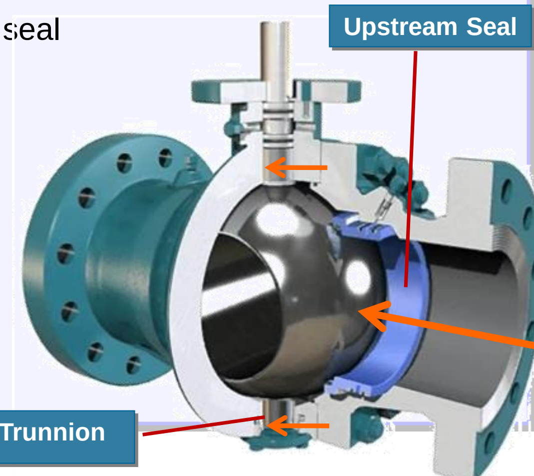
Seat supported ball valve seal on the downstream seat only. They are still bi-directional valves. If the pressure in the figure were coming from the left, then the seat on the right hand side would become the down stream seat.

Because there are fewer parts, the seat supported ball valves are less expensive than the trunion mounted ones.

Full Ball

Trunnion-mounted Ball Valves

- Ball is fixed in place, supported by upper stem and lower trunnion
- Lower torque than seat supported ball valve
  - The force exerted on the ball by line pressure is carried by the bearings
- Upstream seat moves toward ball to seal
  - Upstream sealing only
- Bi-directional shut off with two seats, uni-directional with one seat
- Larger sizes > 6” to 10”
- More expensive than seat supported ball valves. (About 20% more)…



Unlike the seat supported ball valve that allows the ball to move into the downstream seat with increasing shut off pressure acting on the surface area of the upstream ball face, the ball in the trunnion-mounted ball valve is fixed in place. In addition to the upper stem, there is a second shaft at the bottom of the ball, called a trunnion. Both the upper stem and trunnion ride in bearings which keep the ball firmly centered in the body between the seats. The bearings carry all of the force created by the shut off pressure acting on the ball, greatly reducing the friction between the ball and the seats. This keeps the required opening and closing torques within reasonable limits, especially in the case of larger valves and high shut off pressures. Sealing is accomplished by designing the **upstream** seat so that the shut off pressure causes it to float downstream and seal against the ball. Unlike the seat supported ball valve, which seals on the downstream seat, the trunnion mounted ball valve seals on the upstream seat. Trunnion mounted ball valves can be constructed with only one seat, which makes them unidirectional, or with two seats which makes them bi-directional.

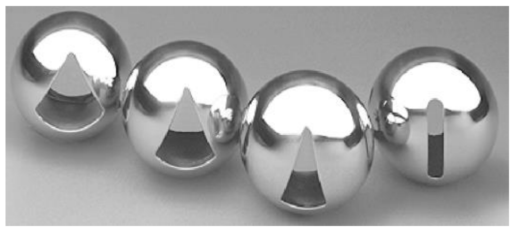
Trunnion mounted ball valves are typically found in sizes above around 6” or 10” or in high pressure applications.

The extra bearings and more complex seats make trunnion mounted ball valves more expensive than seat supported ball valves.

Full Ball

Reduced Trim

- Not offered by most major ball valve manufacturers
- Triangular “V” shaped port on one side of the ball
  - Retains the equal percentage inherent characteristic of ball valves
  - Different “V” widths determine the maximum capacity (Cv) of the valve…



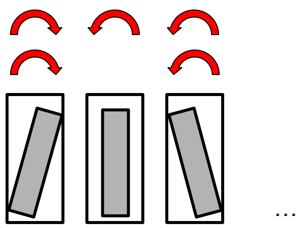
Most of the major valve manufacturers don’t offer the option of reduced trim. If you discover after the plant starts up that a ball control valve is oversized, the only option is to replace it with a smaller valve. Since the smaller valve has a different face-to-face dimension, you will have to alter the piping to accommodate the new valve plus the necessary pipe reducers.

One manufacturer, PBM, offers the option of reduced trim by using balls that have a triangular window on one side of the ball. The triangular, or “V” shape of the window retains the equal percentage inherent flow characteristic of ball valves and the width of the window determines the maximum flow capacity of the valve.

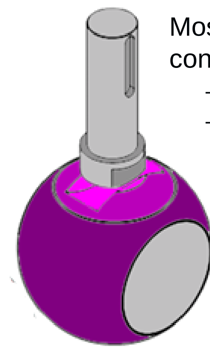
Full Ball

Potential for Backlash

- Backlash
  - Results in process dead time
  - Dead time is very destabilizing to process control



Note that the tolerance between the ball slot and the stem blade is greatly exaggerated in the above figure.



Most common ball to stem connection
- Flat sided stem
- Ball with slot

One problem with using full ball valves for throttling control is the potential for backlash.

In nearly all of the full ball designs, both floating ball and trunnion mounted ball, the top of the ball has a slot in it which is engaged by a flat sided stem as shown in the right hand figure. To allow for manufacturing tolerances, there will always be a certain amount of lost motion or backlash when the control signal changes from turning the valve from one direction to the other.

The figure on the left demonstrates what happens. When the actuator is turning the valve in the clockwise direction the play has already been taken up and small actuator movements are immediately followed by the ball giving quick and accurate flow control. When the control system calls on the ball to move in the opposite direction, and the stem starts to turn in the counter clockwise direction, it looses contact with the slot in the ball and the ball does not move, and there is no correction to the flow until all of the backlash is taken up.

When a Proportional, Integral and Derivative type process controller needs to make small process corrections, the main output from the controller is its integral action. For the typical small errors, the integral action is very slow. The time taken for the control system to turn the stem enough to take up the backlash appears in the process as dead time. Dead time in a process is very destabilizing to process control.

Backlash is lost motion due to looseness or “slop.” Imagine a loose actuator-to-valve stem coupling. Once the “slop” is taken up, as long as motion continues in the same direction, the valve will follow the actuator. When the actuator reverses direction, the valve does not move until the actuator has moved enough to take up the play.

Segment Ball Valve

Wafer or flanged

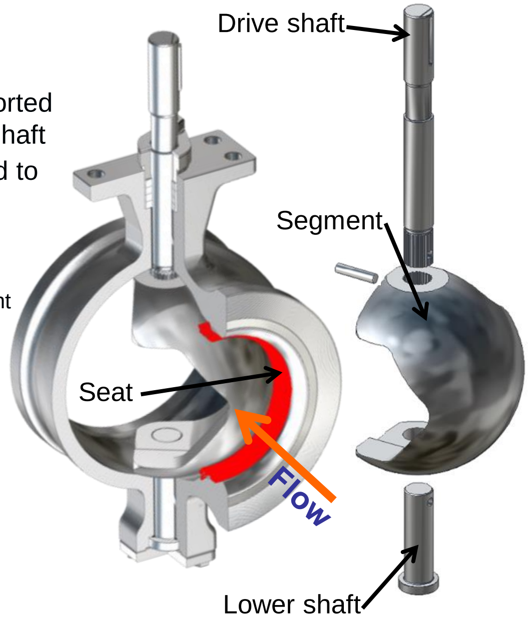
Segment ball valves come in both wafer and flanged versions.

Segment valves have one seat (on the upstream side) and are unidirectional.

Both soft and metal seats are available.

Segment Ball Valve

- Segment is fixed in place, supported by upper drive shaft and lower shaft
- Drive shaft is splined and pinned to the segment
  - Eliminates backlash
- Upstream sealing only
  - Seat is spring loaded into the segment
  - Process pressure provides additional seating force when valve is closed
  - Unidirectional
- Seats
  - Metal (most common)
  - Soft..



The segment ball valve is similar to the trunnion mounted ball valves in that the ball is kept in place by upper and lower shafts that ride in bearings.

Unlike nearly all full ball valves that have the flat blade-to-slotted ball connection, the drive shaft has a splined connection that mates with a spline in the segment. The drive shaft is then pinned to the ball to eliminate any backlash.

The segment ball valves have a single upstream seat that is spring loaded into the segment and their geometry is such that when the valve is closed the process pressure adds additional seating force to ensure good shutoff. Because of the single upstream seat, segment ball valves are unidirectional.

Segment ball valves normally have metal seats, though soft seats are optionally available.

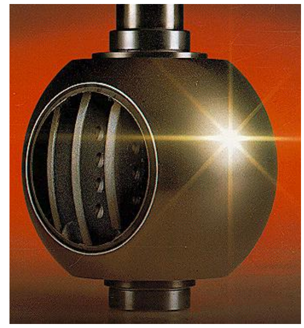
Segment Ball Valve

Reduced Trim in 1” Segment Only

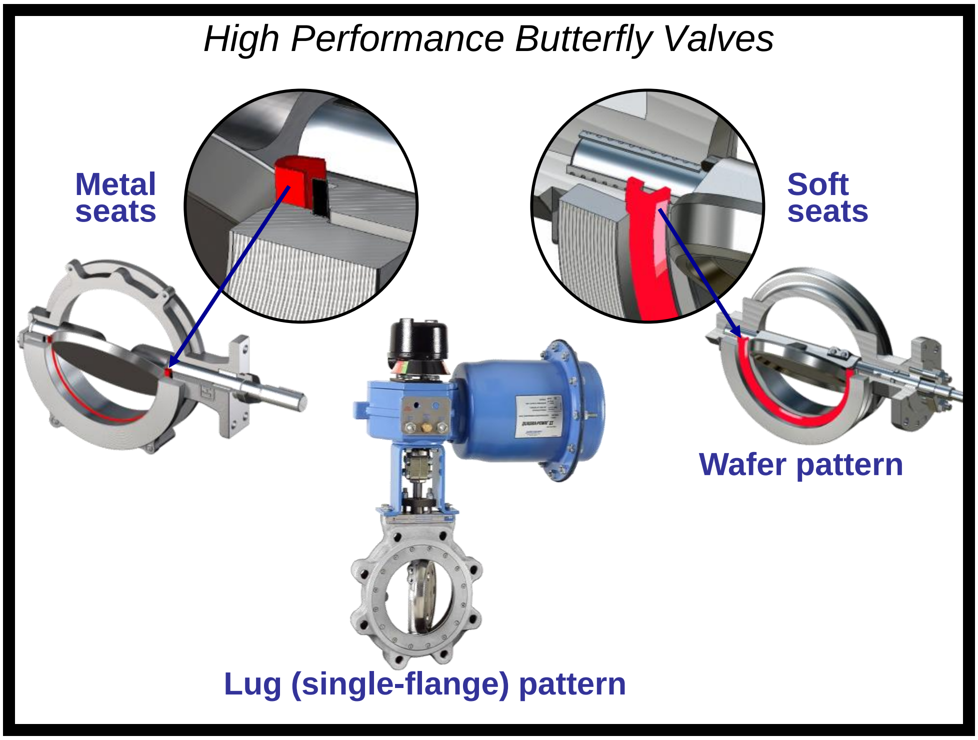
In general, ball valves are not available with reduced trim. The exception is the 1 inch segment valve where several sizes of reduced trim are available. The reductions are achieved by making the notch in the valve narrorer than it is in the full area trim valve.

Ball Valves

Noise and Cavitation Control Trim



The ball valves are not as versatile as the globe valves, but they are available with one optional design for cavitation and noise reduction. Shown here with a segment valve, a similar set of attenuator plates is available for the full ball valves.

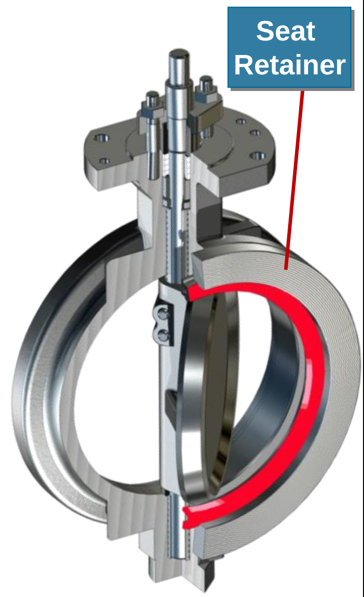


High performance butterfly valves are available both soft seated and metal seated and in wafer and lugged bodies.

The discussion comparing the wafer and flanged designs of the segment ball valves also applies to the single-flanged and wafer designs of the high performance butterfly valves.

High Performance Butterfly Valves

- Higher pressure-temperature rating than rubber lined valves
  - Soft seated, up to 500 deg. F at 400 psi
  - Metal seated, up to the maximum ANSI limit
- Much longer seat life than rubber lined valves
- Materials
  - Body compatible with process media
  - Soft seats – PTFE, Jamesbury *Xtreme*
  - Metal seats – high temp. alloy (Incoloy 825)
    - Metal seated valves cost about 25% (12”) to 75% (4”) more than soft seated valves
- Modified equal percentage inherent flow characteristic
- Seat easily replaced
- No options for reduced trim
- No options for noise or cavitation reduction



High performance butterfly valves get their name from the fact that they can be applied at higher pressures and temperatures than the traditional rubber lined butterfly valves.

Typical rubber lined butterfly valves are rated at around 200 degrees F and 200 psi. Soft seated high performance butterfly valves are rated up to 500 degrees at 400 psi and metal seated valves can be rated up to the maximum ANSI pressure and temperature limits, the same as gate valves are.

The high performance butterfly valves also have much longer seat life. This is due partly to the fact that the way the rubber lined valves seal is by the disc “squeegeeing” into the liner which can scuff the liner and create leak paths, and partly due to the offset shaft in the high performance butterfly valves that we will describe on the next slide.

These are unlined valves and instead of relying on an elastomer liner for corrosion resistance, the body material is chosen to be compatible with the process media.

In soft seated high performance butterfly valves, the seats are typically PTFE which is resistant to attack from nearly everything. In metal seated high performance butterfly valves, the seats are typically made of a corrosion resistant alloy that retains its strength at high temperatures. The Neles metal seated valve uses an Incoloy 825 seat. Metal seated valves typically cost 25 to 75% more than soft seated valves. The difference is less in larger valves than in smaller valves.

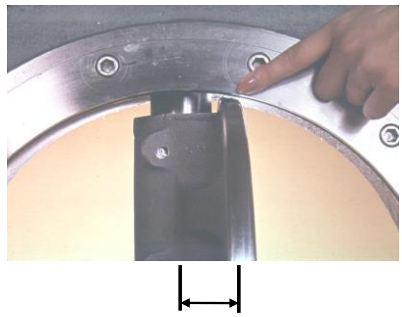
High performance butterfly valves have a modified equal percentage flow characteristic which makes them more suitable for control valve applications than rubber lined valves.

Also note that the shaft does not go through the seat as it does in rubber lined valves, but is behind the seat. The seat can be replaced by simply removing the seat retainer ring.

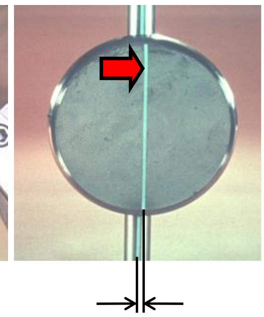
The high performance butterfly valves are the least versatile, with no options for reduced trim or noise and cavitation reduction.

High Performance Butterfly Valves

- Offset shaft – <u>**Soft seat** (Jamesbury)</u>
  - Double offset
  - Disc cams away from seat
  - Reduced seat wear
  - Less potential for seat leakage
- Drop tight shutoff…



Offset 1
Shaft offset behind disc



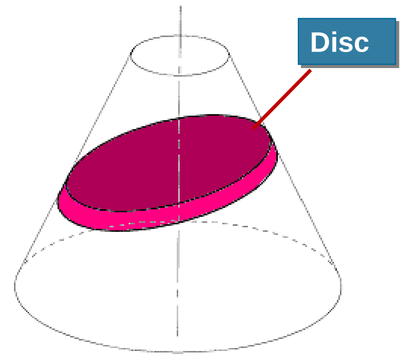
Offset 2
Shaft offset from centerline of disc

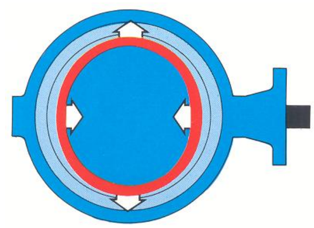
Another feature of the high performance butterfly valve is the offset shaft. In soft seated high performance butterfly valves, the offset is a double offset. The first shaft offset places the shaft behind the disc, instead of going through the center as it does in the rubber lined valves. The second offset moves the centerline of the shaft away from the centerline of the disc. The result of this double offset is that as soon as the disc starts to rotate away from the closed position, it cams away from the seat, breaking contact with the seat at the 12 and 6 o'clock positions. You can see in the picture that the disc is not touching the seat. When the disc goes from closed to open and back again it is not rubbing against the seat, wearing a potential leakage path. Jamesbury was the first manufacturer to make high performance butterfly valves with the double offset shaft. Now, all of the many brands of soft seated high performance butterfly valves have the double offset.

Soft seated high performance butterfly valves give the same drop tight shutoff that soft seated ball valves do.

High Performance Butterfly Valves

- Offset shaft – **<u>Metal seat</u>** <u>(Neles)</u>
  - Triple offset
  - Offsets 1 and 2 same as double offset
  - Offset 3
    - The disc is shaped like an oblique slice through a cone which produces an elliptical shape
    - The elliptical disk stretches the round seat to conform around the disc ensuring a tight seal between the disc and seat…





Unlike soft seated high performance butterfly valves that have the "double offset," most metal seated high performance butterfly valves have a triple offset, with the third offset being the shape of the disc. The disc is machined to have an elliptical shape. Its shape is the same as would be obtained by taking an oblique slice from a cone.

The elliptical disc, when cycling into the round seat, stretches the seat to tightly conform around the disk, ensuring a tight seal around the full circumference of the disc. This third offset is not required in soft seated high performance butterfly valves because the PTFE seat has enough flexibility that cycling a round disc into a round seat can provide a drop tight seal.

Control Valve Selection

<u>OLD METHOD</u>

Use the same type valve we always have.

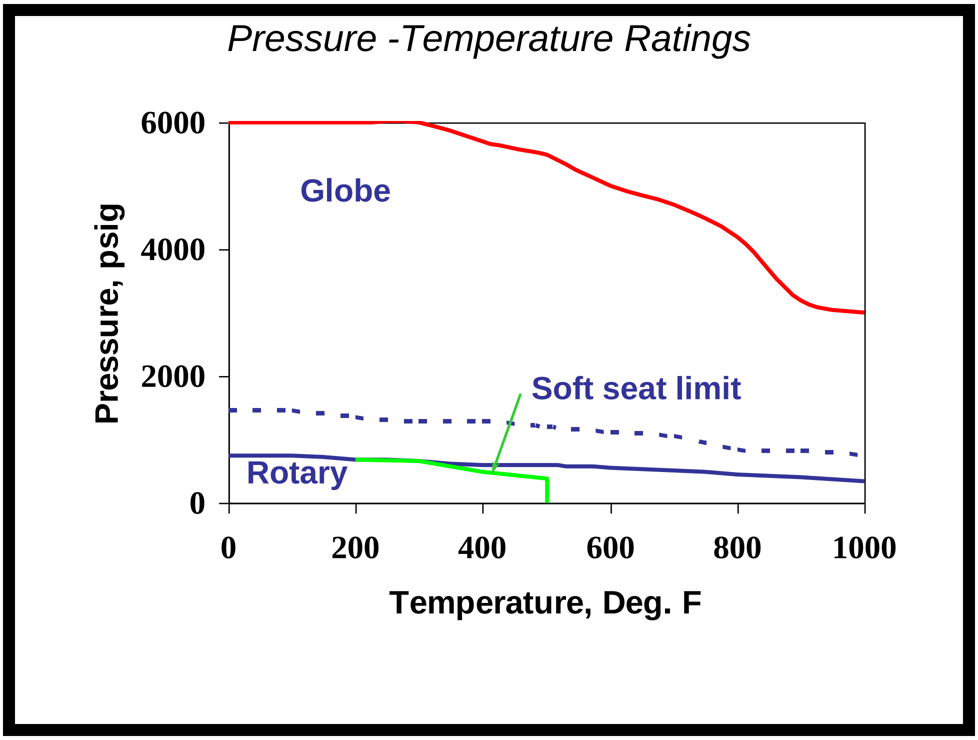
<u>NEW METHOD</u>

Select the best valve type for each application

- **Operating temperature and pressure**
- **Cost**
- **Weight**
- **Flow characteristic**
- **Cavitation potential**
- **Stem sealing**

If one control valve type was the best for all applications, then of course we, and everyone else, would only make that one type of valve

In the old days it was easy to decide what type of valve to use. Now that everyone must be more competitive and watch expenses, we need to take a close look at all aspects of an application and choose the most appropriate valve



One of the most important considerations in selecting a vale style is its pressure temperature capability.

Globe valves are very versatile, rugged valves. They are also very expensive valves. If you need a valve for 3000 psi and 600 degrees, then the only choice is a globe valve. On the other hand, a majority of the applications fall into the realm of the much less expensive ball, butterfly and segment ball valves.

These are approximate pressure temperature ratings based on standard catalog products.

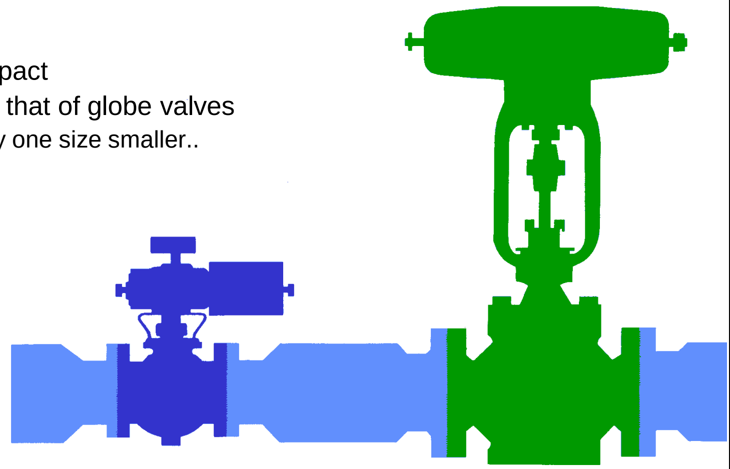
Globe valves are available as standard products through ANSI 2500.

Some rotary valve styles are available in ANSI 600, but the vast majority are normally available through ANSI 300. The dashed line represents the limit for Class 600 valves.

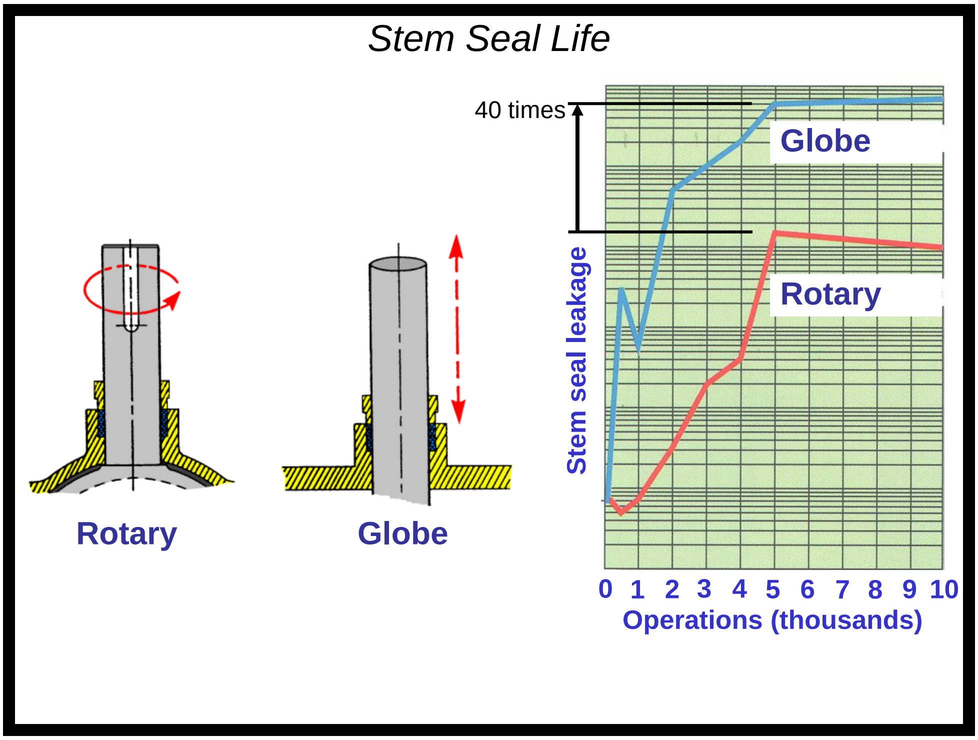
Flow Capacity and Compactness

Rotary valves

- More compact
- Cv double that of globe valves
  - Usually one size smaller..



Because the actuator on rotary valves lays over on its side instead to standing straight up, the rotary valves are more compact than the globe valves. Because the flow capacity of the ball and butterfly valves is about twice that of the same size globe valve, they tend to be one size smaller in the same application making them even more compact.



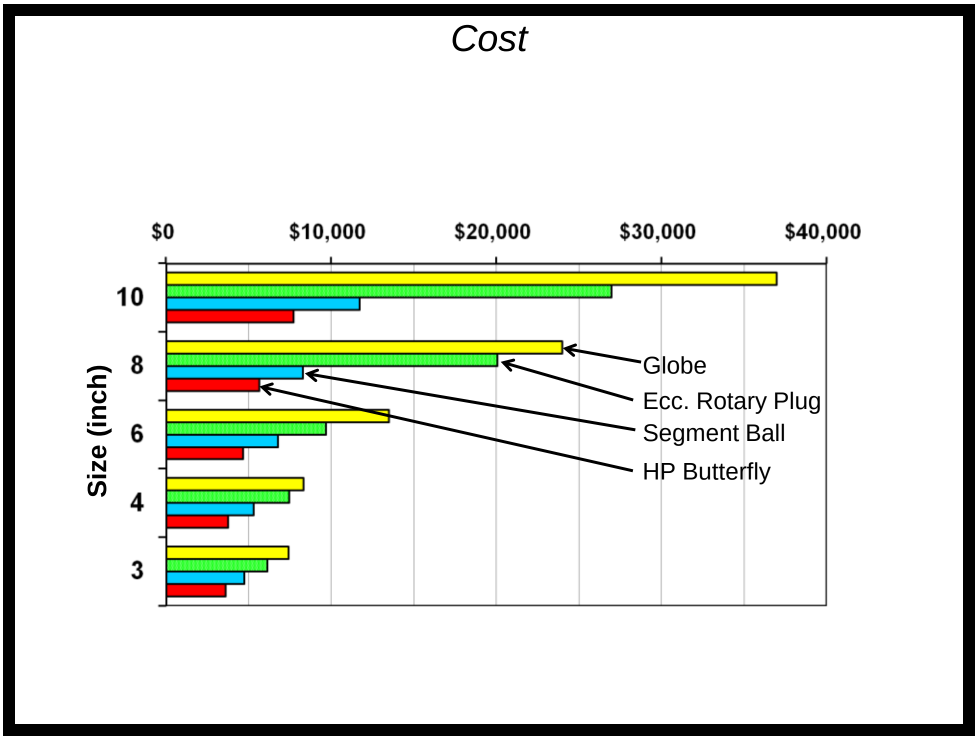
Rotary valves tend to have a much longer stem seal life than linear motion valves. With the linear motion valves, any dirt on the stem is dragged through the packing. Even a very small amount of corrosion on the stem inside the valve when pulled up through the packing can damage the packing.

With rotary valves the same portion of the stem is always in contact with the valve stem.

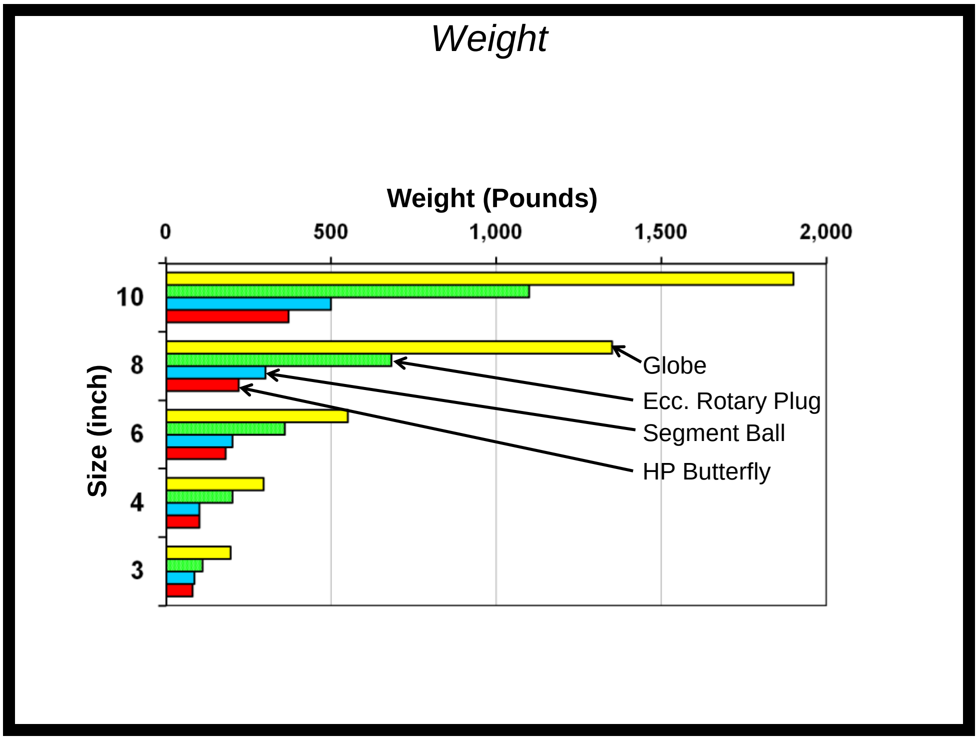
The graph is data from a test comparing a globe valve with a rotary valve.

After 5000 operations, the stem leakage from a typical globe valve is 40 times greater than from a rotary valve.

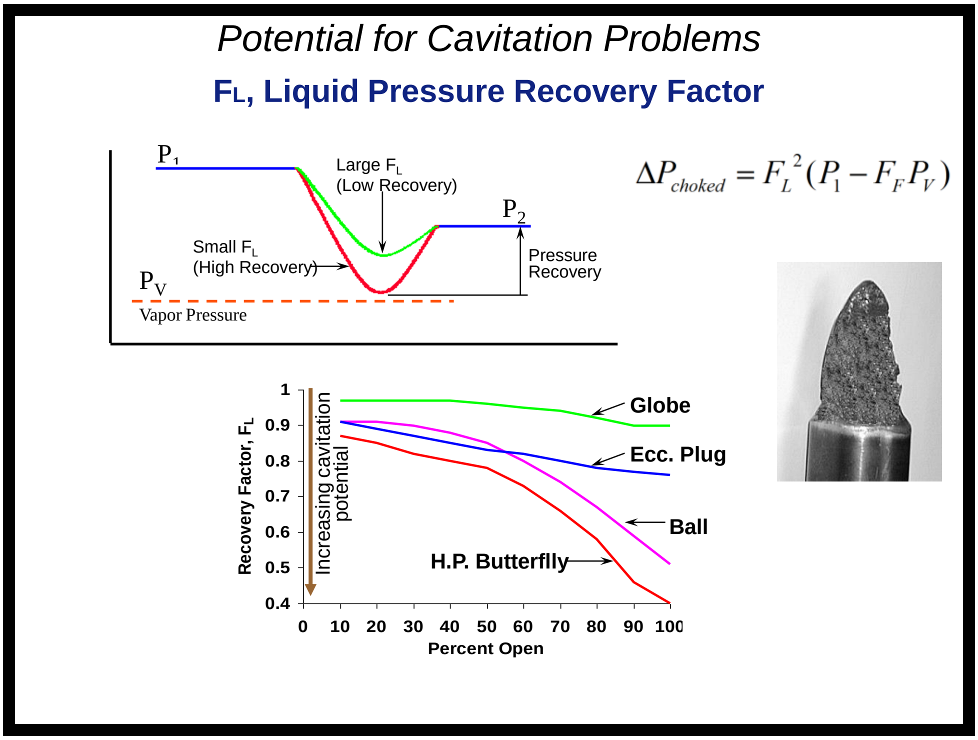
Note that the leakage scale on the graph is a logarithmic scale.



Everyone is concerned with cost. This graph is pretty much self explanatory. Not so obvious, however is the fact that the rotary valves have greater capacity size for size. For example, it is possible that an application which requires a 10 inch globe valve may be able to use an 8 inch butterfly or segment ball valve, making the savings even more dramatic.



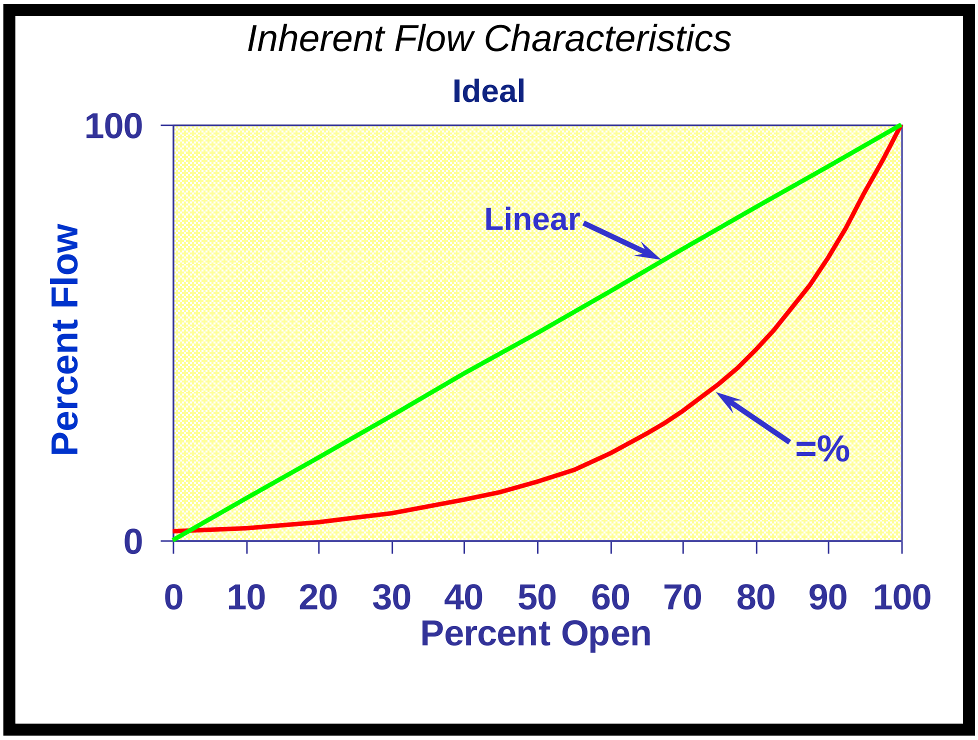
A graph of comparative weight looks quite similar to the graph of comparative cost. In reality, what you are paying for is metal. Weight is an important consideration for ease of installation and maintenance and influences the need for additional pipe supports.



A valve's F_L or pressure recovery factor represents its tendency to thave cavitation problems. The higher capacity and lower cost rotary valves are also the high recovery valves which have a greater tendency to cavitate. This does not mean that the ball and butterfly valves are not good for control, but it is important to make sure that these valves are being properly applied.

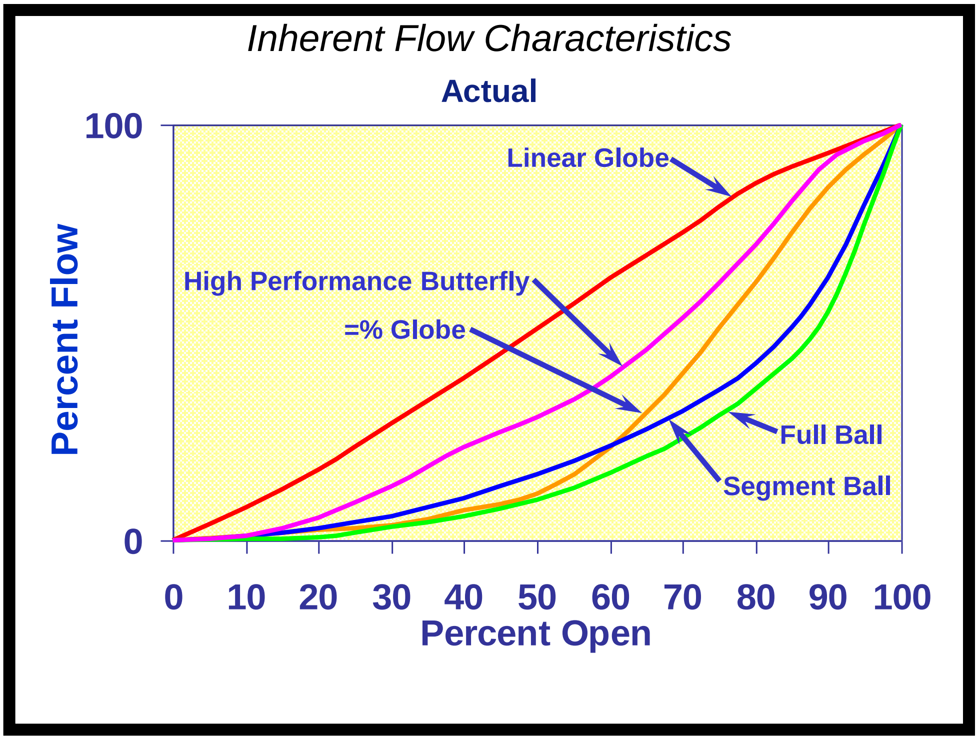
A full discussion of liquid pressure factor and cavitation is included in the section on Incompressible Flow in Control Valves.

For the purposes of the current section, the point is that different valve styles have different tendencies to have cavitation problems and those with LOWER values of F_L have a greater potential for cavitation.



It is also important to select a valve with a flow characteristic that matches the requirements of the process. This will be discussed in a later section.

These are the “ideal” inherent flow characteristics that are often referred to when discussing control valves.



Let's see what is actually available. This graph shows typical characteristics, plotted from actual manufacturers data.

Globe valves, the most versatile of the control valves, are available with either linear or equal percent trim.

The ball valves, both full ball and segment ball, in practice provide very nearly an ideal equal percent characteristic, and that is the only way they come. In fact, the typical ball valve's characteristic comes closer to the ideal equal percent curve that the typical equal percent globe valve.

The high performance butterfly has a characteristic that is half way between linear and equal percent.

Summary

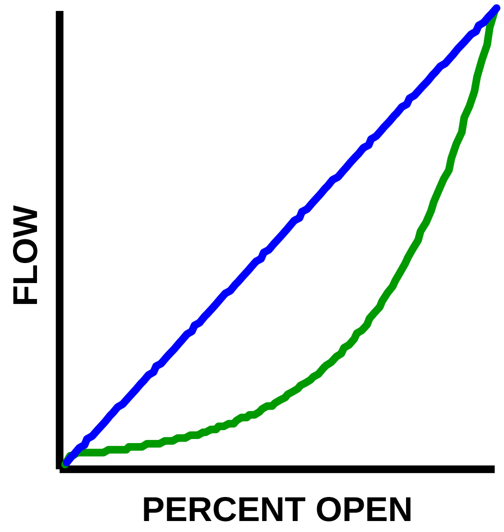
Valves	Globe Top Guided	Globe Cage Guided	Ball Full/Segment	Eccentric Rotary Plug	High Performance Butterfly
Cost	High	High	Medium	Medium	Low
Weight	High	High	Medium	Medium	Low
Flow Capacity compared to globe	1 X	1X	2 X	1 X	2X
Cavitation Potential	Low	Low	Medium	Medium	High
In-line repairable	Yes	Yes	No	No	No
Inherent Flow Characteristic	=%, Linear, Quick Opening	=%, Linear, Quick Opening	=%	Modified Linear	Modified =%
Cav./Noise Reduction Options	No	Yes	Some	Some	No
Suitable for High Pressure Drop	Limited	Yes	Limited	Yes	Limited
Suitable for Dirty Service	Yes	No	Yes	Yes	Yes
Suitable for Slurries	Limited	No	Yes	Yes	Limited
Suitable for Pulp Stock	No	No	Yes	No	Limited

This chart is a summary of the various characteristics of the different control valve types.

Flow Characteristics

Control Valve Characteristics

- **Inherent**
- **Installed**



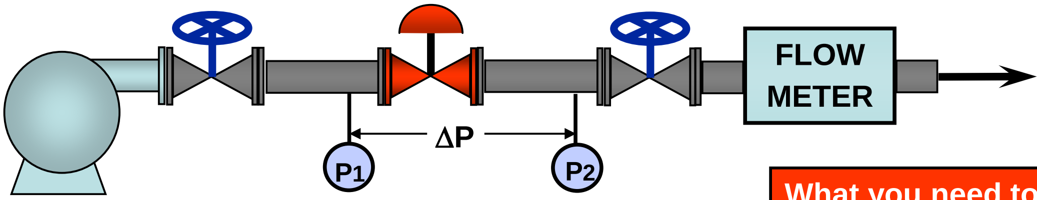
When we talk about control valve characteristics, what we mean is the relationship between valve opening and flow.

A control valve has two characteristics, the INHERENT characteristic, and the INSTALLED characteristic.

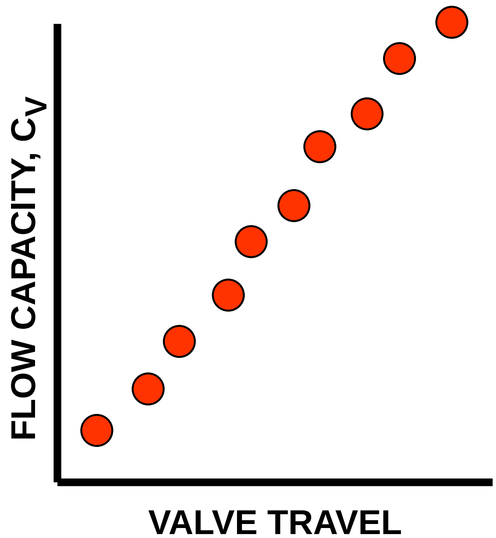
The user is concerned with the installed characteristic, that is, what is the relationship between valve opening and flow in the particular system the valve will be installed in.

The valve manufacturer does not know what sort of system the valve will go in, so he can only publish the inherent characteristic, that is the relationship between valve opening and flow ignoring all system effects.

Inherent Characteristic



$$Cv = Q\sqrt{\frac{G}{\Delta P}}$$



What you need to remember about the inherent characteristic:

1. It is the relationship between valve opening and flow capacity (C_V) of the valve while the pressure drop is held constant. (No system effects.)

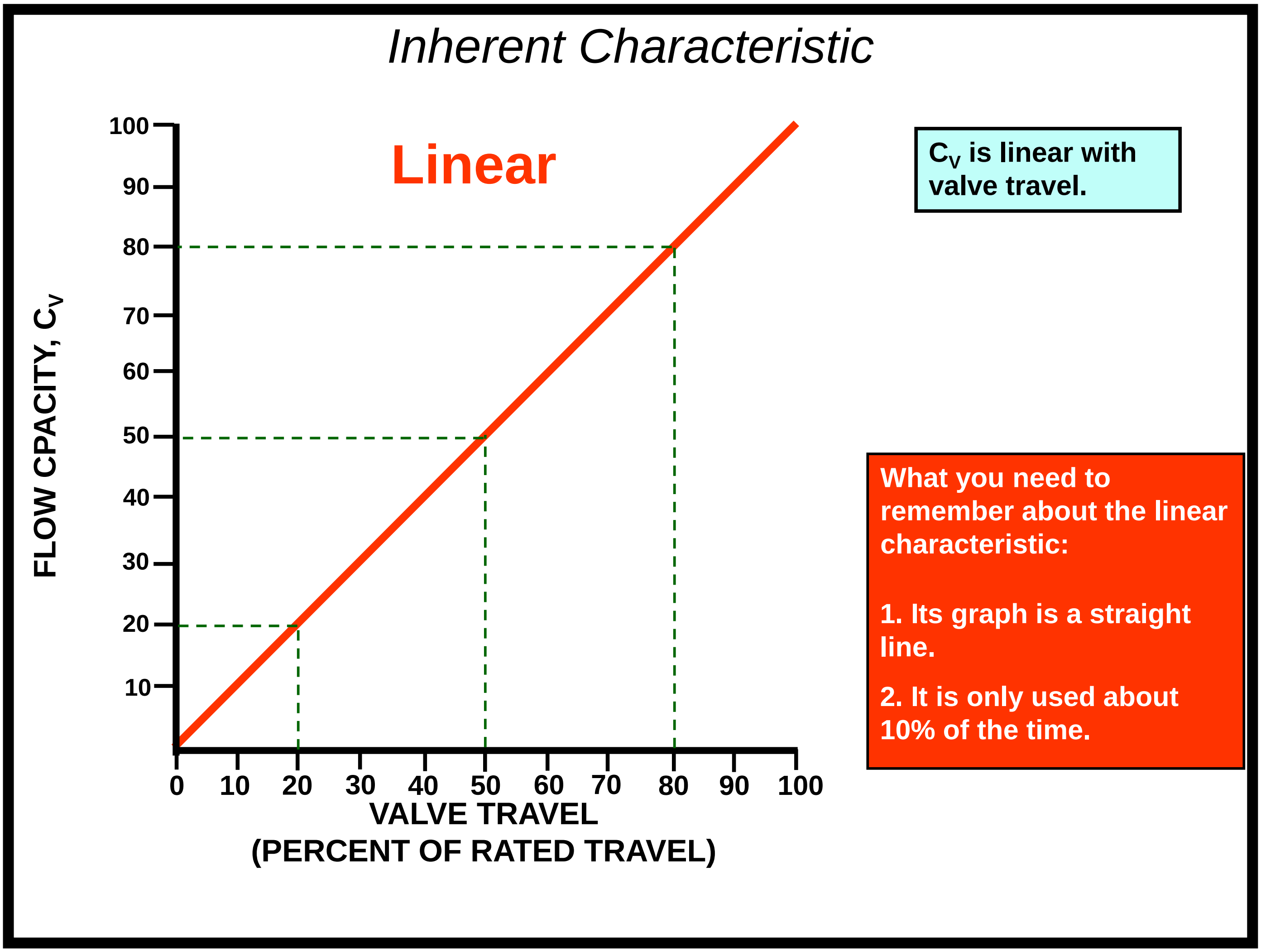
2. It is the characteristic that is published by the manufacturer.

The way valve manufacturers determine the inherent characteristic of a control valve is to put it into a test flow loop. The test valve is opened to 10% open, the pump is started, then the manual valves are adjusted until the pressure difference across the test valve (P1 – P2) is equal to the pressure specified in the ISA standard on control valve flow capacity testing. The flow is then measured and a mark is made on the graph paper at the appropriate place.

Next the valve is opened to 20% open. Because this causes the flow to increase, the pump pressure decreases, and there is more pressure loss in the piping and manual valves. As a result P1 and P2 change. Before anything else is done, the manual valves are readjusted until the pressure difference across the test valve is restored to its original value. Then the flow is measured and another mark is made on the graph paper.

This process is repeated at each 10% increment of valve opening up to 100%, always maintaining the same pressure differential across the test valve..

The main point is that the inherent characteristic is determined by eliminating any system effects, so that the resultant graph represents valve behavior only.



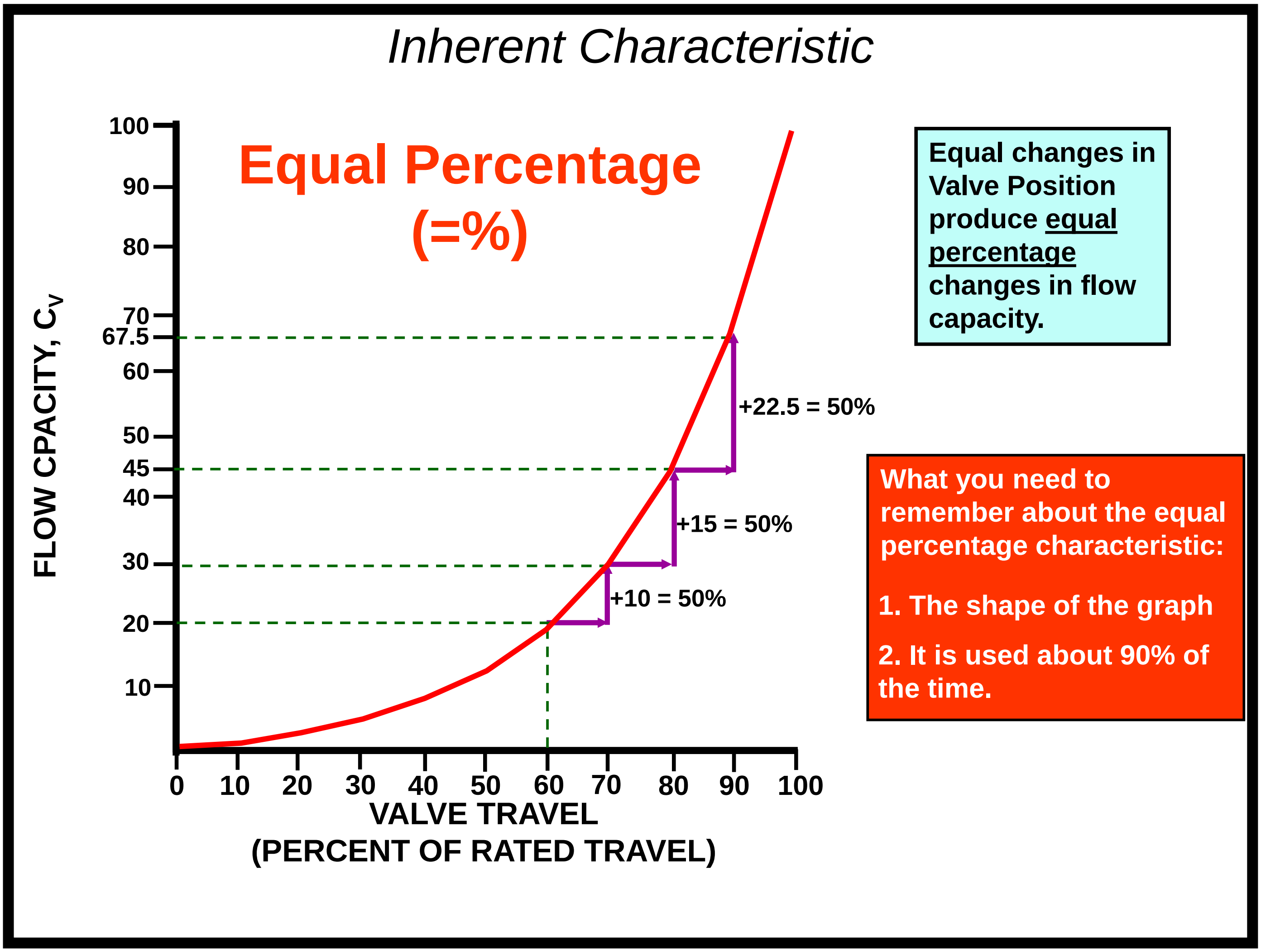
There are two common inherent characteristics

The linear characteristic is the easiest to understand:

At 20% open the valve has 20% of its maximum flow capacity (Cv).

At 50% open the valve has 50% of its maximum flow capacity (Cv) and so on..

It turns out that the linear characteristic is only used about 10 percent of the time.



The equal percent is the most common characteristic. Somewhere between 80 and 90 percent of control valves are specified to have an equal percent characteristic.

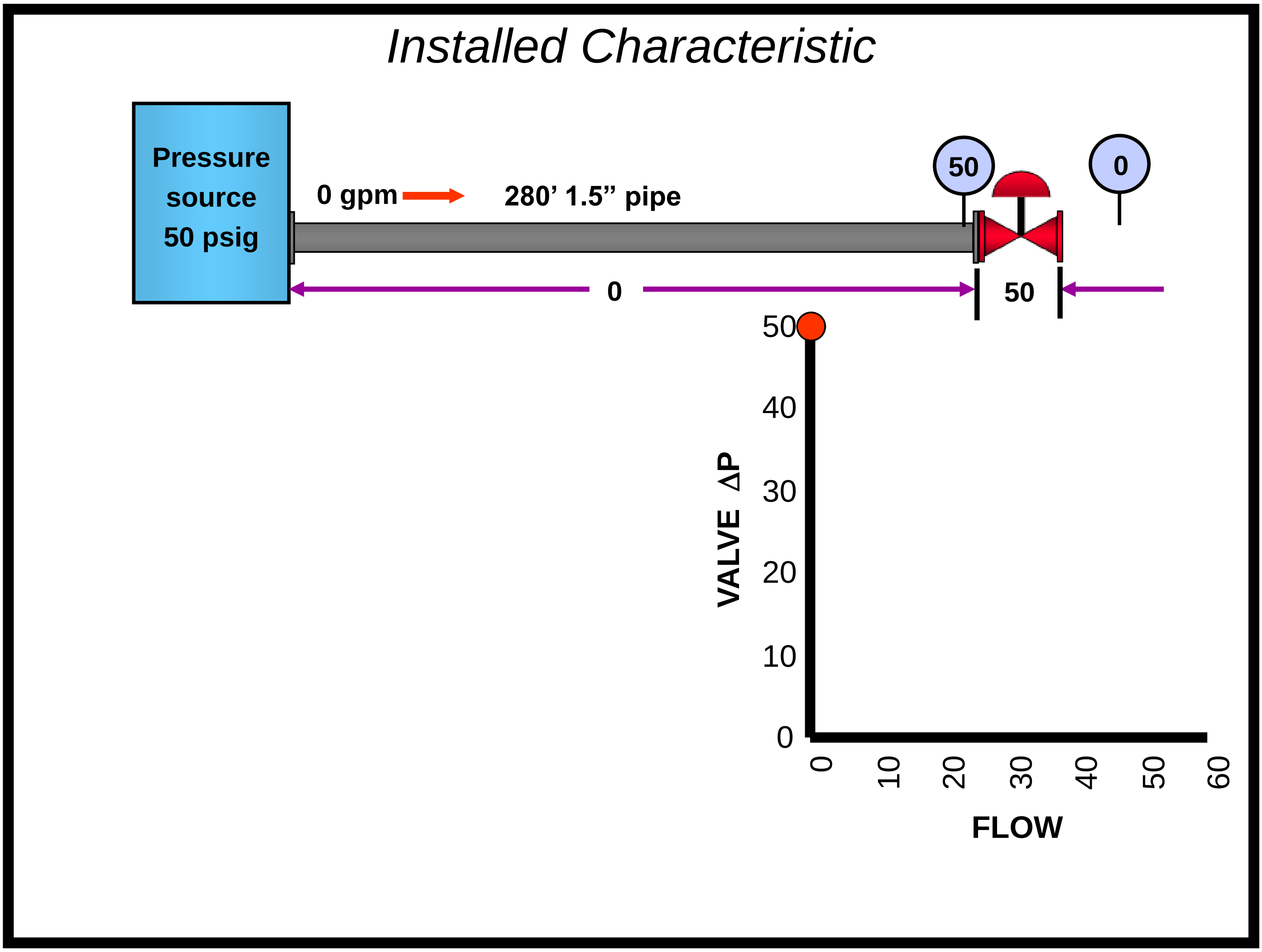
The figure shows how the equal percent characteristic works. The example is for a *typical* equal percentage valve with a full open (rated) Cv of 100. For example, if we were to start with this *typical* equal percent valve at 60% open its Cv would be 20. Moving the valve open by 10 percentage points to 70% open causes the Cv to increase to 30. This is an increase of 10 Cv units. Note that at this point 10 Cv units is an increase of 50% of the original Cv of 20.

Now we open the valve by another 10 percentage points from 70 to 80 percent open. The Cv at this point is 45. This represents an increase of 15 Cv units. An increase of 15 is an increase of 50% over the previous value of 30.

Again we open the valve another 10 percentage points from 80 to 90 percent open. At this point the Cv is 67.5. This represents an increase of 22.5 Cv units. An increase of 22.5 Cv units represents an increase of 50% of the previous value of 45.

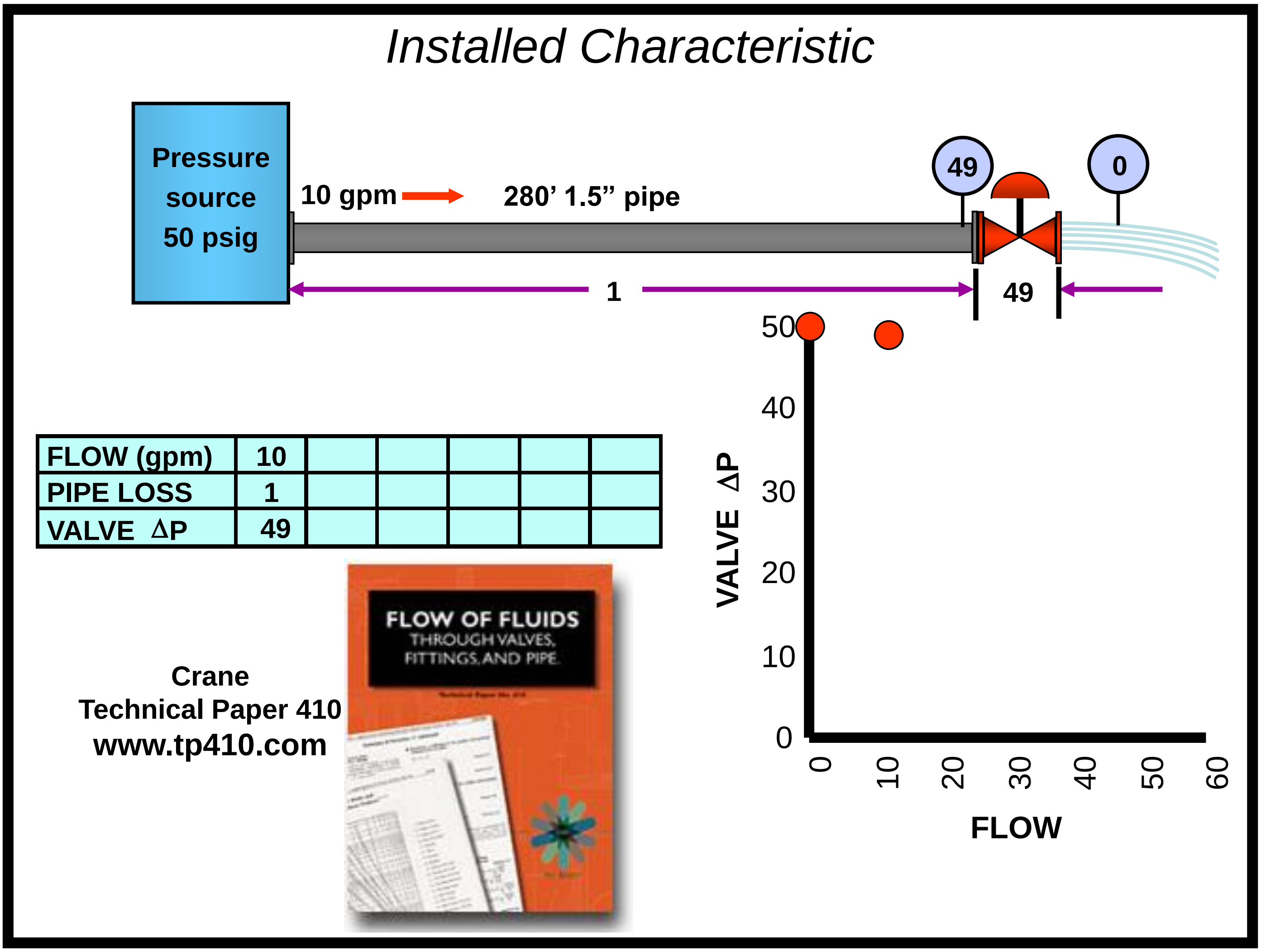
This illustrates the definition of the *equal percentage* characteristic in the box at the upper right. The valve movements of 10 percentage points are equal changes in valve position (they are all the same size changes) and they each produce equal percentage, (in this case, 50%) changes in Cv. As the valve opening becomes larger, the 50% increases represent larger changes in Cv. (Note: The 50% figure in the example is typical, but will vary slightly depending on the valve type.)

It turns out that systems that behave in a linear manner are easier to control than non linear systems. (We will see why on subsequent pages.) If this is true, why would the equal percentage characteristic, which is very non linear, be used so widely? The answer is because of the **installed characteristic.**



Most control valves are installed in systems that have a lot of pipe.. The example shows how pressure loss would vary in 280 feet of 1.5 inch pipe as the control valve adjusts flow over a range of 0 gpm to 60 gpm.

When the valve is closed, there is, of course, no flow. So if the pressure source has a pressure of 50 psig, the pressure at the inlet to the valve is also 50 psig. Because we are discussing control valves, what we are going to investigate in the example is the behavior of the pressure drop across the valve as the valve opens to increase the flow in the system. The pressure at the inlet to the valve (P1) is 50 psi. The valve outlet is exposed to the atmosphere, where the pressure (P2) is zero (0) psig. So the pressure drop across the valve is 50 minus 0, which equals 50. The first point on the graph is then at zero flow and 50 psi.

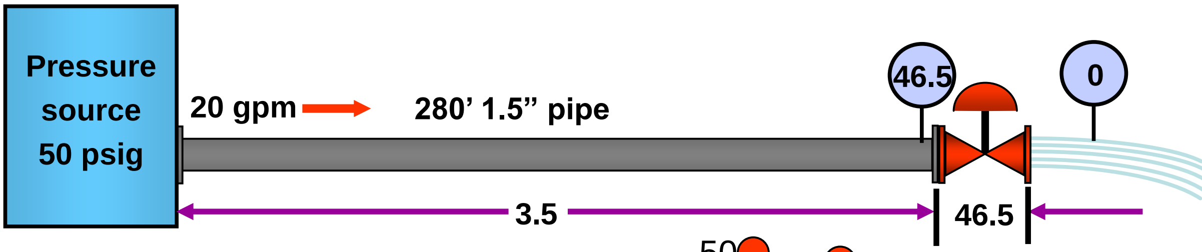


FLOW (gpm)	10					
PIPE LOSS	1					
VALVE ΔP	49					

We now open the valve until the flow equals 10 gpm. Looking up 280 feet of 1.5 inch pipe in the "Crane Handbook" *Flow of Fluids through Valves Fittings and Pipe (Technical Paper 401) we* find that the pressure loss in the pipe will be 1 psi. This means that the pressure at the valve inlet (P1) is 50 minus 1 or 49 psig. Because the flow is going to atmosphere where the pressure (P2) is zero, the pressure drop across the control valve is 49 minus zero or 49 psi. Forty nine psi is entered in the table for "Valve delta p" and a point is put on the graph at 10 gpm and 49 psi.

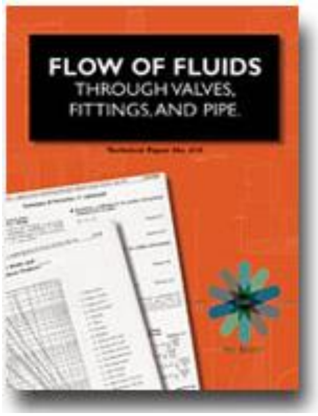
Note that the Crane Technical Paper 410, although is updated every few years, has been around since 1942.

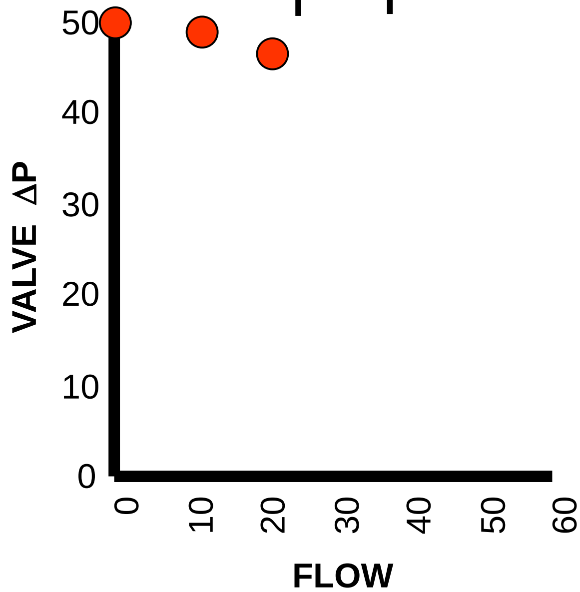
Installed Characteristic



FLOW (gpm)	10	20				
PIPE LOSS	1	3.5				
VALVE ΔP	49	46.5				

Crane
Technical Paper 410
www.tp410.com



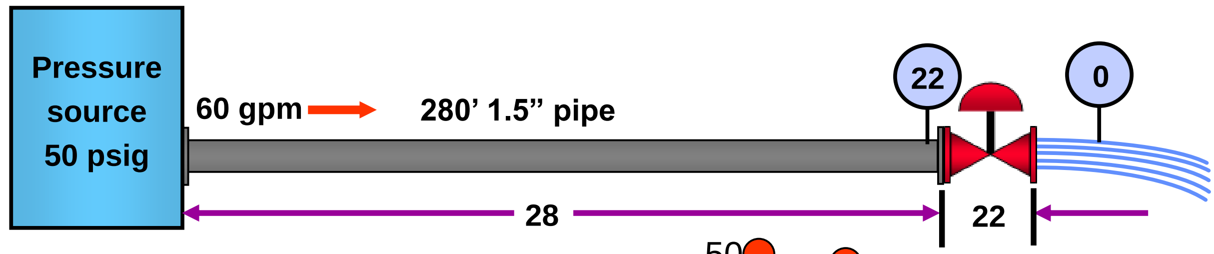


Next we open the valve until the flow increases to 20 gpm. When we look up the pressure drop at 20 gpm we see something interesting. Doubling the flow (from 10 to 20 gpm) results in much more than a doubling of the pressure loss.

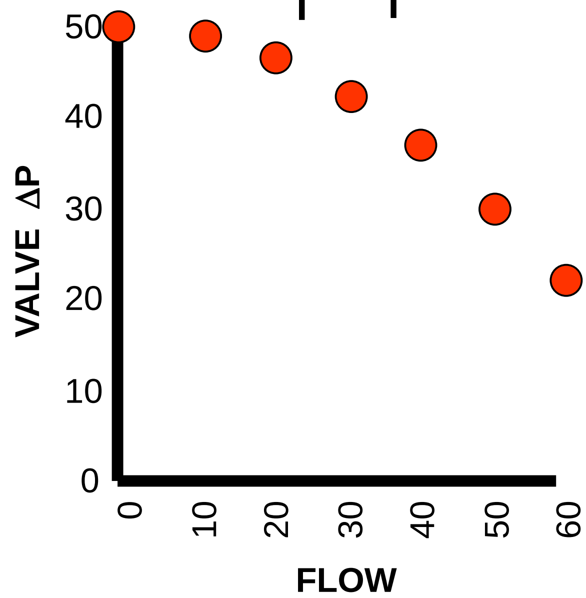
(Pressure loss in the pipe increases from 1 psi to 3.5 psi!)

The next page shows how the pressure drop across the control valve decreases as the valve is opened to increase the flow in 10 gpm increments.

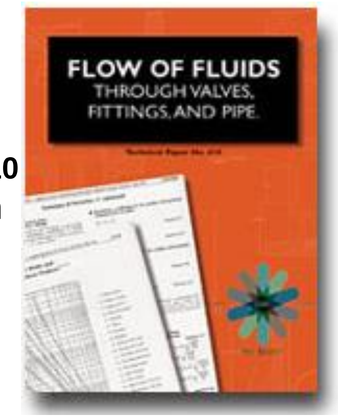
Installed Characteristic



FLOW (gpm)	10	20	30	40	50	60
PIPE LOSS	1	3.5	7.5	13	20	28
VALVE ΔP	49	46.5	42.5	37	30	22



Crane
Technical Paper 410
www.tp410.com

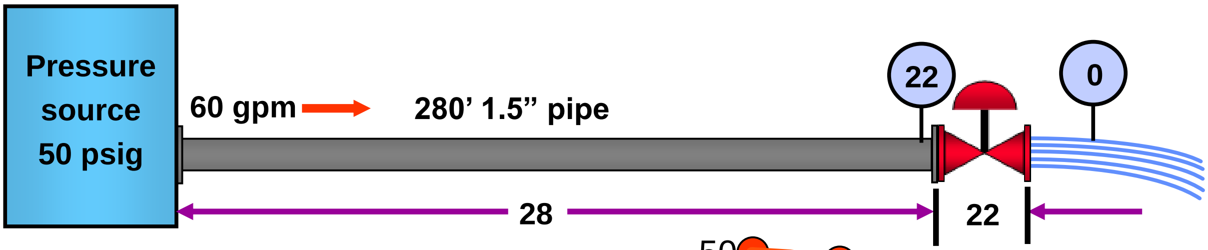


Here we have graphed the remaining flow points (30 gpm through 60 gpm). The pressure drops and pressures are shown along the pipe for the 60 gpm case. For each case, we have looked up the pipe pressure loss and subtracted the pipe loss from the 50 psig pressure source to obtain the pressure at the valve inlet. This pressure minus atmospheric pressure (0 psig) is the valve pressure drop which is entered in the table and the graph.

Notice that the pressure drop across the control valve decreases at a much faster rate than the flow is increasing, giving a graph that is non linear.

Typically, the relationship between flow in a pipe and the pressure loss in a pipe is approximately a flow squared relationship.

Installed Characteristic



FLOW (gpm)	10	20	30	40	50	60
PIPE LOSS	1	3.5	7.5	13	20	28
VALVE ΔP	49	46.5	42.5	37	30	22

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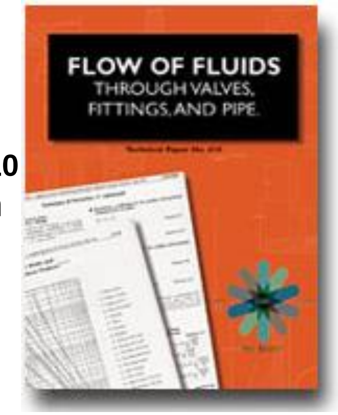
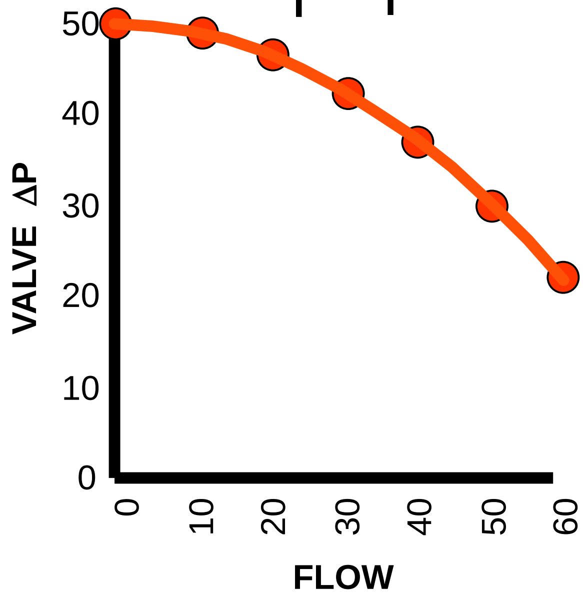
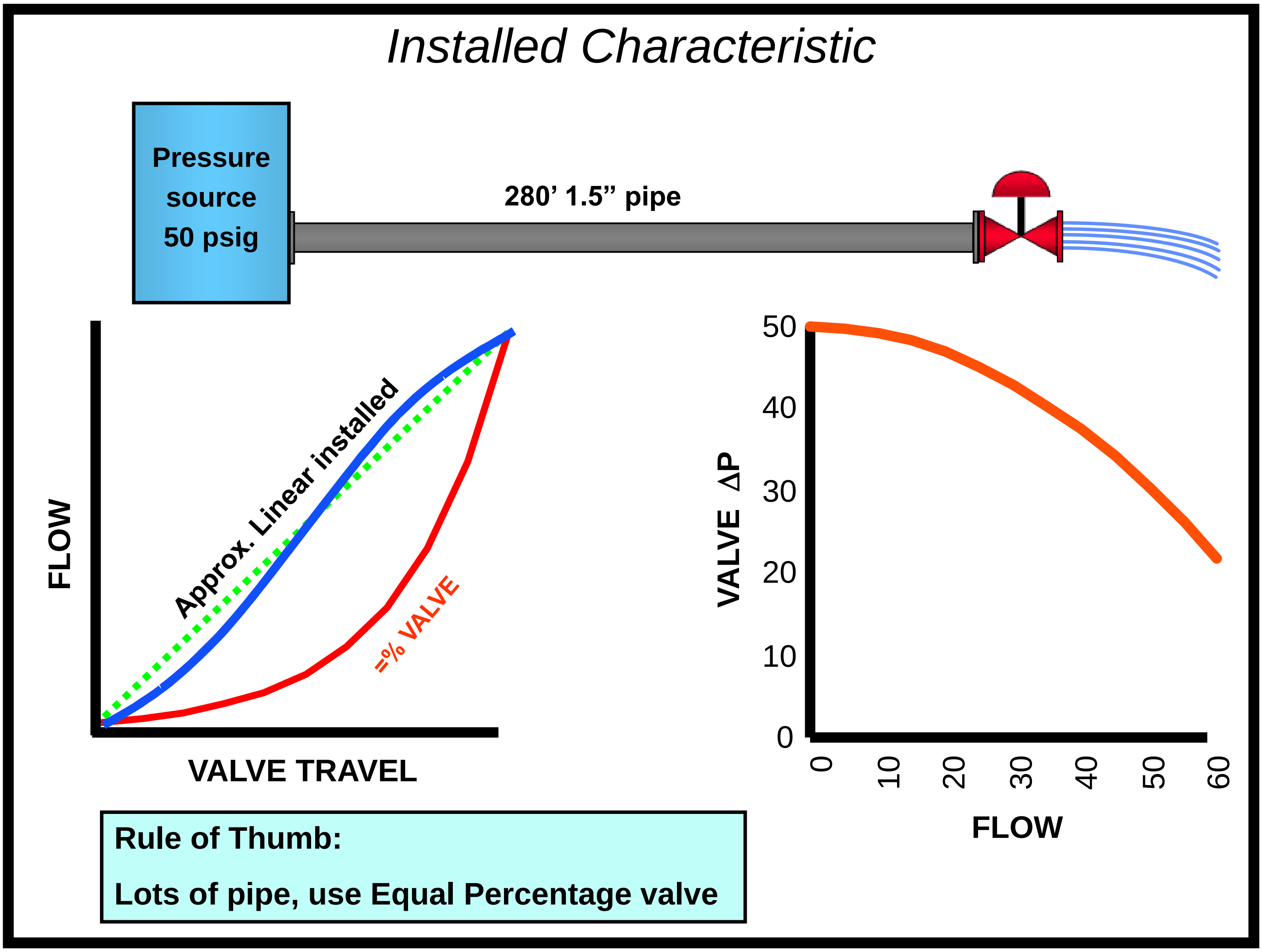




Here the dots have been connected with a smooth curve.



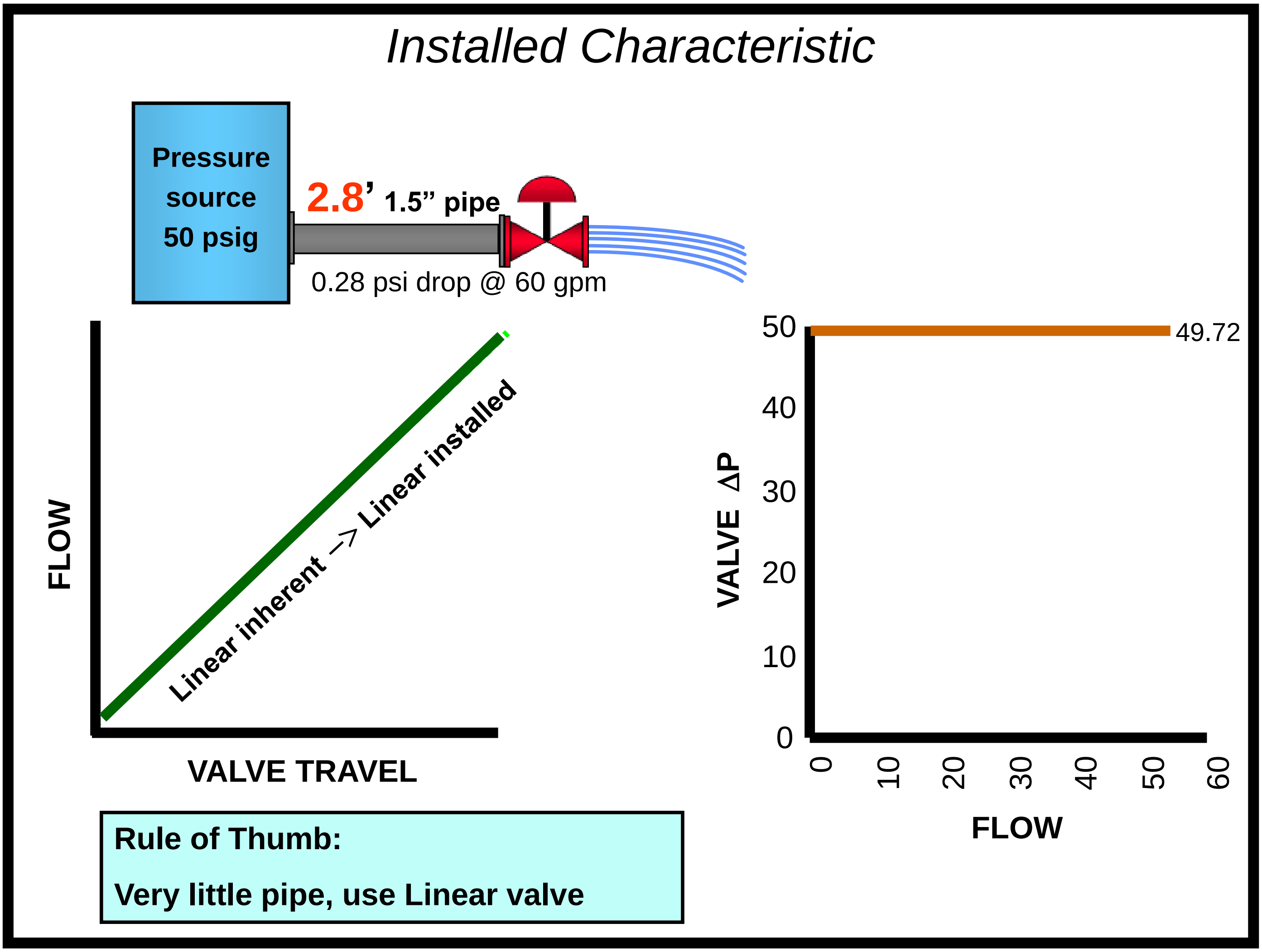
Let’s see what happens when we install an equal percentage valve in this typical system with a lot of pipe. Starting with the valve 100% open, the flow would be at its maximum value. Then start closing the valve. As the flow decreases, the pressure drop across the valve increases (as shown in the Right hand graph).

The increasing pressure drop partially resists the decreasing flow.

The result is an INSTALLED characteristic that is very nearly linear. The reason that so many equal percentage valves are used is that most systems include a lot of pipe or other process equipment (centrifugal pumps have the same effect). Using equal percentage valves in these systems gives a nearly linear installed characteristic, which makes the system easier to control. We will see later why a linear installed characteristic makes the system easier to control.

The system characteristic graph and the equal percentage graph are not exact mirror images of each other, so the installed characteristic tends to be somewhat “S” shaped although not always as symmetrical as shown here.

As a general rule of thumb, an equal percentage valve is the best choice for systems with a lot of pipe (or other pressure consuming elements such as fittings, heat exchangers, etc).



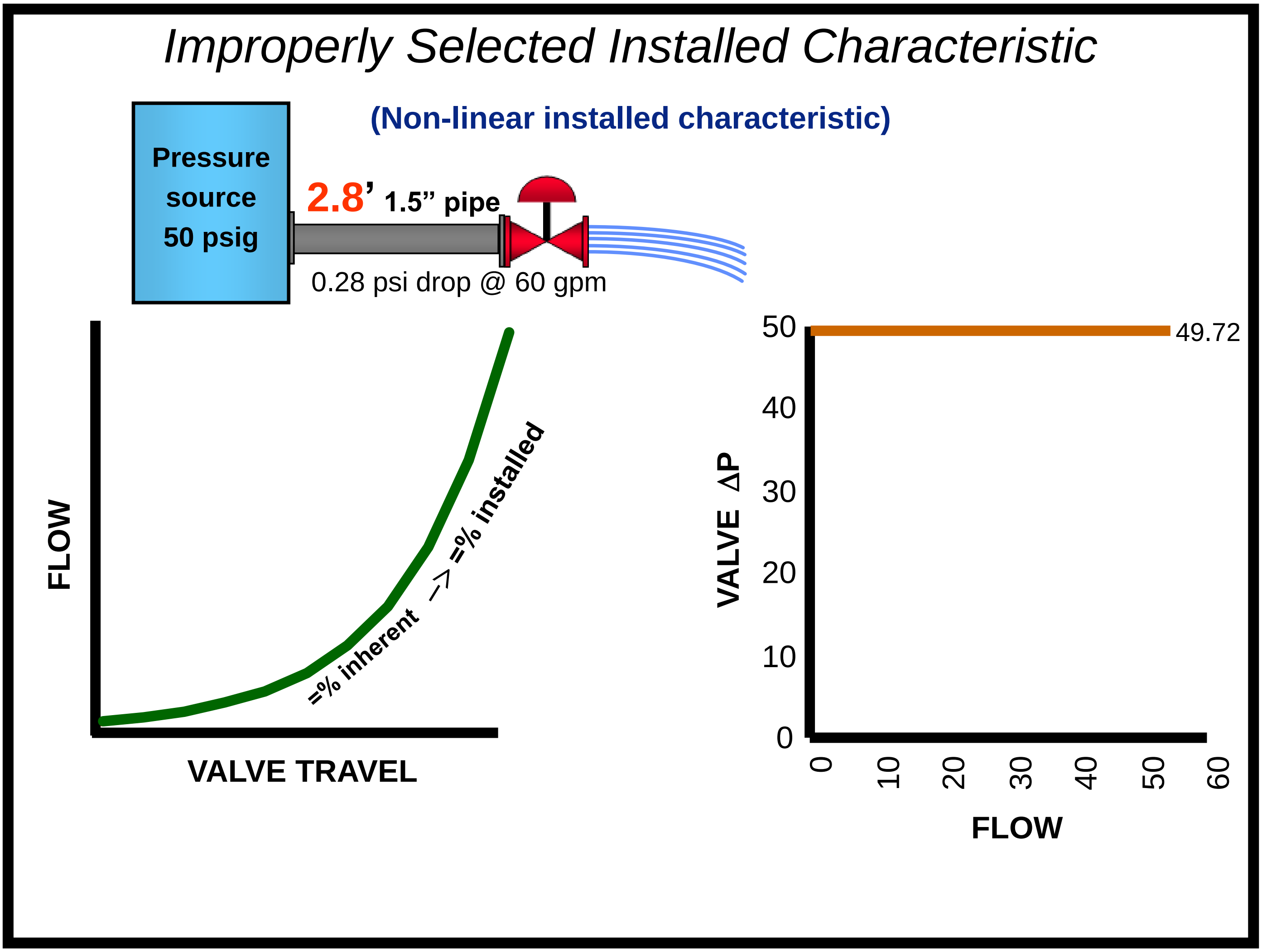
If we had only 2.8 feet of pipe instead of 280 feet, (1 one hundredth as much pipe) the pressure loss in the piping would be 1 one hundredth of the loss in the 280 feet of pipe. At 60 gpm the pressure loss would be about 0.28 psi instead of 28.

As the valve goes from zero to 100% open the pressure drop across the valve goes from 50psi to 49.72 and the graph of pressure drop across the control valve is essentially a flat line.

If we installed a valve with a linear inherent characteristic in this system, as the valve starts to close from 100% open the flow goes down, but the pressure drop remains constant, so the installed characteristic is the same as the inherent characteristic of the valve which is linear.

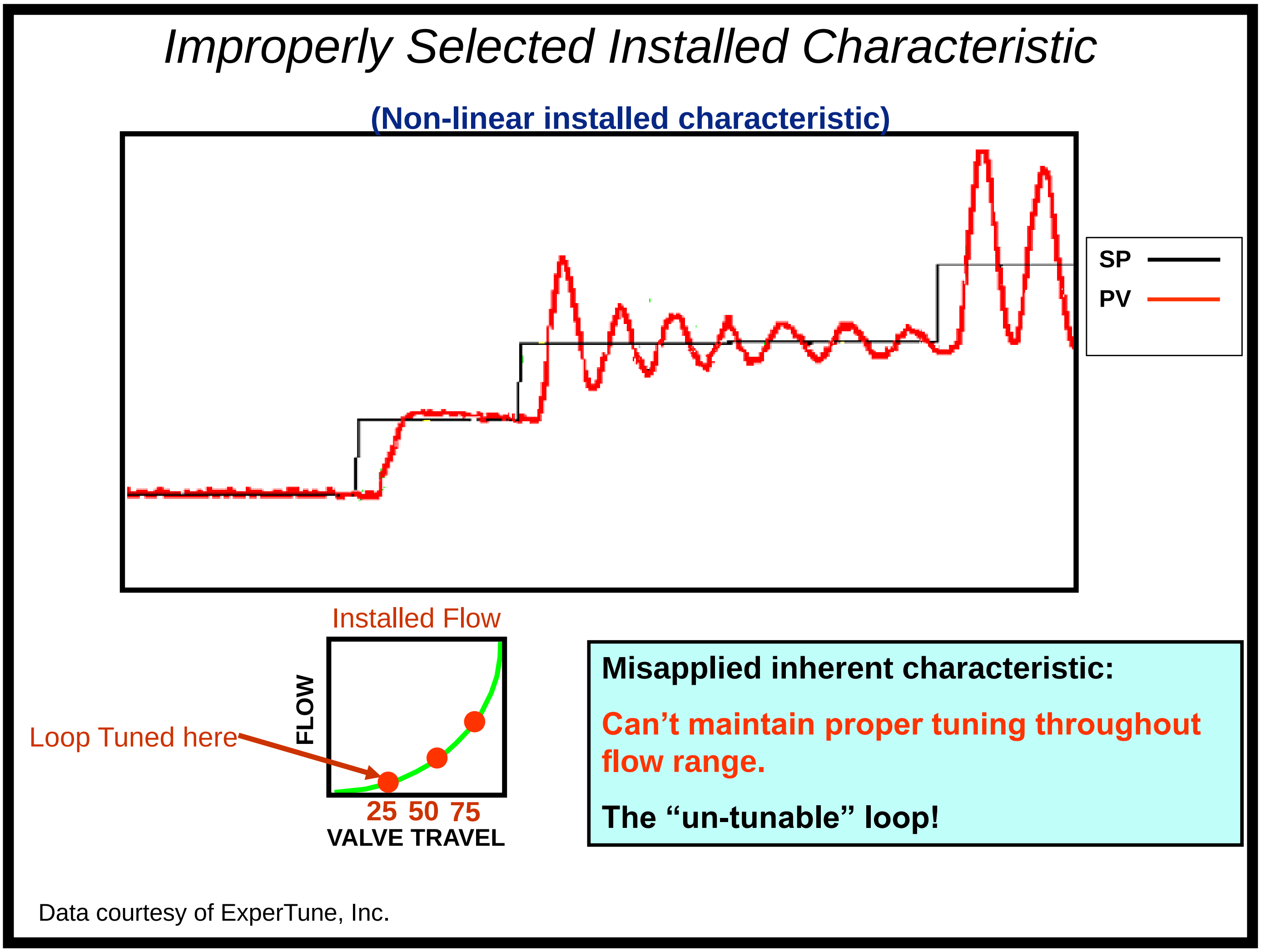
Remember that the way the inherent characteristic is determined by the manufacturer is to measure flow throughout the valve's opening in a system where the pressure drop remains constant, so it is not surprising that the installed characteristic is the same as the inherent characteristic in a system where the pressure drop remains constant.

As a general rule of thumb, a linear valve is the best choice for systems with very little pipe.



Here, we have misapplied an equal percentage valve in a system with very little pipe.

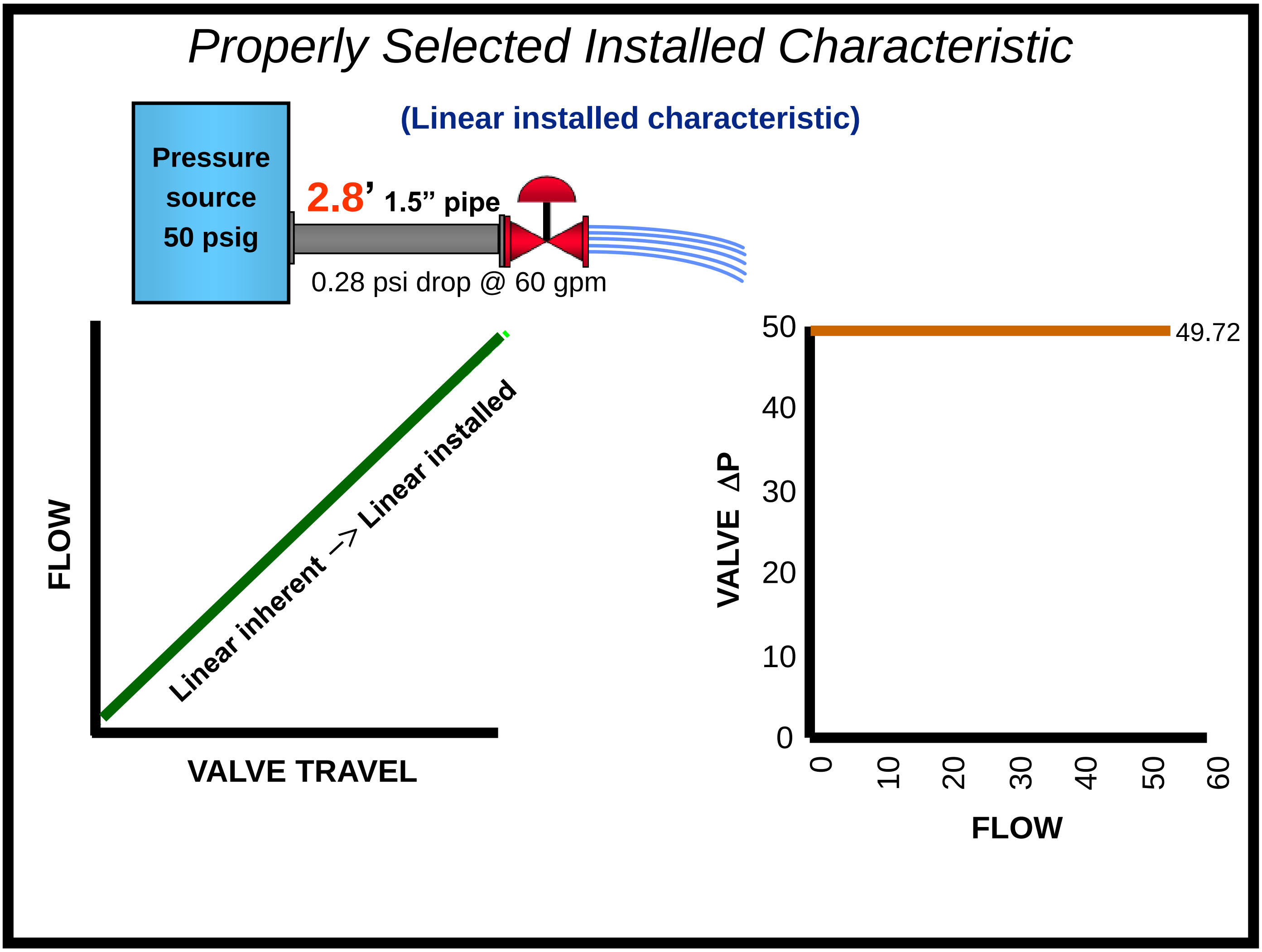
Just as the linear inherent valve retained its characteristic in the system with very little pipe, so does the equal percentage inherent valve. This results in a nonlinear equal percentage installed characteristic, which as we will see on the next page, is undesirable for most situations.



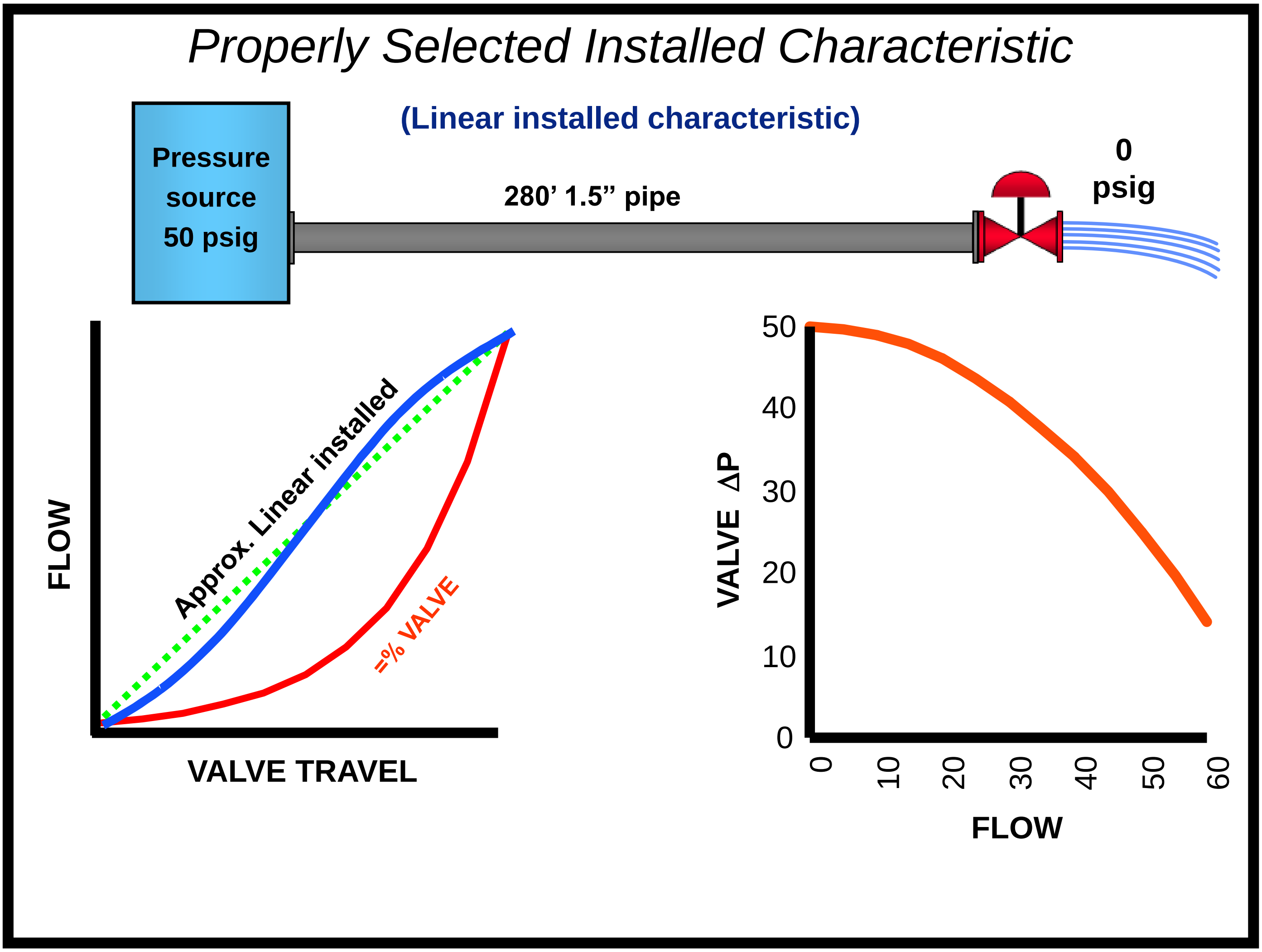
When a valve with the wrong inherent characteristic is put in a system, its installed characteristic will be non-linear as shown in the small graph. If the loop were tuned when the system was running at a control valve opening of 25%, and where the flow rate through the valve is fairly insensitive to changes in controller output (and valve travel), a high value of proportional gain would be required to get good response. When system demand increases to where the control valve is 50% open, the valve is more sensitive to changes in controller output and as a result the tuning parameters selected at a valve opening of 25% are too aggressive and a step change in set point results in an unstable response. When the system demand results in a control valve opening of 75%, the flow rate through the valve is very sensitive to changes in travel and the situation is even worse and a set point change results in extremely oscillatory response.

If instead, the loop were tuned when the control valve was 75% open, a lower value of proportional gain would have been used, and we would get fast stable response to a step change in set point, but if we then operated at lower loads, the response would be very sluggish.

If we had misapplied a linear valve in a system with a lot of pipe the situation would be the opposite of what is shown here. Just as the system with a lot of pipe pushes the equal percentage inherent characteristic upward into a linear installed characteristic, it would push a linear inherent characteristic upward into a quick opening characteristic. Now we would have a system that would be very sensitive at low valve openings and very insensitive at large openings. The system would still be difficult or impossible to tune so as to get fast stable response throughout the flow range.

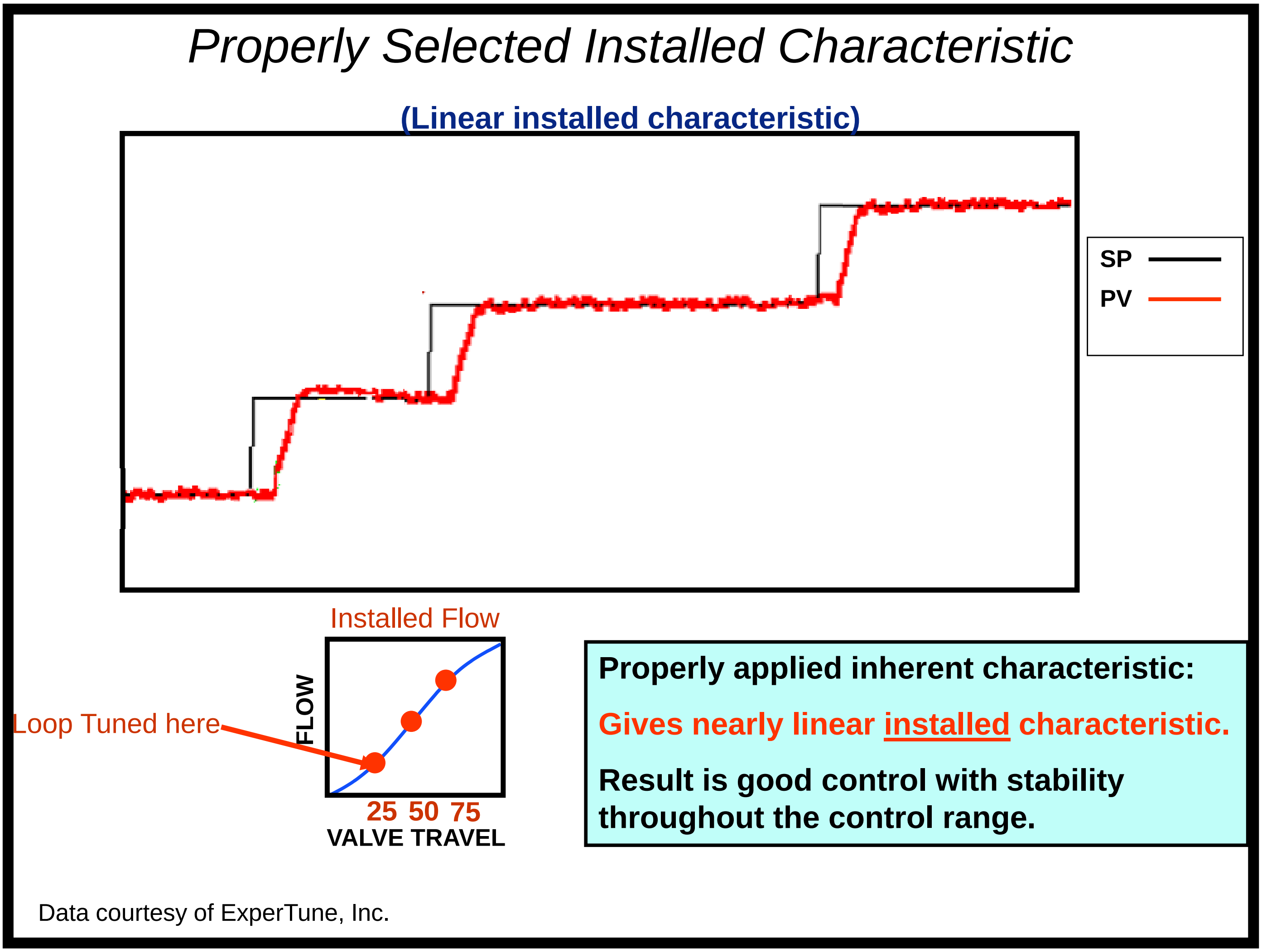


This is the page we saw a moment ago, showing a properly applied valve, specifically a linear valve in a system with only a small amount of pipe resulting in a **linear installed characteristic**.



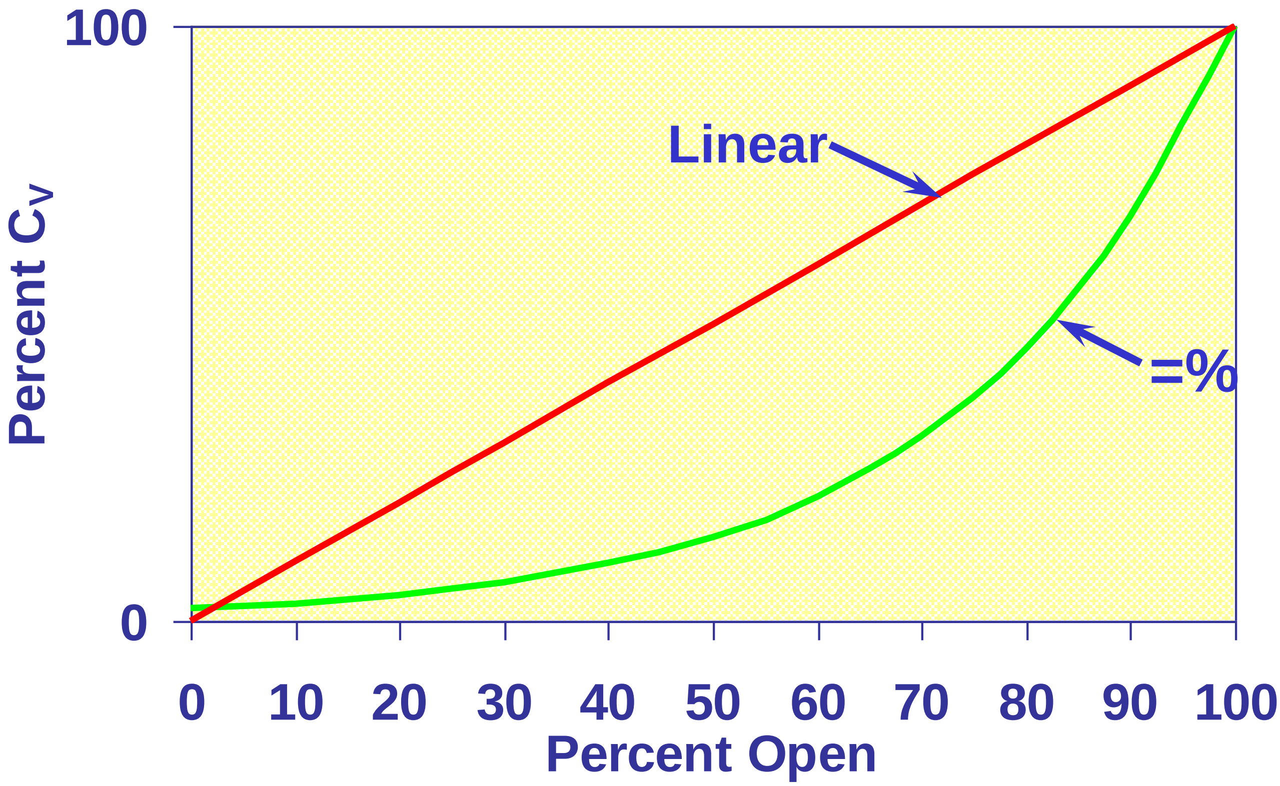
This is the figure we just saw when we discussed how the combination of an equal percentage valve and a system with a lot of pipe results in an **installed characteristic that is nearly linear**.

We just saw why a non-linear installed characteristic is bad. Next we will see why a **linear installed characteristic is good**.

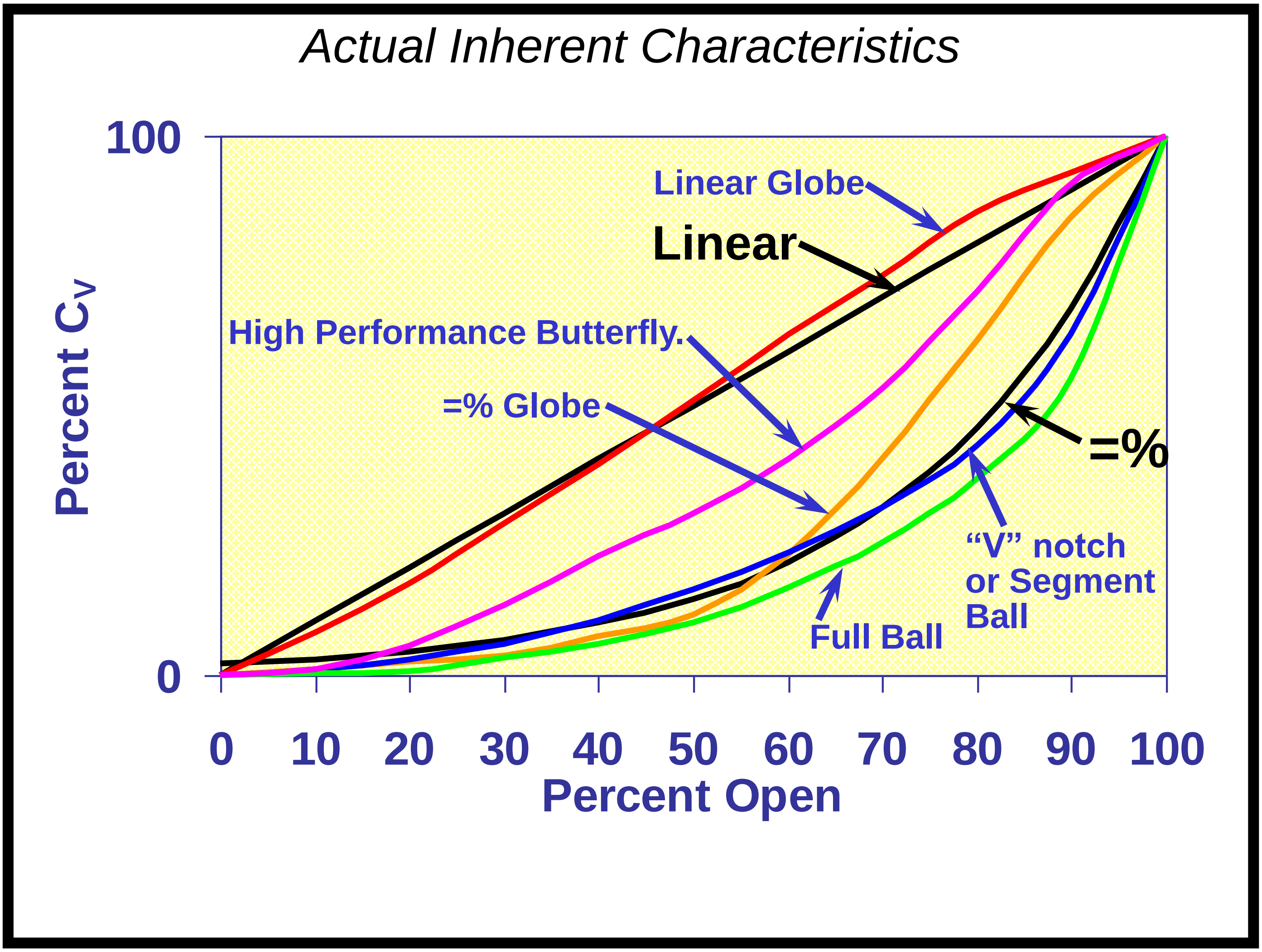


Here we have a linear installed characteristic (shown in the small graph at the bottom of the figure). Since the sensitivity of this system to changes in valve position remains constant, the same set of controller tuning parameters will give fast response with stability throughout the control range.

Ideal Inherent Characteristics



These are the “ideal” inherent characteristics we discussed earlier.



Let's see what is actually available. This graph shows typical characteristics, plotted from actual manufacturers data.

Globe valves, the most versatile of the control valves, are available with either linear or equal percentage trim.

The ball valves, both full ball and segment ball, in practice provide very nearly an ideal equal percentage characteristic, and that is the only way they come. In fact, the typical ball and segment ball valve's characteristics comes closer to the ideal equal percentage curve that the typical equal percentage globe valve.

The high performance butterfly has a characteristic that is half way between linear and equal percentage which can sometimes be ideal for systems with a moderate amount of pipe or other pressure consuming elements.

Points to Remember

- **If a set of loop tuning parameters only works at one end of the control range and not the other, the valve's flow characteristic is most likely the wrong one**
- **If a system has a lot of pipe*, use an equal percentage valve**
- **If a system has very little pipe*, use a linear valve**

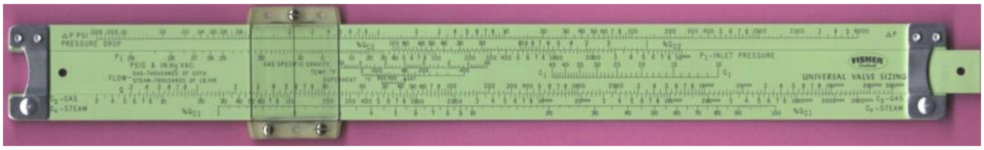
***** Or other pressure consuming elements**.**

This is a summary of the important things that we have been discussing.

Control Valve Sizing

Valve Sizing

$$Cv = Q\sqrt{\frac{G}{\Delta P}}$$



- Control Valve Sizing: The process of selecting a control valve that will do the best job of controlling the process.
- Most often by using a computer program
- Too small
  - Won't pass the required flow
- Too large
  - More expensive than necessary
  - Too sensitive
  - Difficult or impossible to adjust exactly to the required flow…

In this section we will discuss the importance of proper control valve sizing.

This slide is pretty much self explanatory. It is meant as an explanation of what we mean by control valve sizing, and why both undersized valves and oversized valves are undesirable.

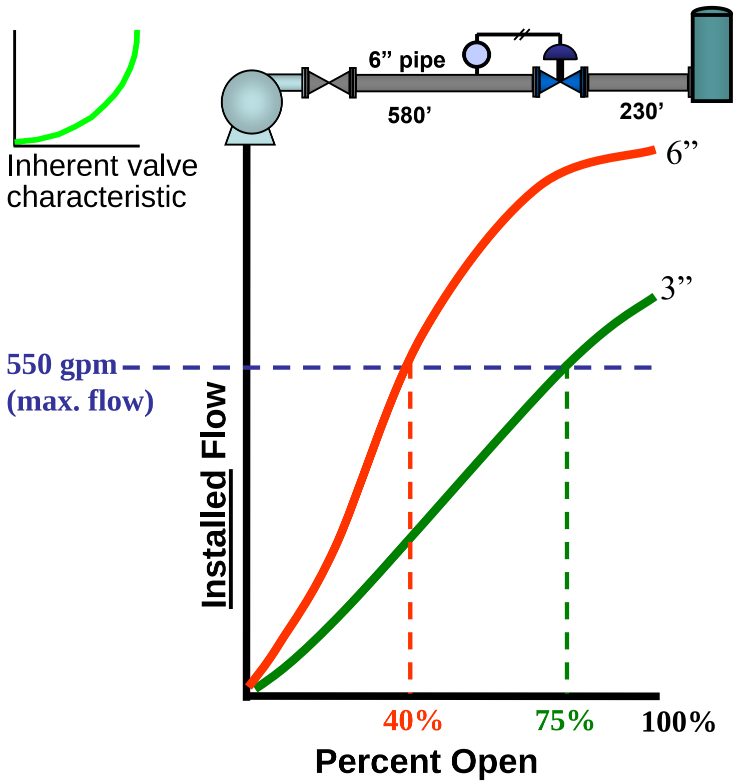
Over sizing is much more common than under sizing.

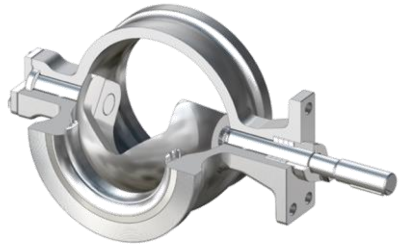
The biggest problem with an over sized valve is that it will be too sensitive, making good control difficult or impossible to achieve. This is demonstrated on the next pages.

When I started out in the control valve business we used specially designed slide rules to perform the valve sizing calculations. The one pictured was widely distributed by Fisher Controls. In fact if you look, almost every week one comes up for sale on eBay.

Now days we use computer programs. Nearly every valve manufacturer has a valve sizing program that they offer at no charge to their customers. Many are specific to the valves of that manufacturer.

Valve Sizing





Rule of Thumb

Select a control valve that operates between 60% and 80% open at the maximum required flow.

- **Uses most of the valve's control range**
- **Adequate but not excessive safety factor**
- **CONTROLS BETTER!**

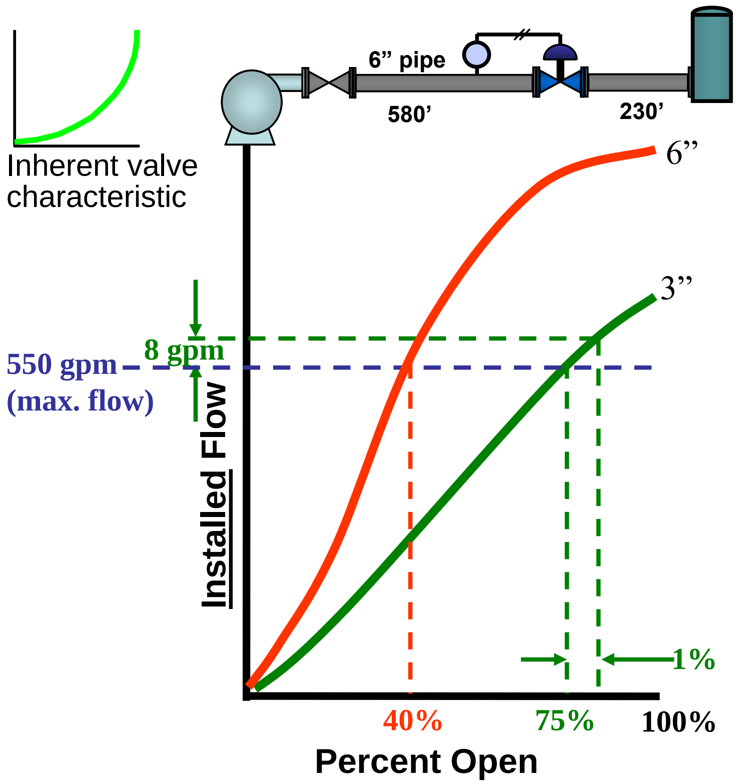
Here are graphs of the *installed* characteristics of two different valves installed in the same system. (These are both equal percent valves, and the system has a lot of pipe. Note that up to the specified maximum flow rate of 550 gpm they both have reasonably linear installed characteristic.)

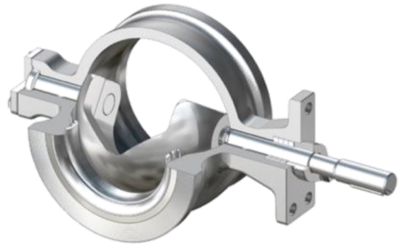
The 3 inch valve (lower curve) is a properly sized valve and the 6 inch valve (upper curve) is an oversized valve.

The reason I say that the 3 inch valve is a properly sized valve is because it meets the criteria of the rule of thumb of being between 60% and 80% open at the maximum required flow of 550 gpm. This rule of thumb has served valve users well because it gives a good balance between using as much of the valve's control range as possible, giving good flow resolution, while providing adequate safety factor. (The 3 inch valve can increase flow about 20% above 550 gpm which should be adequate, while the 6 inch valve can increase the flow about 60% above 550 gpm which is more than should ever be required.)

In the next few pages we will see how proper sizing adds up to better control.

Valve Sizing





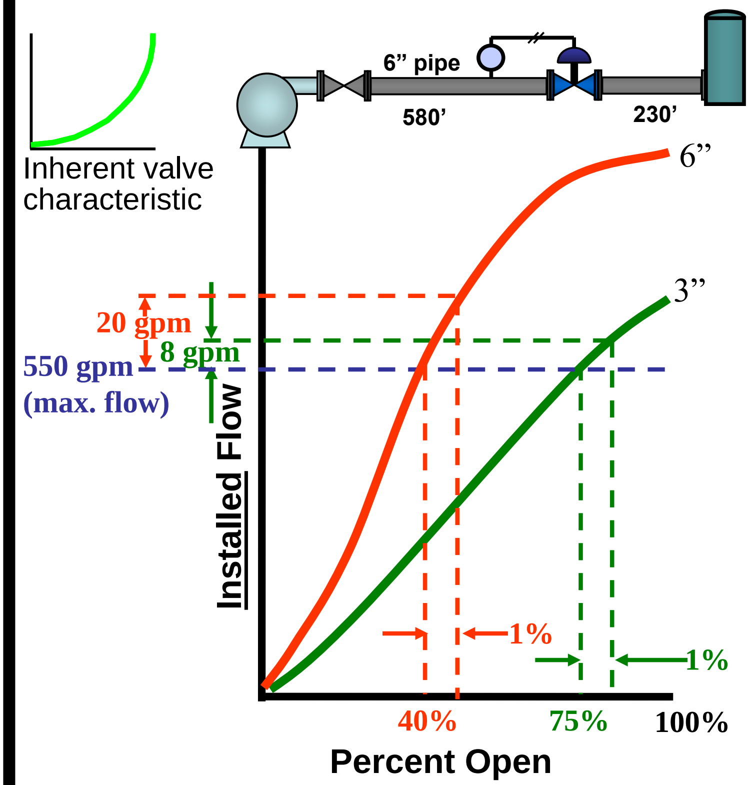
Rule of Thumb

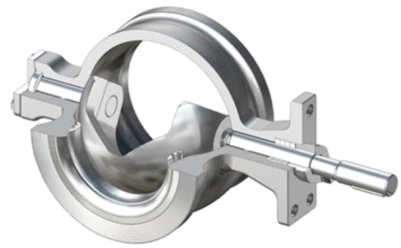
Select a control valve that operates between 60% and 80% open at the maximum required flow.

- **Uses most of the valve’s control range**
- **Adequate but not excessive safety factor**
- **CONTROLS BETTER!**

If the properly sized valve opens by one percentage point the flow increases by 8 gpm.

Valve Sizing





Rule of Thumb

Select a control valve that operates between 60% and 80% open at the maximum required flow.

- **Uses most of the valve's control range**
- **Adequate but not excessive safety factor**
- **CONTROLS BETTER!**

If the over sized valve opens by one percentage point the flow will increase by 20 gpm

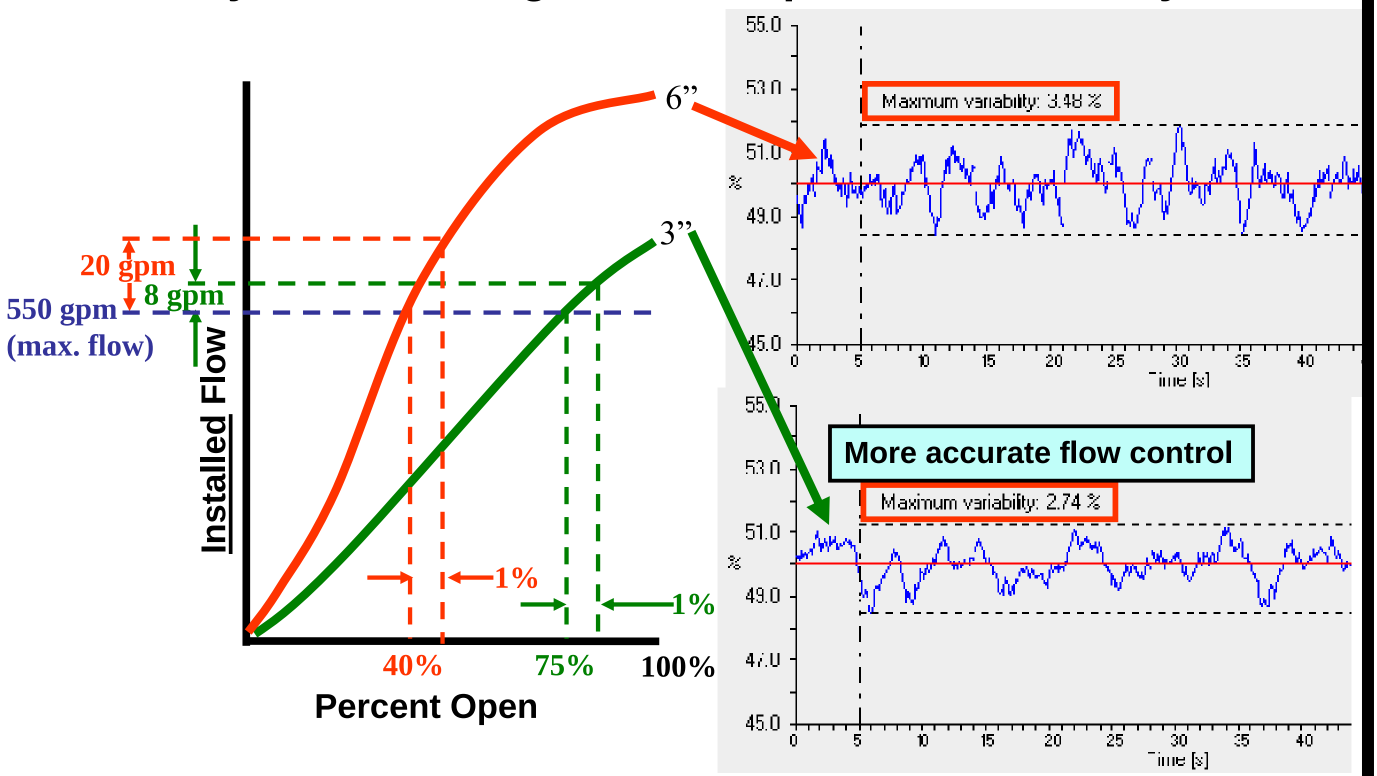
All valves exhibit a certain amount of stickiness. After a valve has been in service for a long time, especially if someone has been a little overzealous in adjusting the packing, it is not unusual to find that the smallest increment that the valve can move is 1%.

Anyway, for the sake of discussion, if the best each of these valves can do is to position themselves in 1% increments, the 3 inch valve will be able to control flow within 8 gpm increments and the 6 inch valve will only be able to control flow within 20 gpm increments.

In general, the more oversized a control valve is, the poorer the accuracy of control will be.

Valve Sizing

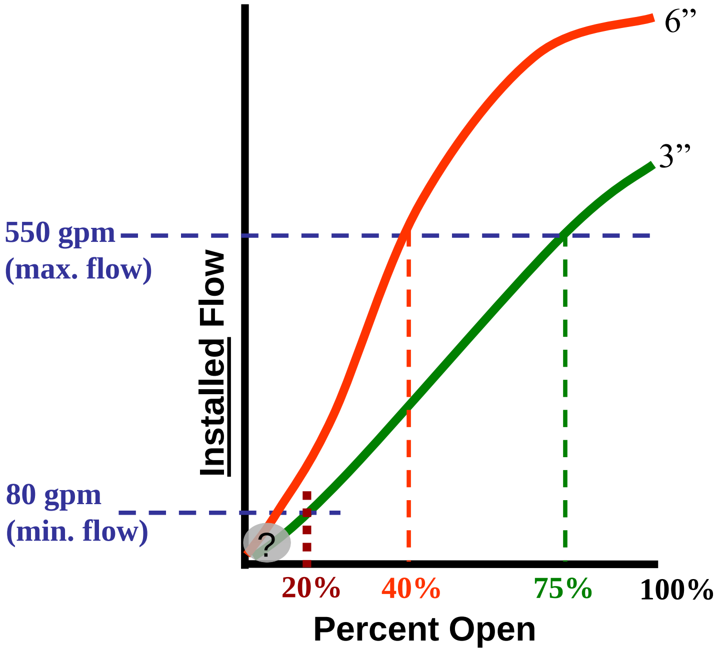
Correctly sized valve gives lower process variability



The process variability is lower with the properly sized valve.

Assuming both valves have the same amount of stickiness, the properly sized valve will be able to adjust the flow in smaller increments and therefore be able to adjust it more accurately to the required flow.

Valve Sizing



At the minimum required flow the valve shouldn’t be much less than 20% open.

- **Adequate safety factor at the low end**
- **Avoid control accuracy problems..**

It is preferable to have the minimum opening about 20% to provide some safety factor at the low end, but this isn’t always possible.

Normally, control valve manufacturers publish Cv values in 10% increments beginning at 10% open, so you have no idea what happens to the Cv below that point.

Some manufacturers of rotary control valves publish Cv data in 10 degree increments, starting at 10 degrees (approximately 11%). Check to see if between 10 and 20 degrees the Cv increases more than around 200%. If it does, that means that Cv is changing drastically between 10 and 20 degrees and below 20 degrees you could encounter control problems and unpredictable results.

It is not uncommon for ball valves and segment ball valves to have dead angles (the amount of rotation from fully closed to where the waterway in the ball rotates to where it is no longer covered by the seat) of 5 to 15 degrees.

When a valve manufacturer publishes Cv tables for rotary valves in 10% increments, these are usually expressed as a percentage of the active rotation of the ball. That is, for a ball or segment valve with a dead angle of 10 degrees, the Cv published at 10% is really the Cv at 10% of the 80 degree active range of rotation, which in this case would be 18 degrees.

Valve Sizing Procedure

Selecting a properly sized control valve is essential to achieving good process control. Now days the control valve sizing calculations are usually performed using a computer program. Most manufacturers of control valves offer control valve sizing software at no cost, though most are specific to that manufacture's valves only. One program, Neles' Nelprof, includes a number of generic valves to choose from. A detailed set of instructions for using the Nelprof program is available at no cost from: www.control-valve-application-tools.com

There is a set of comprehensive Excel spreadsheets that follow the methods of ANSI/ISA-75.01.01 (IEC 60534-2-1 Mod)-2007 *Flow Equations for Sizing Control Valves* that are available at no cost. These spreadsheets are applicable to the valves of all manufacturers and can be downloaded from the same web site that is mentioned above. The file download includes detailed instructions for using these sheets and also a detailed explanation of exactly how these sheets work.

If you have a preferred control valve supplier, the advantage of using their sizing software is that most of them have built-in tables of valve parameters (C_V, F_L, X_T) which are automatically incorporated into the calculations.

Selection of Control Valve Style

The choice of control valve style (globe, ball, segment ball, butterfly, etc.) is often based on tradition or plant preference. For example, a majority of the control valves in pulp and paper mills are usually ball or segmented ball valves. Petroleum refineries traditionally use a high percentage of globe valves. The section on "Introduction to Control Valves" includes information about some of the things that should be considered when selecting the most common control valve types.

Flow Characteristic

The section on "Control Valve Flow Characteristics" discusses the importance of selecting a valve with the correct flow characteristic.

As a general rule, systems with a significant amount of pipe and fittings (the most common case) are best suited to equal percentage inherent characteristic valves. Systems with very little pipe (where the pressure drop available to the control valve remains constant and as a result the inherent characteristic of the valve is also the installed characteristic) are better suited to linear inherent characteristic valves. An analysis of control valve installed gain, available in the Nelprof program and the Excel sheets mentioned above, can go a long way toward selecting a flow characteristic that matches a particular process.

Pipe Reducers

Nearly all control valve installations involve a control valve that is smaller than the piping into which it is installed. To accommodate the smaller valve, it is necessary to attach pipe reducers. Because the control valve size is usually not known at the time the pressure drop available to the control valve is being calculated, it is common practice to ***not*** include the reducers in the piping pressure loss calculations. Instead, the pressure loss in the reducers is handled as part of the valve sizing process by the inclusion of a “Piping Geometry Factor,” F_P. All of the modern computer programs for control valve sizing, including the Excel spreadsheets mentioned above, include the F_P calculation.

Process Data

It goes without saying that a valve sizing calculation will only be reliable if the process data used in the calculation accurately represents the true process. Sizing calculations should be performed at both the minimum and maximum design flow rates. Many people also perform a calculation at the anticipated normal flow rate. In the author’s experience there are two areas where unreliable data enters the picture. The first involves the addition of safety factors to the design flow rate and the second involves the selection of the sizing pressure drop, ΔP.

There is nothing wrong with judiciously applying a safety factor to the design flow. The problem arises when several people are involved in the design of a system and each adds a safety factor without realizing that the others have done the same.

Perhaps the most misunderstood area of control valve sizing is the selection of the pressure drop, ΔP, to use in the sizing calculation. The ΔP cannot be arbitrarily specified without regard for the actual system into which the valve will be installed. What must be kept in mind is that all of the components of the system except for the control valve (pipe, fittings, isolation valves, process equipment, etc.) are fixed and at the flow rate required by the system, the pressure loss in each of these fixed elements is also fixed. Only the control valve is variable and it is connected to a control system. The control system will adjust the control valve to whatever position is necessary to establish the required flow At this point the portion of the overall system pressure differential (the difference between the pressure at the beginning of the system and at the end of the system) that is not being consumed by the fixed elements ***must*** appear across the control valve. The correct procedure for determining the sizing pressure drop in a system that is not running is: Start at a point upstream of the valve where the pressure is known, then at the design flow rate, subtract the system pressure losses until you reach the valve inlet, at which point you have determined P1. Then go downstream until you find another point where you know the pressure, then at the design flow rate work backward (upstream) adding (you add because you are moving upstream against the flow) the system pressure losses until you reach the valve outlet at which point you have determined P2. You can now subtract P2 from P1 to obtain ΔP. This analysis should be performed at each flow rate for which a sizing calculation will be performed. If you have a say in selecting the pump (or other pressure source) the section on “Control Valve Installed Gain” describes a method for using an installed gain calculation for selecting a pressure source that will provide sufficient, but not excessive, pressure differential for the control valve to control accurately.

Points to Remember

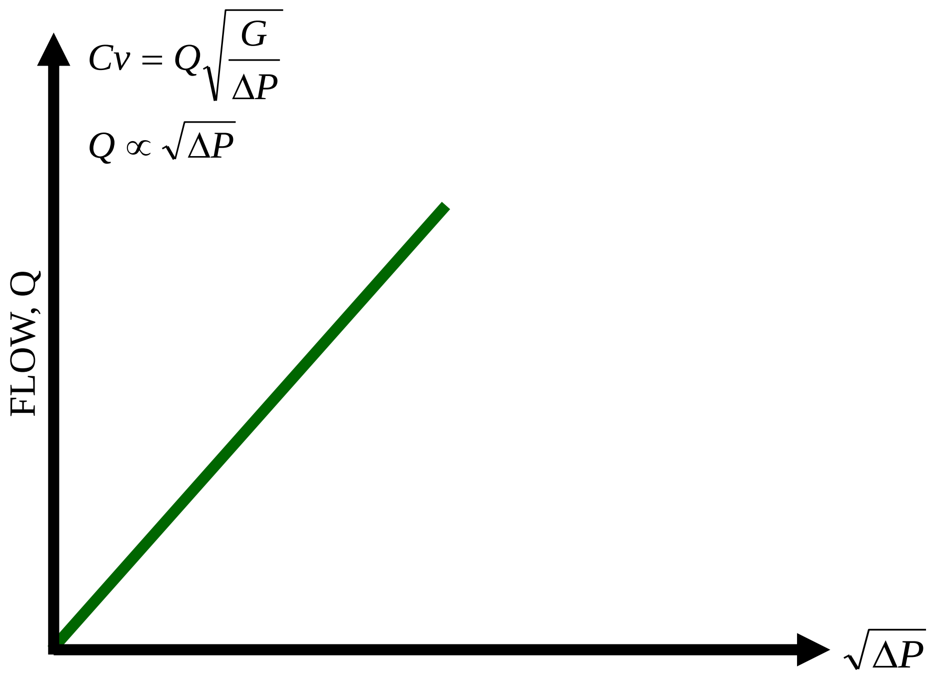
- **A control valve that is sized to operate around 60% to 80% open at the maximum required flow will give the best control**
  - **Properly sized full ball, segment ball ("V" notch) and high performance butterfly valves are usually two sizes smaller than the line**
  - **Properly sized globe valves are usually one size smaller than the line**
- **To avoid potential control problems at the low end of the control range the valve opening should not be much less than 20% whenever possible**
- **Oversized control valves are very common**
- **The pressure drop (ΔP) used in sizing calculations cannot be arbitrarily specified**
  - **Must be based on a static analysis of the system at each design flow rate**

This is a summary of the important things that we have been discussing.

The statements about the sizes of properly sized valves are not rules, but simply how things often turn out. If your calculations are different that these you might want to check your work. You may have made a mistake or the piping designer may have made a mistake.

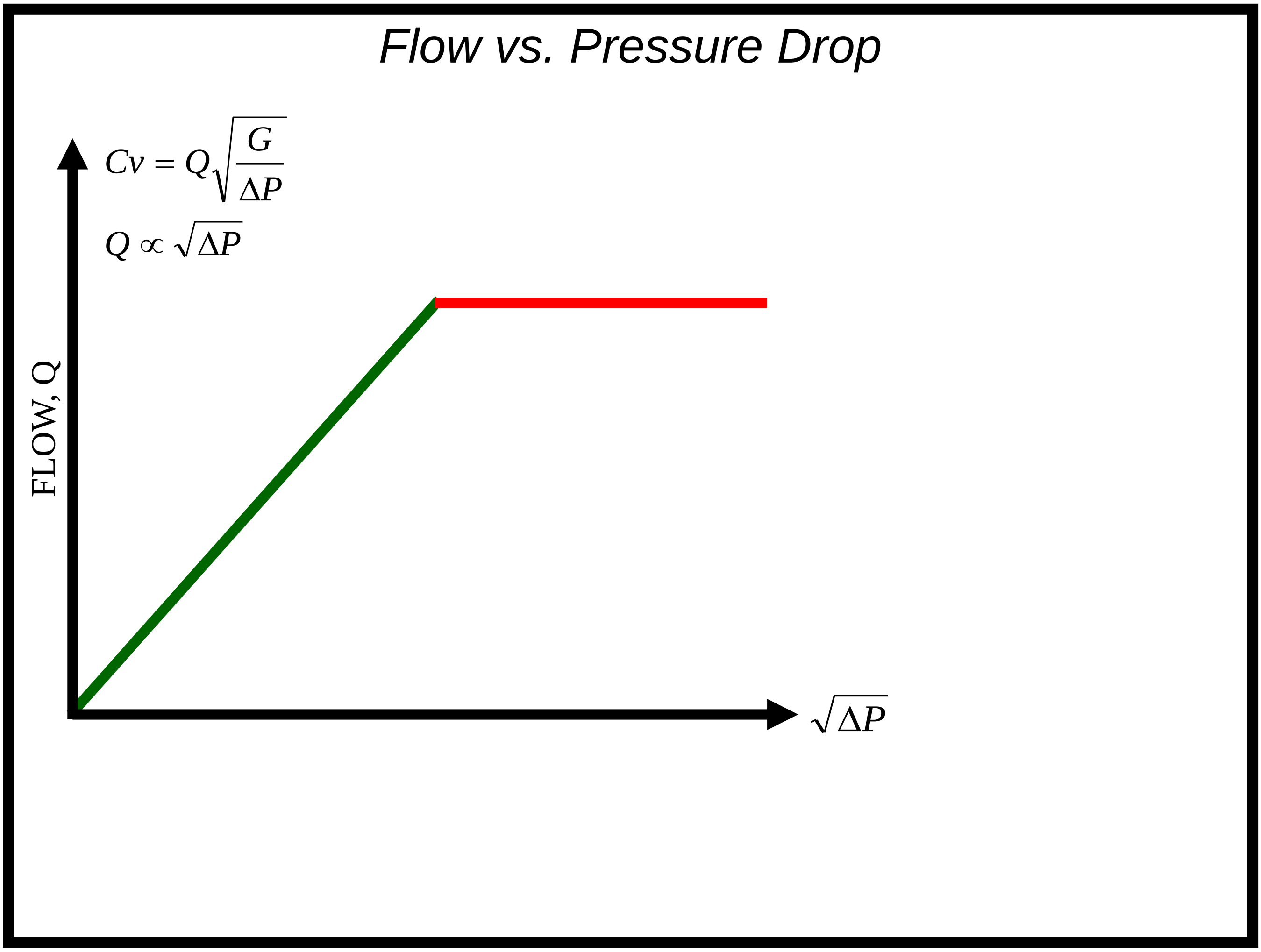
Incompressible Flow

Flow vs. Pressure Drop

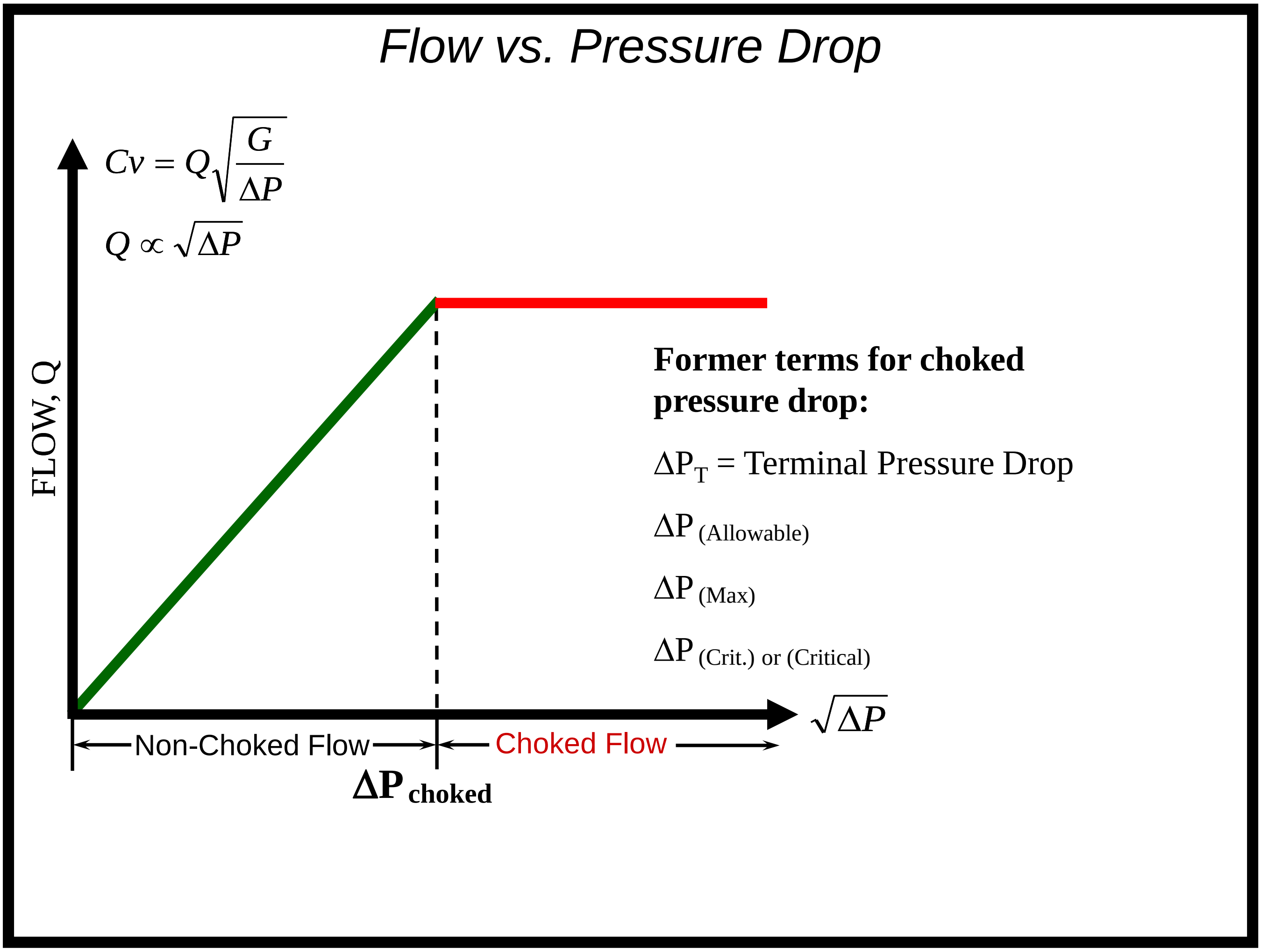


The basic liquid sizing equation tells us that the liquid flow rate through a control valve is proportional to the square root of pressure drop. This simple relationship is shown graphically.

Note that the units of the horizontal axis is the square root of pressure drop.



This linear relationship does not always hold true. As the pressure drop is increased the flow first reaches a point where increasing pressure drop only produces a small increase in flow, and then very soon after that, no matter how much the pressure drop is increased, the flow does not increase at all.

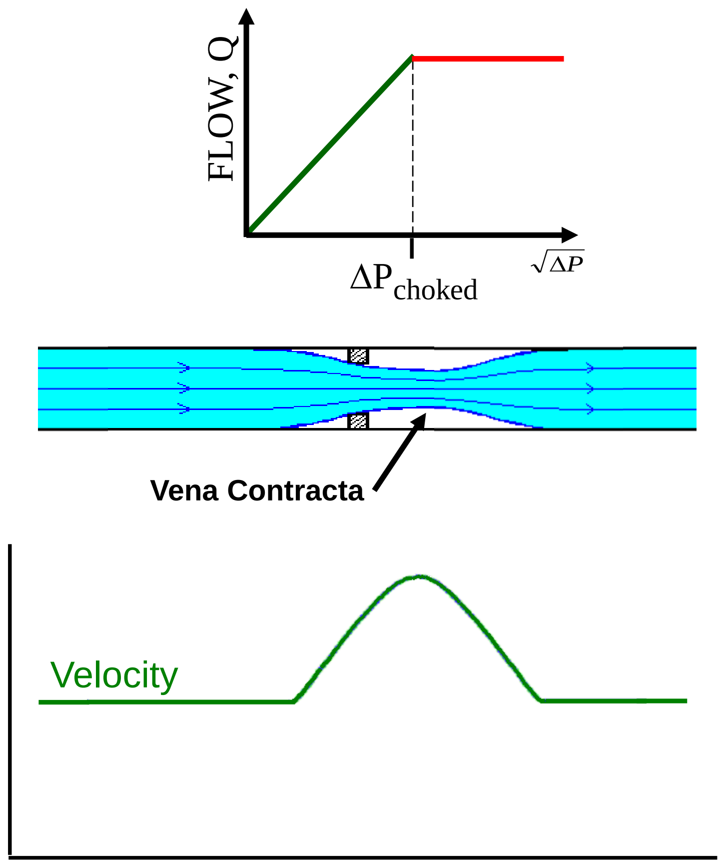


When this happens, we say that the flow has become choked. Prior to the issuance of the 2011 edition of the IEC valve sizing equation standard and the 2012 version of the ISA valve sizing equation standard there was no official name given to the dividing line between non-choked flow and choked flow, so valve manufacturers made up their own names. Some of the most common ones are listed in the figure.

In the 2011 version of the IEC control valve sizing equations and the 2012 version of the ISA S75.01.01 Standard have given it a name "Δp_{choked}." However, it may be some time before everyone updates their literature to agree with the Standards.

Throughout the rest of this discussion on liquid flow in control valves we will use the terminology "Δp_{choked}" so as to be in agreement the latest versions of the IEC and ISA control valve sizing equation standards.

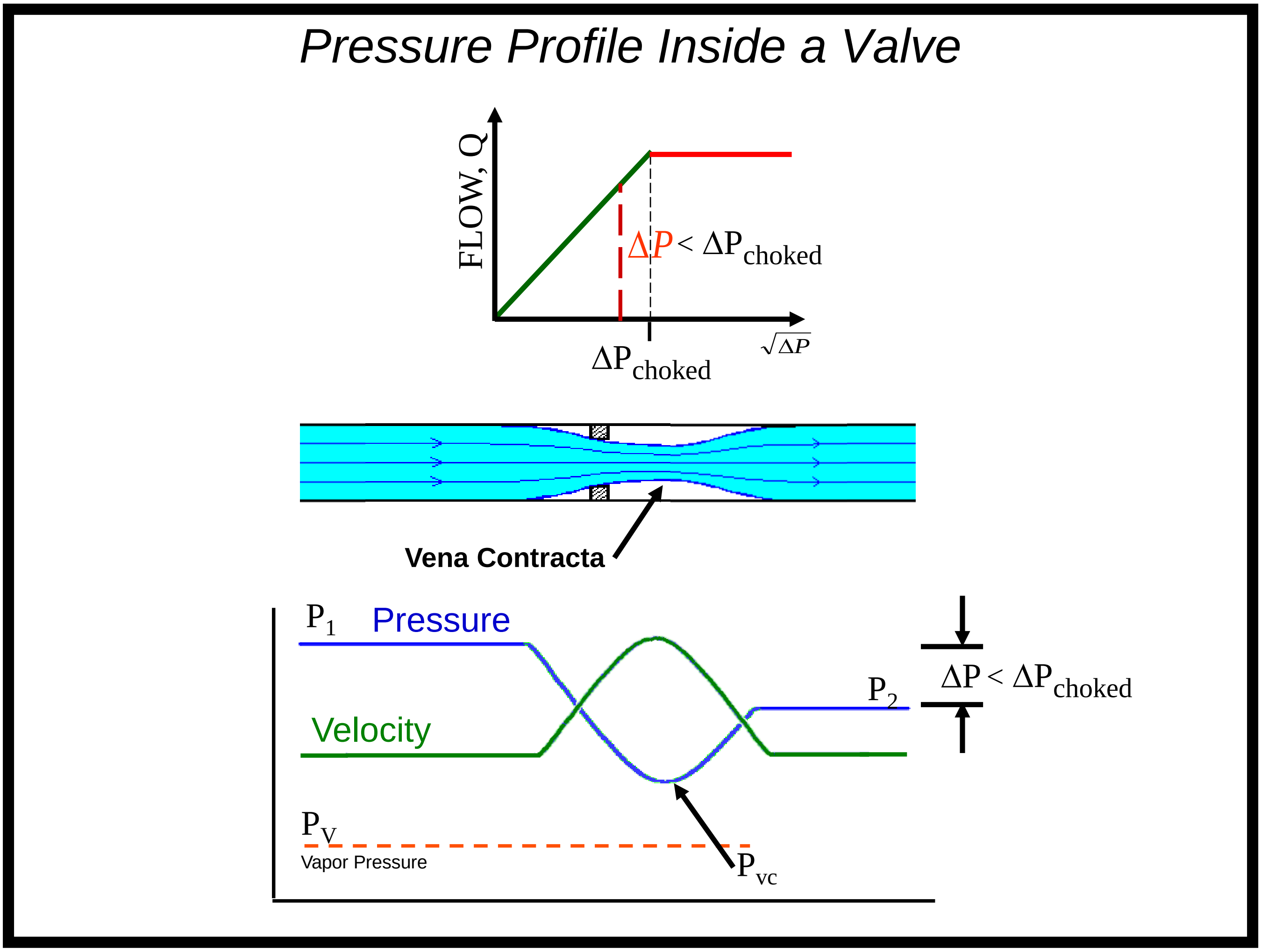
Velocity Profile Inside a Valve



Let’s take a look at what is happening inside the valve to cause this choking of the flow. In order for all of the liquid that is entering the valve to pass through the restriction, the velocity must increase at the vena contracta.

Because the flow emerging from the restriction cannot change direction immediately, the stream continues to contract to a minimum area just beyond the restriction. This point of minimum flow area is called the “Vena Contracta.”

Because of the law of conservation of energy, when the velocity increases (increased kinetic energy)…. CONTINUED ON NEXT PAGE



CONTINUED FROM PREVIOUS PAGE...the pressure (potential energy contained in the static pressure) must decrease correspondingly. As the flow continues downstream the area opens up, the velocity decreases and some of the kinetic energy is converted back into potential (pressure) energy and the pressure increases.

Notice that there is some permanent pressure loss. This is because as a result of the throttling process there is a lot of turbulence, and the friction of the liquid molecules rubbing on each other turns some of the energy into heat energy, leaving less energy to be converted back into pressure energy.

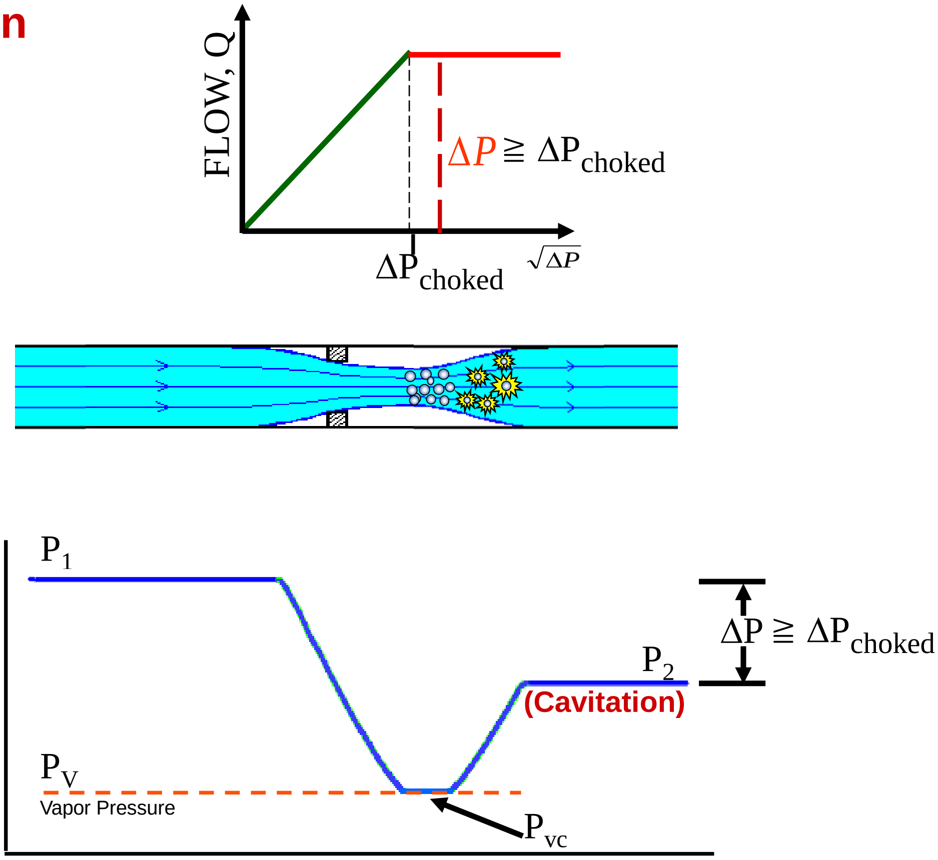
At this point we need to introduce the concept of vapor pressure. All liquids have a tendency to become a vapor. Because vapor has a much greater volume than the liquid, in order for a liquid to vaporize, it must displace the surrounding liquid and therefore the pressure which the liquid exerts on its surroundings in its attempt to expand into a vapor must equal the surrounding pressure. the pressure the liquid exerts on the surroundings in an attempt to vaporize is called the vapor pressure. Vapor pressure increases with increasing temperature.

The best known example is the vapor pressure of water at 212 degrees F, which is 14.7 psia. As we heat water to 212 degrees F its vapor pressure increases until at 212 deg. it is equal to the atmospheric pressure (assuming a location at sea level) and there is sufficient pressure to allow the water to push the earth atmosphere aside and to form vapor bubbles that float to the top and we say the water is boiling.

Water at 80 deg. F has a vapor pressure of 1/2 psia. Therefore a handful of 80 deg water does not have sufficient vapor pressure to overcome the atmospheric pressure and it remains as water. However, if we were to take the handful of 80 deg water up to the upper atmosphere, where the atmospheric pressure is only 1/2 psia, the handful of water would boil.

Pressure Profile Inside a Valve

Cavitation



It is possible to have 80 degree water (vapor pressure = 1/2 psia) entering a valve and for the velocity at the vena contracta to be high enough that the pressure at the vena contracta drops to 1/2 psia (the vapor pressure of the water)

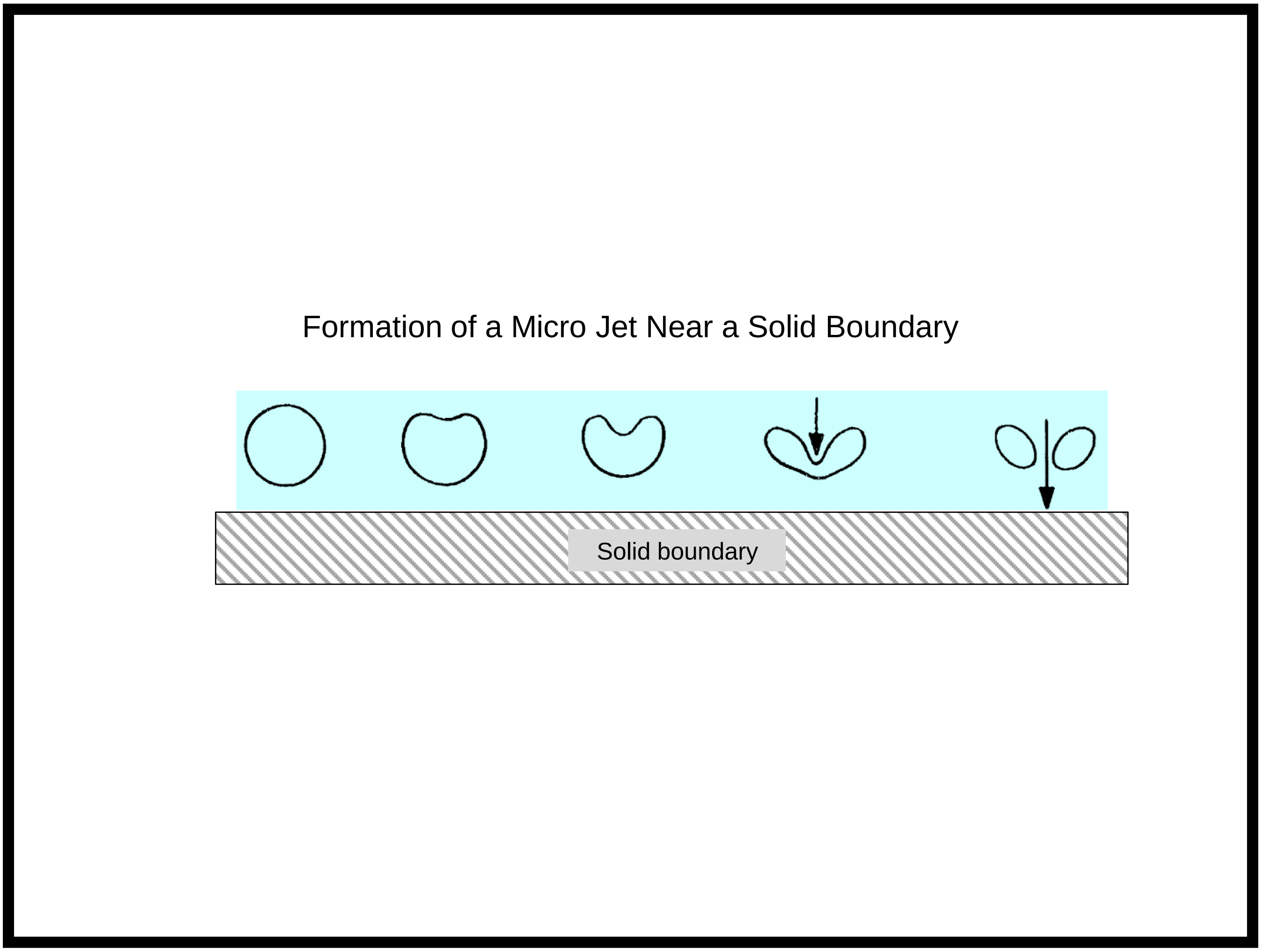
When that happens, vapor bubbles form at the vena contracta. Any additional decrease in the downstream pressure causes more bubbles to form, but the pressure at the vena contracta does not decrease below the vapor pressure. **At this point it is worth noting that the flow through a control valve depends on the pressure difference between P_1 and P_{vc} (the pressure at the vena contracta), and since the vena contracta pressure does not decrease below the vapor pressure, the flow does not increase, resulting in choked flow. (1)**

As the bubbles move down stream, the cross sectional flow area opens up, the velocity goes down and the pressure goes up. Now we have bubbles with an internal pressure of 1/2 psia surrounded by a higher pressure. The bubbles collapse in on themselves. This combination of bubble formation and the resulting choked flow, along with the collapse of the bubbles downstream is called <u>CAVITATION.</u> When the bubbles collapse they make a popping sound. The result is a noise like gravel going through the valve. This noise can be loud enough to be very annoying and even loud enough to damage the hearing of a person who is exposed to it for long periods.

Also, when the bubbles collapse, they create shock waves that can knock small pieces of metal off any nearby valve parts.

It is the violent collapse of these vapor bubbles near valve component surfaces which cause cavitation damage. In control valves, this process typically occurs near or at the exit of the valve trim.

(1) ISA-S75.01-1985 (R 1995) Annex G, p43: *"Flow rate is a function of the pressure drop from the valve inlet to the vena contracta. Under nonvaporizing liquid flow conditions, the apparent vena contracta pressure (p_{vc}) can be predicted from the downstream pressure (p_2), because the pressure recovery is a consistent fraction of the pressure drop to the vena contracta. The effect of this pressure recovery is recognized in the valve flow coefficient (C_v). Under choked flow conditions, there is no relationship between p_2 and p_{vc} because vaporization affects pressure recovery"*

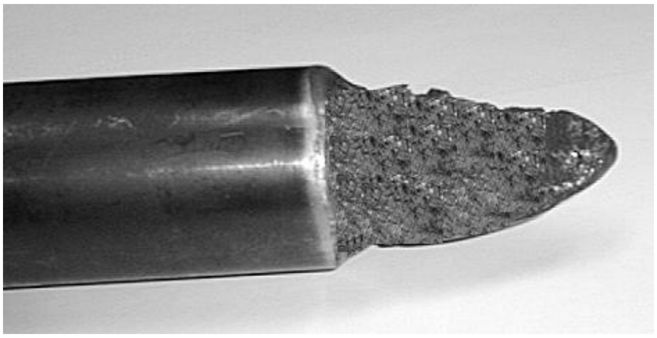


Two mechanisms have, been identified in the literature by which damage occurs at solid boundaries. One is the high pressure shock waves generated by collapse of the cavities.

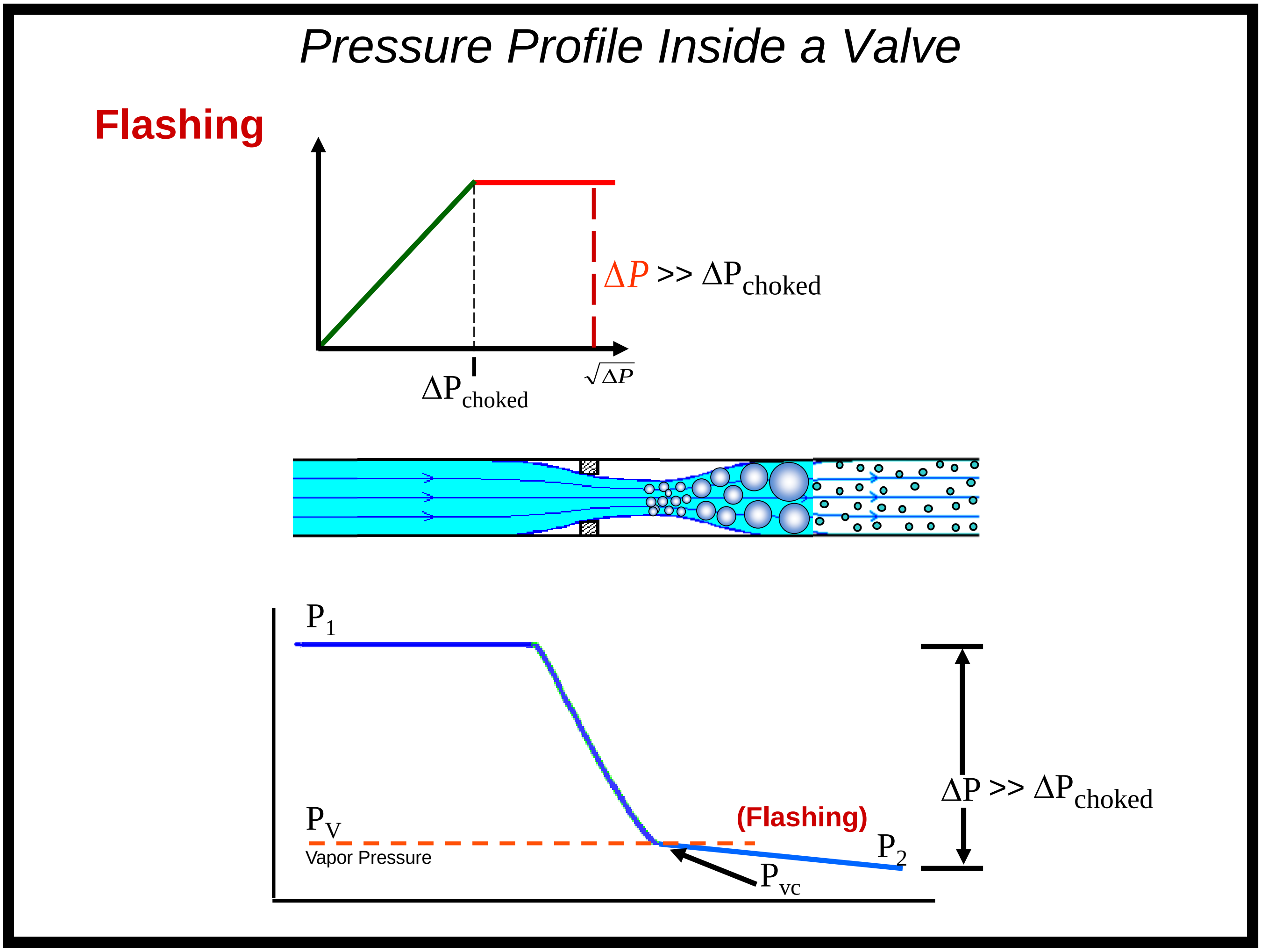
The other source of potential damage is caused by micro jets. When a bubble collapses near the boundary, the pressure distribution around the bubble is unsymmetrical due to the presence of the boundary. As the bubble collapses, the boundary causes a resistance to the flow trying to fill the void from the boundary side and allows the liquid on the side of the bubble away from the wall to attain a higher velocity. This causes the bubble to collapse faster on that side forming a jet which shoots through the center of the bubble. This jet attains high velocities and creates a local pit when it impacts the wall.

Some authorities report that after the microjet occurs, that there is a remaining cloud of bubbles left that then collapse causing a spherical shock wave that can also contribute to cavitation damage.

Cavitation Damage



The appearance of cavitation damage is a rough, cinder like, look. This damage can happen very quickly (a few weeks or months). Because cavitation damage happens so quickly, we try to avoid cavitation at all costs. Very hard materials give some improvement, but usually the improved performance is not enough to justify the cost.



If we continue to decrease the downstream pressure, we reach a point where the pressure downstream of the valve is less that the vapor pressure of the liquid. Now, instead of collapsing, the bubbles continue downstream This is called <u>FLASHING.</u>

Typically, as the flow leaves the vena contracta the bubbles get bigger until the flow changes from a liquid with bubbles to a vapor with drops of liquid in it.

The noise caused by flashing is much less that the noise caused by cavitation. In fact, there is no method for calculating flashing noise, but experience shows that it will usually be less than 85 dBA.

The preceding classical discussion of choked flow, cavitation and flashing is the discussion you will see in most texts and manufacturer's literature and is usually considered adequate for an understanding of the subjects. We will see later that this classical discussion of choked flow, in reality, requires a slight modification. Numerous test have shown that the pressure at the vena contracta must actually drop to slightly below the vapor pressure for vaporization to occur at the vena contracta and for flow to choke.

We will also see that because of this fact, there are uncommon but possible situations where there may be flashing without choked flow.

Flashing Damage

The appearance of flashing damage is quite different from cavitation damage, with mooth, shiny rivers and valleys. The damage mechanism is a sand blasting effect. Downstream of the vena contracta the flow consists of a large volume of vapor with many tiny drops of liquid. Because the volume increases greatly when liquid vaporizes, the downstream velocity can be several hundred feet per second, and the high velocity liquid droplets can erode away a valve part.

The damage caused by flashing does not usually happen as quickly as that caused by cavitation. The use of hard or erosion resistant materials can often bring the damage to within tolerable limits. Trim parts made of the hard stainless steels such as 17-4 ph hold up quite well, and 316ss or chrome moly bodies do much better than carbon steel.

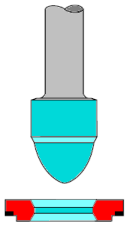
Flashing Damage Reduction Strategies

Valve Discharging Directly into Vessel

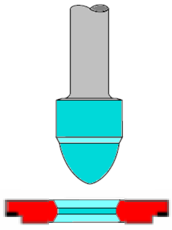
Installing the control valve so that it discharges directly into a vessel will eliminate the potential for flashing damage to the downstream piping.

Flashing Damage Reduction Strategies

Reduced trim



Small valve with full area trim.

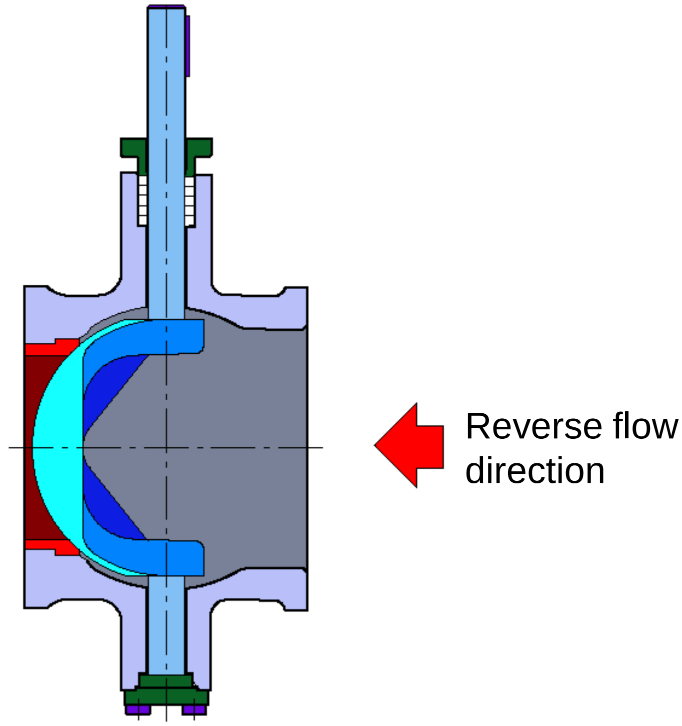


Larger valve body with same Cv as smaller valve reduces fluid velocity in valve body.

Using oversized valve bodies with reduced trim, gives the proper Cv for the application, but a large body which keeps the velocity of the flashing stream lower and less erosive to the valve body.

Flashing Damage Reduction Strategies

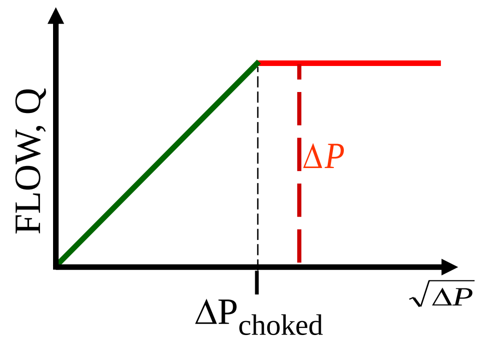
Segment Ball Valve with Erosion Resistant Trim

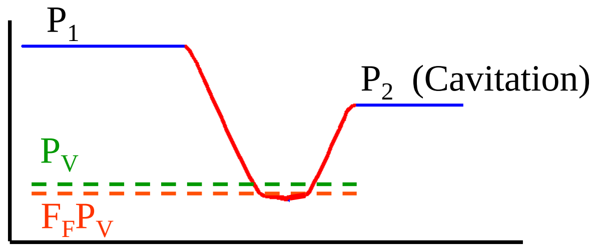


Normally, segment ball valves are configured for flow into the face of the ball.

An erosion resistant version flows in the reverse direction, and downstream of the segment is a one piece seat and body liner made of solid stellite. (A very hard, erosion resistant alloy.)

Predicting Choked Flow





$$\Delta P_{choked} = F_L^{\ 2}(P_1 - F_F P_V)$$

P_1 = Pressure upstream of valve

P_V = Vapor pressure of liquid

$$F_F = 0.96 - 0.28\sqrt{\frac{P_V}{P_C}}$$

(F_F will always be between 0.68 and 0.96)

(F_F, **Critical Pressure Ratio Factor**)

Example

For 193F water, P_V = 10 psia

F_F= 0.94

So, $F_F\ P_V$ = 0.94 X 10 = 9.4 psia

We are interested in being able to predict whether flow will be choked because choked flow results in (1) Limiting of the flow rate. (2) The potential for noise. (3) The potential for damage.

Since we have defined the choked pressure drop, ΔP_{choked}, as the dividing line between choked and non-choked flow, we need to determine the value ΔP_{choked} and then compare it with the actual pressure drop.

The **process related** factors that go into the ΔP_{choked} calculation are, P_1, the upstream pressure, and P_v, the liquid's vapor pressure.

There is also a correction factor, F_F, the CRITICAL PRESSURE RATIO FACTOR, that, for reasons of simplicity, was not mentioned in our earlier discussions of choked flow, which falls within the range of 0.96 to 0.68. Numerous choked flow tests have shown that the pressure at the vena contracta must drop below the upstream vapor pressure for vaporization to take place in the vena contracta and for flow to choke. The formula shown above for F_F gives a good approximation of how much below the upstream vapor pressure the vena contracta pressure has to drop for flow to choke. The example in the lower left of the figure shows that for 193F water, the vena contracta pressure must drop to 94% of the upstream vapor pressure, or to 9.4 psi.

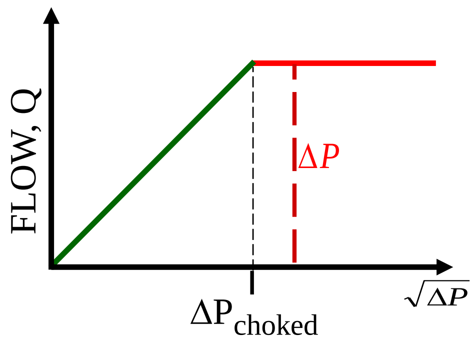
Multiplying the vapor pressure by F_F (a number smaller than 1) mathematically lowers the vapor pressure line on the right hand graph to the point that the vena contracta pressure must drop to in order for flow to choke. That is, the vena contracta pressure must drop to $F_F P_V$.

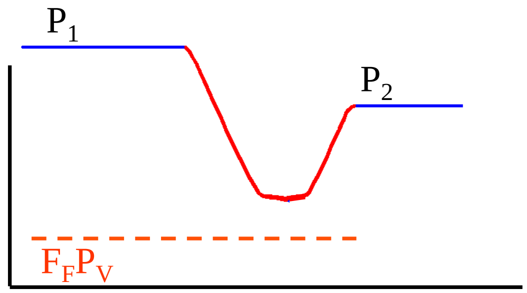
F_F, The Critical Pressure Ratio Factor is a function of the ratio of the vapor pressure to the thermodynamic critical pressure of the liquid.

(The critical pressure of a liquid is the pressure above which the liquid cannot be made to boil, regardless of temperature.)

For the case shown, if the pressure drop (P_1 - P_2) were held constant, but P_1 were increased, …. CONTINUED ON NEXT PAGE

Process Changes to Avoid Choked Flow





$$\Delta P_{choked} = F_L^{\,2}\left(P_1 - F_F P_V\right)$$

P_1 = Pressure upstream of valve

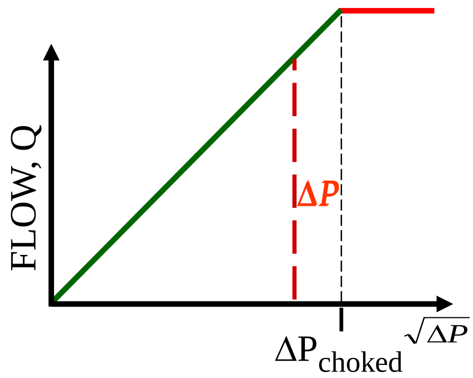
P_V = Vapor pressure of liquid

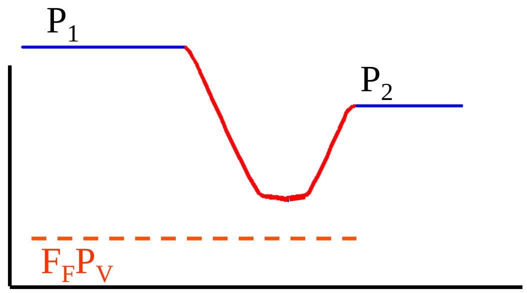
$$F_F = 0.96 - 0.28\sqrt{\frac{P_V}{P_C}}$$

(F_F, Critical Pressure Ratio Factor)

…..the entire curve would move upward until the dip no longer touched the $F_F P_V$ line and flow would no longer be choked. CONTINUED ON NEXT PAGE

Process Changes to Avoid Choked Flow





$$\Delta P_{choked} = F_L^{\,2} (P_1 - F_F P_V)$$

P_1 = Pressure upstream of valve

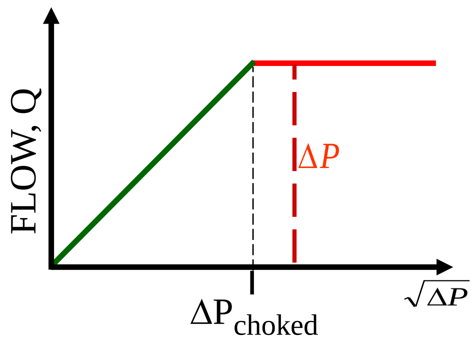
P_V = Vapor pressure of liquid

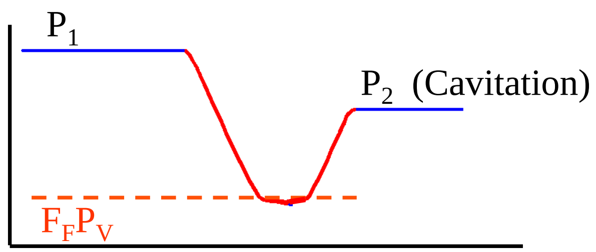
$$F_F = 0.96 - 0.28\sqrt{\frac{P_V}{P_C}}$$

(F_F, Critical Pressure Ratio Factor)

…..Looking at the delta P_{choked} equation, when P_1 increases, the terminal pressure drop also increases. This is shown graphically in the Left graph, where the terminal pressure drop is now greater than the actual pressure drop, indicating that flow is no longer choked. CONTINUED ON NEXT PAGE

Process Changes to Avoid Choked Flow





$$\Delta P_{\text{choked}} = F_L^{\,2}\,(P_1 - F_F P_V)$$

P_1 = Pressure upstream of valve

P_V = Vapor pressure of liquid

$$F_F = 0.96 - 0.28\sqrt{\frac{P_V}{P_C}}$$

(F_F, Critical Pressure Ratio Factor)

On this page, we have put the upstream pressure, P_1, back to where it was originally. If the temperature of the liquid was decreased, and thus the vapor pressure, Pv, was decreased,….
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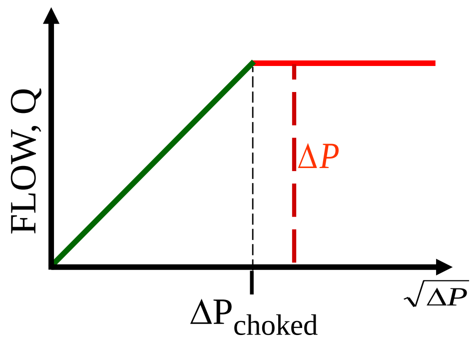
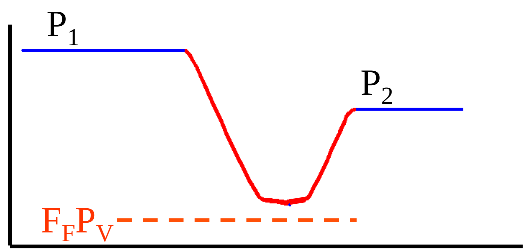
Process Changes to Avoid Choked Flow





$$\Delta P_{choked} = F_L^{\,2}\,(P_1 - F_F\,P_V)$$

P_1 = Pressure upstream of valve

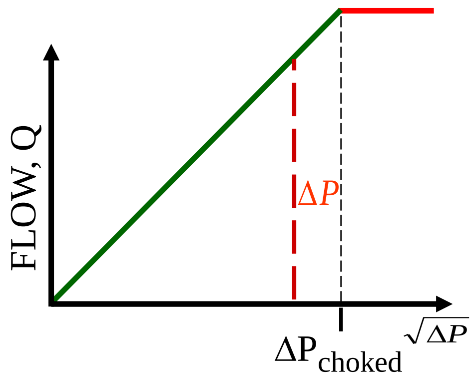
P_V = Vapor pressure of liquid

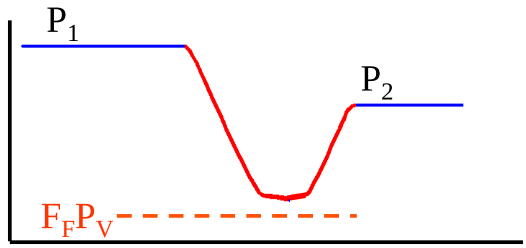
$$F_F = 0.96 - 0.28\sqrt{\frac{P_V}{P_C}}$$

(F_F, Critical Pressure Ratio Factor)

….the vapor pressure line on the graph would be lower down, and again, the pressure dip would no longer touch the $F_F P_V$ line line.

Process Changes to Avoid Choked Flow





$$\Delta P_{choked} = F_L^{\,2}(P_1 - F_F P_V)$$

P_1 = Pressure upstream of valve

P_V = Vapor pressure of liquid

$$F_F = 0.96 - 0.28\sqrt{\frac{P_V}{P_C}}$$

(F_F, Critical Pressure Ratio Factor)

Looking at the delta P_{choked} equation, when Pv decreases, the choked pressure drop increases. This is shown graphically in the Left graph, where the choked pressure drop is now greater than the actual pressure drop, indicating that flow is no longer choked.

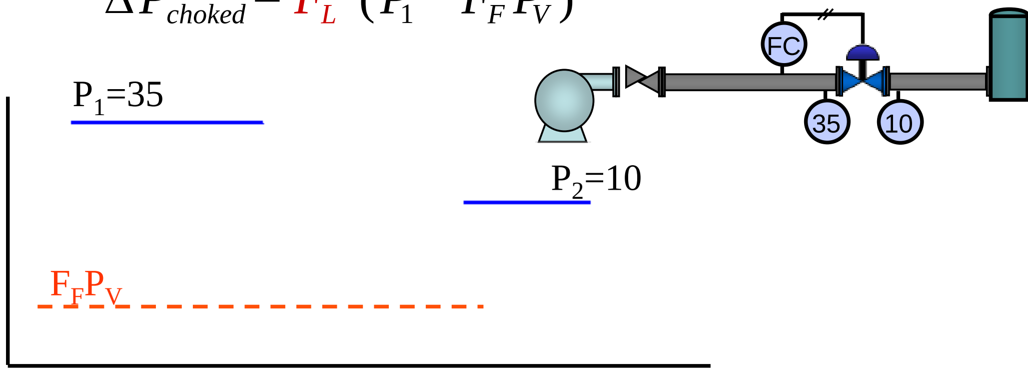
F_L, Liquid Pressure Recovery Factor

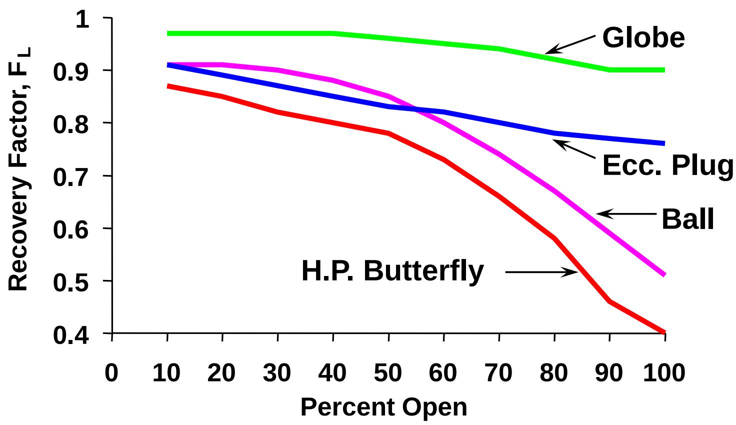
$$\Delta P_{choked} = F_L^{\,2}(P_1 - F_F P_V)$$

Q = 500 gpm

$P_1 - P_2 = 25$ psi

C_V=100





In addition to the process factors which affect the choked pressure drop, the particular valve style being considered is included in the ΔP_{choked} calculation in the form of the LIQUID PRESSURE RECOVERY FACTOR, F_L.

Let's see what would happen if we were to test two different valves, a globe valve and a butterfly valve, each installed in the same application. For each valve, the automatic control system will adjust the valve to the position necessary to produce the required flow. Both valves will then have the same flow rate and the same P_1 and P_2, so externally to the valve, both systems will behave identically.

If we were to look at the pressure profile inside the valves,…(SEE THE NEXT PAGE)

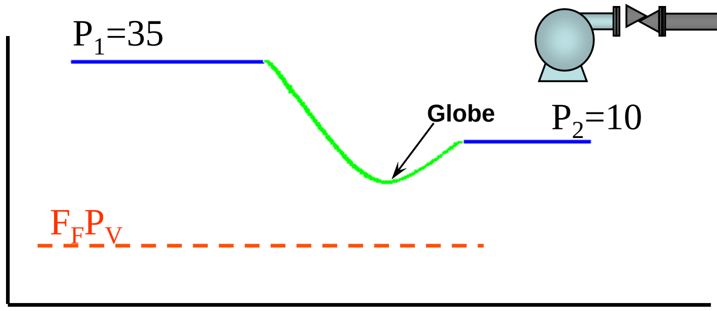
F_L, *Liquid Pressure Recovery Factor*

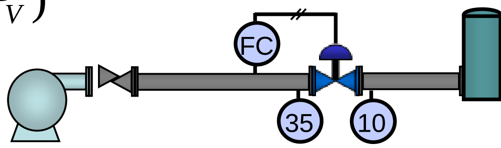
$$\Delta P_{choked} = F_L^{\,2}(P_1 - F_F P_V)$$

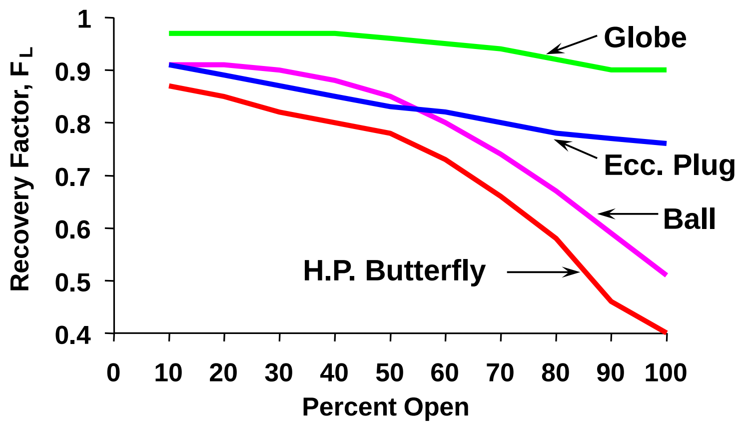
Q = 500 gpm

$P_1 - P_2 = 25$ psi

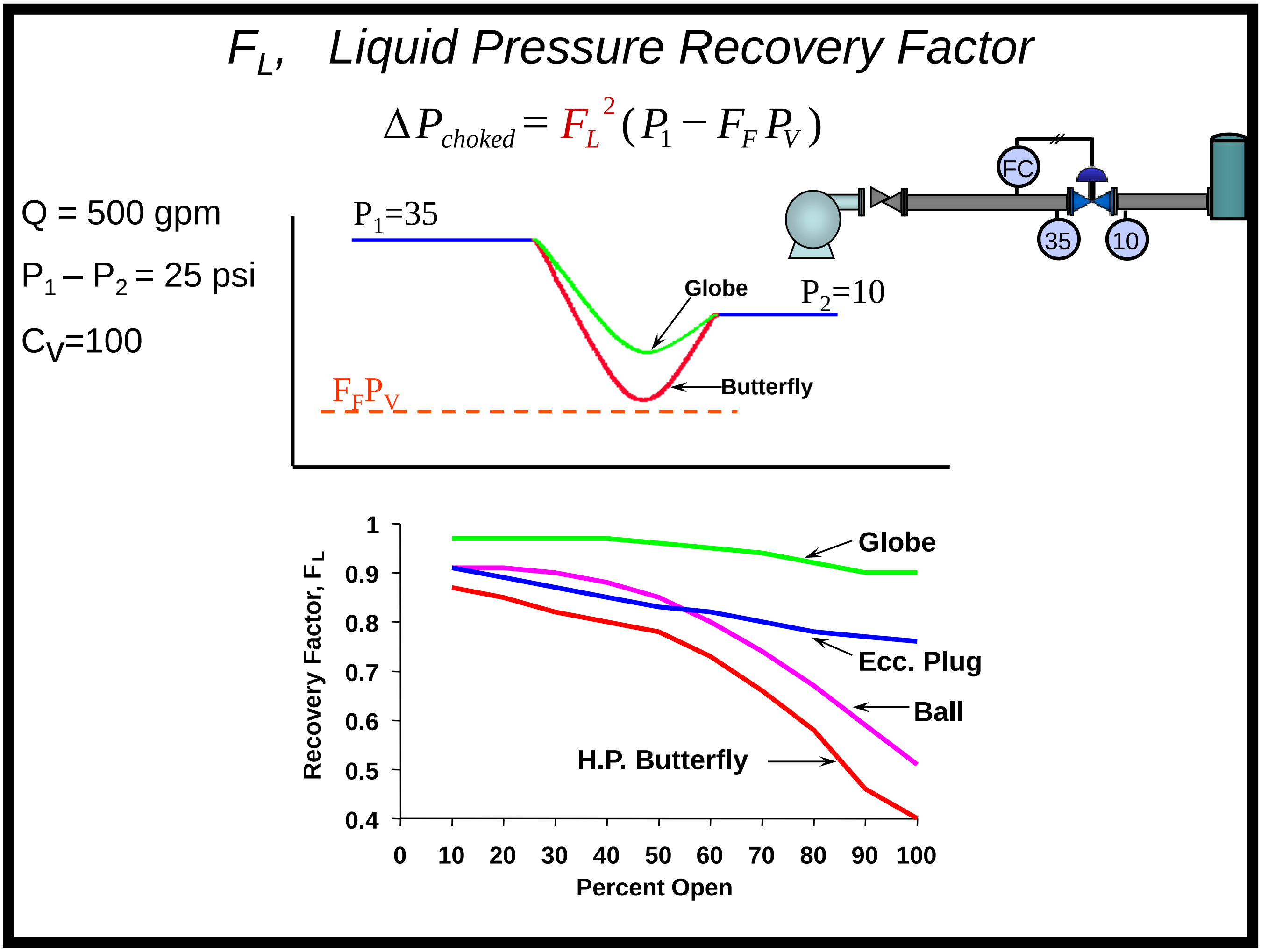
C_V=100



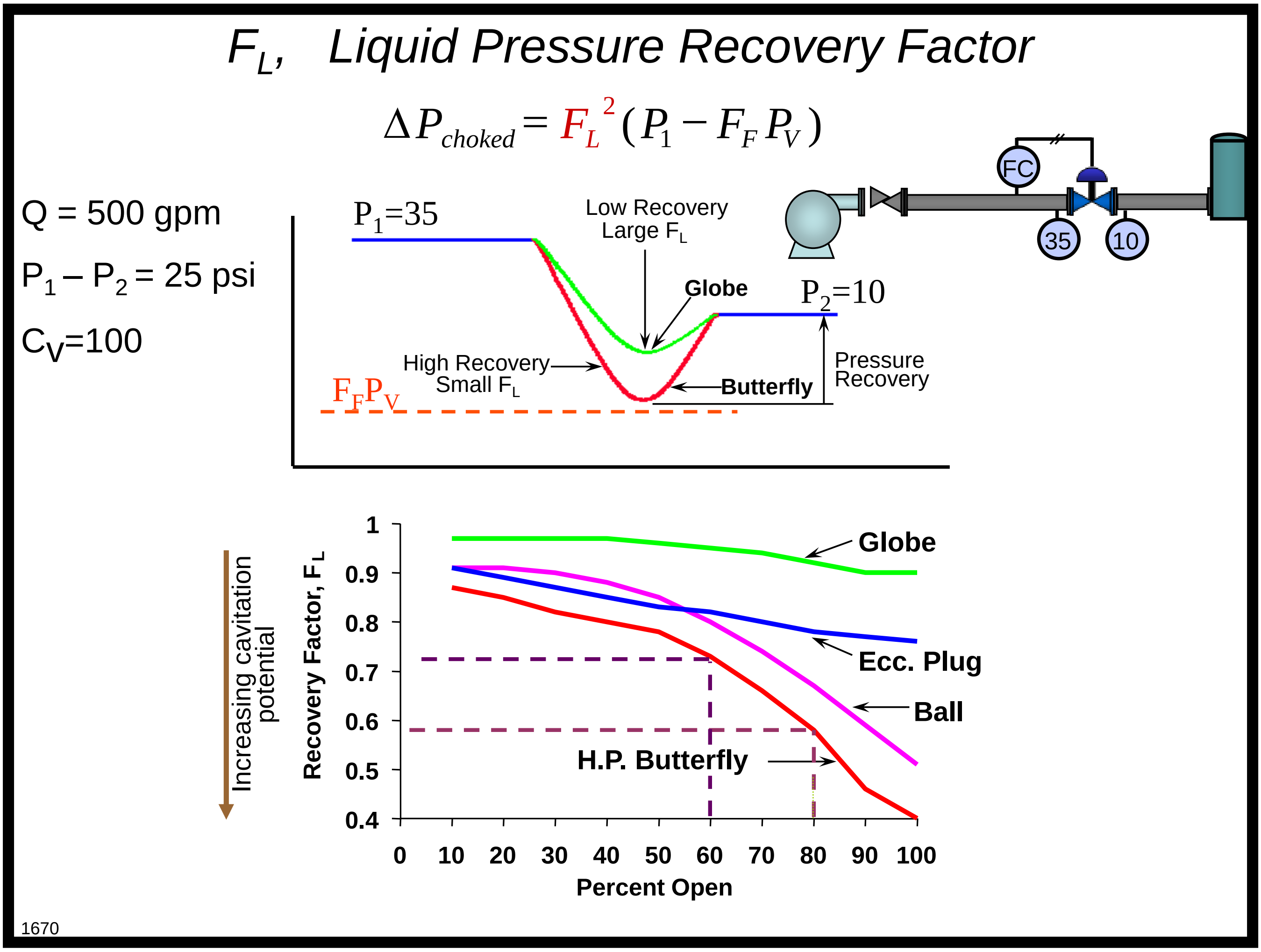




….we would see that the globe valve has a small pressure dip,… CONTINUED ON NEXT PAGE



...and the butterfly valve has a large pressure dip (corresponding to the two curves in the top graph). The valve with the large dip (typical of the high capacity valves such as ball and butterfly)… CONTINUED ON NEXT PAGE



...we call a HIGH RECOVERY valves (because a large percentage of the lost pressure at the dip is recovered. The valve with the small dip (typical of low capacity valves such as globe valves) we call a LOW RECOVERY valve. You can see that for the same application, the high recovery valves are more likely to cavitate because the pressure dip is more likely to touch the F_F P_V line.

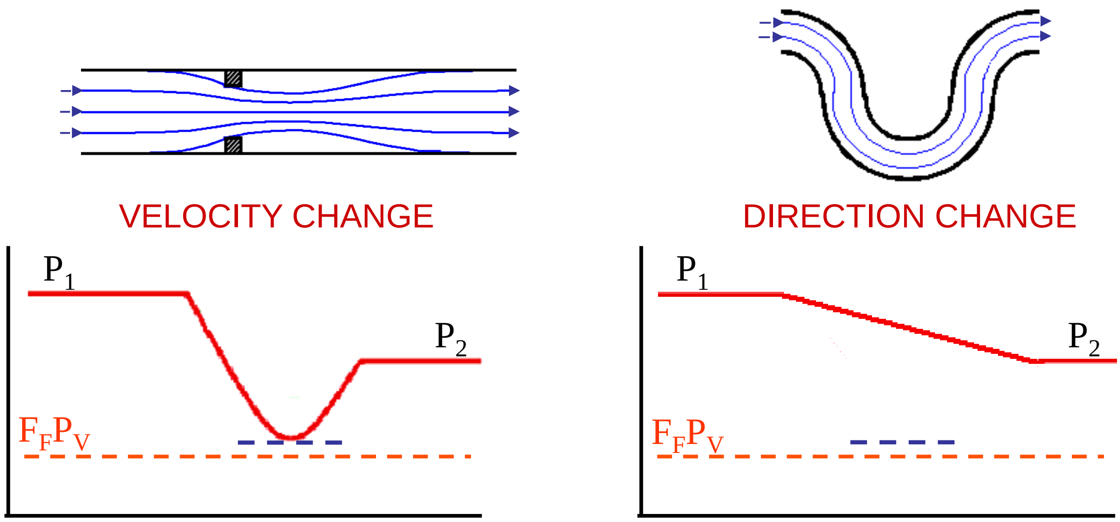
F_L values are determined for each valve type in a flow test where pressure drop is increased until flow chokes, then the ΔP_{choked} equation is solved for F_L. The lower values of F_L (0.5, 0.6, 0.7) correspond to the high recovery valves and the high values of F_L (0.85, 0.95) correspond to the low recovery valves.

From the lower graph, you can see that F_L is a function of both valve type and percent opening, although the effect of percent opening on globe valves is not very strong, and some globe valve manufacturers ignore this change, publishing only one value.

By examining the lower graph we can see one of the tricks that is sometimes used with rotary valves to overcome a cavitation problem. Suppose for a particular application we determined that a 10 inch butterfly valve would be 80% open. Using the F_L for an 80% open butterfly we found that the valve would be operating close to ΔP_{choked}. One solution would be to go to a 16 inch globe valve which has a comparable Cv, but a 16 inch globe would cost around $70,000 and take 6 months or more to get, compared to the 10 inch butterfly which would cost about $4,500 and would very likely to be in the valve manufacturer's stock. Another possible solution is to use a 12 inch butterfly which in the same application would operate at about 60% open. From the graph you can see that the F_L of a butterfly increases considerably going from 80% to 60% open. Although the 12 inch butterfly would cost more than the 10 inch, it is still a lot less than the globe valve and would also be in stock. This approach doesn't always solve the problem, but it is worth remembering.

Valve Geometry Effects F_L

Turbulence and friction cause permanent pressure loss

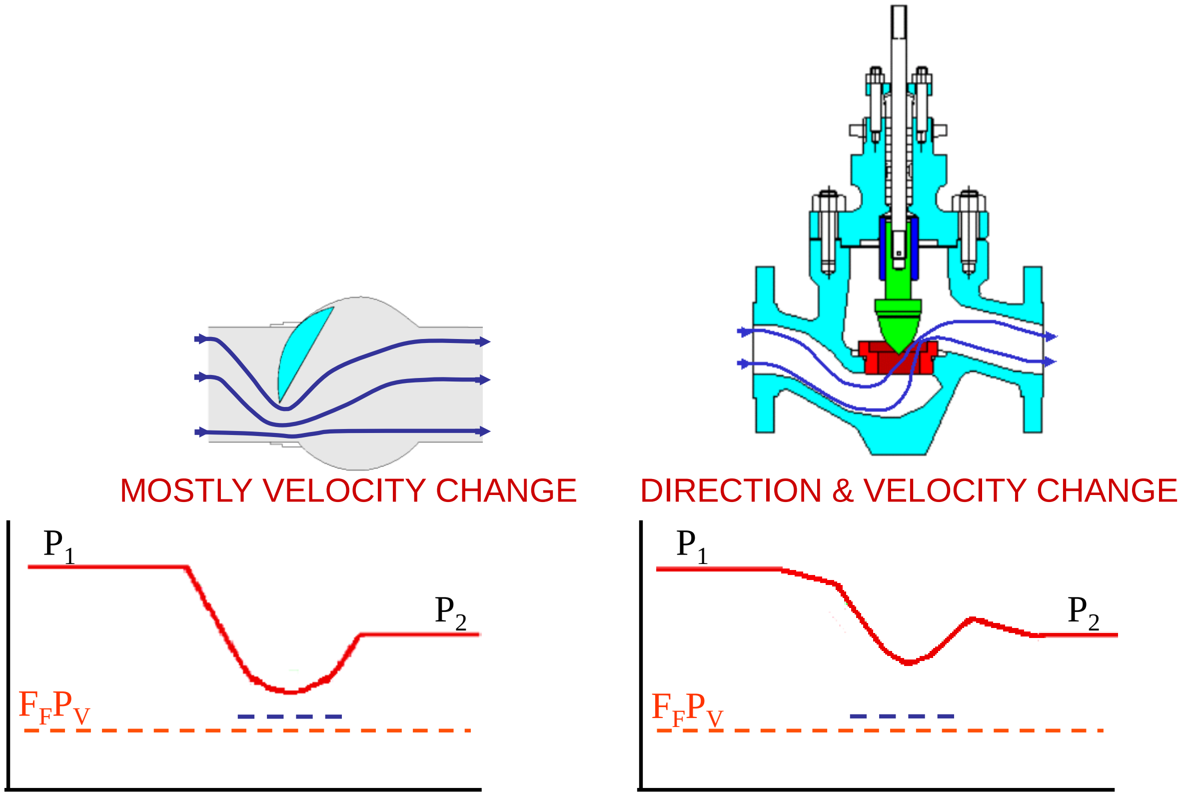


Valve geometry accounts for the differences in liquid pressure recovery factor, F_L, between the different styles of control valves.

In the previous discussions we talked about the flow passing through a restriction, where the increase in velocity causes a dip in pressure as some of the potential energy stored as pressure is converted to kinetic energy. When the area opens back up, kinetic energy is converted back to potential (pressure) energy. We also discussed that the turbulence caused by the throttling process causes some of the energy to be turned into heat energy, resulting in some permanent pressure loss. The overall pressure profile is as shown in the figure on the Left.

Permanent pressure loss can also be caused when the flow direction is changed as shown in the figure on the Right. Turbulence and friction is caused by the change in direction, and some of the energy is converted to heat energy. If the pressure loss is caused entirely by direction change, there is a permanent pressure loss without the dip that goes below p_2 when there is a velocity increase as in the case of a restriction.

Valve Geometry Effects F_L



In the more streamlined valves, like ball valves or segment valves (a segment valve is schematically shown in the Left figure) most of the permanent pressure loss is caused by the velocity increase at the restriction, and very little of the permanent pressure loss is caused by direction change.

In the less streamlined valves, such as the globe valve shown in the figure on the Right, part of the permanent pressure loss is caused by change of direction. This means that for the same amount of total permanent pressure loss a smaller percentage is caused by velocity increase, and as a result for the same amount of permanent pressure loss, the pressure dip at the vena contracta will be smaller. Since smaller pressure dips are associated larger values of F_L, the less streamlined the valve, the larger the F_L will be.

Other Recovery Factor Symbols

$$F_L = C_f = \sqrt{K_m}$$

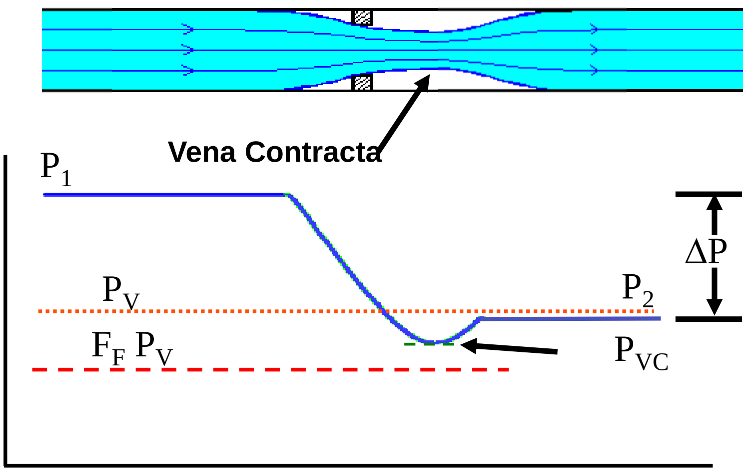
$$F_L{}^2 = C_f{}^2 = K_m$$

Before the ISA issued its standard on control valve sizing, in which it introduced the symbol F_L for the Liquid Pressure Recovery factor, both Fisher and Masoneilan had already published literature using their own symbols for the same thing. Masoneilan's C_f is identical to F_L and Fisher's Km is identical to $F_L{}^2$.

In their recent literature both Fisher and Masoneilan have switched to using the F_L terminology, but there is still a lot of the older literature in circulation, so you need to be aware of these older symbols.

Flashing Without Choked Flow

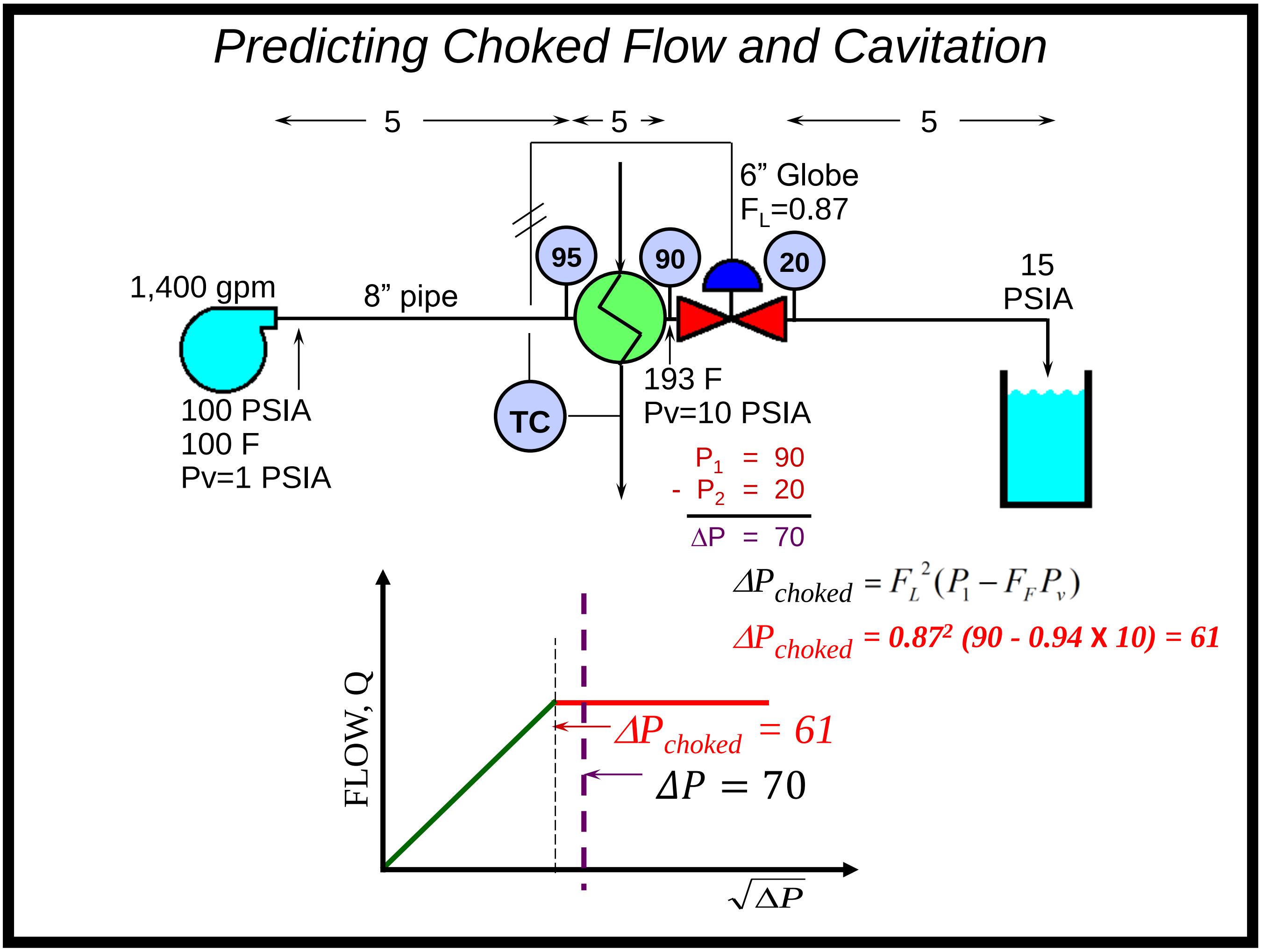
Not common, but possible



Something that does not happen very often, but is possible, is to have flashing conditions without choked flow.

As mentioned previously, for flow to choke, as defined in the ISA and IEC control valve sizing equation standards, the pressure at the vena contracta must drop to $F_F\ P_V$. Given just the right combination of P_1, F_F, P_V and P_2 it is possible to have the situation pictured in the figure. The downstream pressure, P_2, can drop to (or just below) the vapor pressure, P_V. Vaporization, and thus “flashing,” can begin somewhere downstream of the vena contracta. But the condition required for choked flow (that the vena contracta pressure drops to $F_F\ P_V$) is not met.

If your control valve sizing program states that there is flashing, but not choked flow, this is the explanation.



Let's take a look at an example where our task is to determine whether we will have a choked flow problem, with the potential for cavitation damage and if so look at some of the things we can do to eliminate the problem.

This is a temperature control system where a hot process is to be cooled in a heat exchanger. 100 degree F cooling water heats up to 193 degrees as it goes through the heat exchanger and then is sprayed into a cooling pond at atmospheric pressure. We will use a GLOBE valve to regulate the cooling water flow and thus control the temperature of the process.

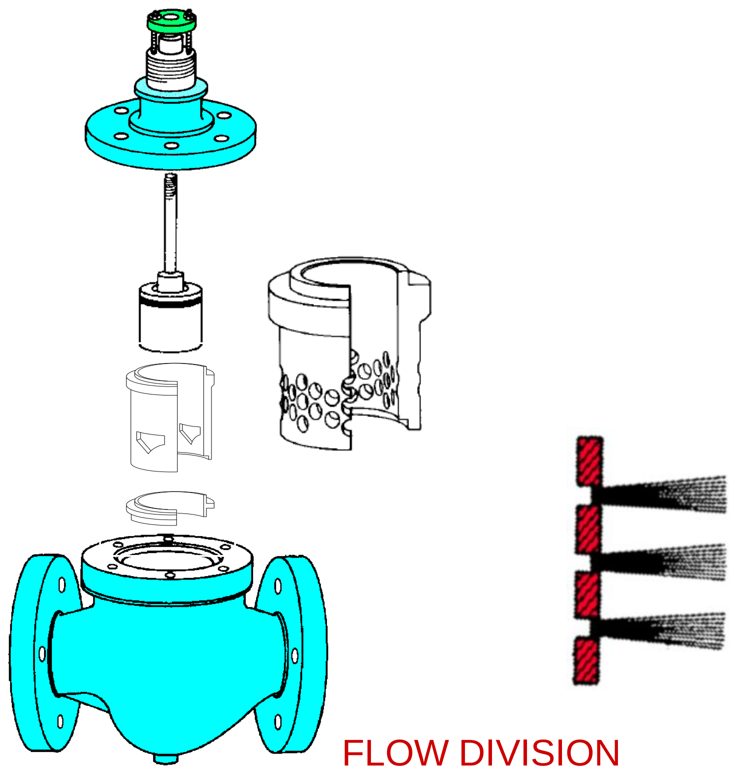
We need to calculate the terminal pressure drop and compare it with the actual pressure drop to see if there will be choked flow and a cavitation problem.

The correct way to determine the actual pressure drop across a control valve is to start at a point upstream of the valve where we know the pressure (in this case, the pump outlet) and then subtract the pressure losses in the upstream piping, and equipment, until we get to the valve where we will then know the pressure immediately upstream of the valve, P_1. In this case the pump pressure is 100 psia, the loss in the upstream piping is 5 psi, (from a hydraulic table like the Crane book) and 5 psi loss through the heat exchanger (based on information from the manufacturer) which leaves 90 psia just upstream of the control valve, so P_1 is 90 psia. (We are using absolute units because the formula requires pressure in absolute units.)

The next step is to go downstream of the valve to a location where we know the pressure, in this case where it sprays out into the air where the pressure is atmospheric (approximately 15 psia), then work upstream toward the valve adding in the pressure losses in the piping until we reach the valve where we will then know the pressure immediately downstream of the valve, P_2. (We add because we are going in the opposite direction to the flow and the direction of pressure loss.) In this case P_2 = 15 psia +5 psi = 20 psia. We can now subtract P_2 from P_1 and we get ΔP = 90 - 20 = 70.

We now calculate the choked pressure drop, ΔP_{choked}, by substituting the value of P_1 we just determined, the given vapor pressure, the calculated value of F_F and using 0.87 as a typical F_L for a globe valve. We get a choked pressure drop of 61 psi. The actual pressure drop is 70 psi, which is to the right of the terminal pressure drop which means that we are in the choked flow region and cavitation with the potential for valve damage will be a problem.

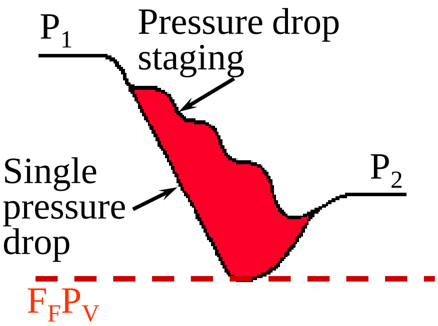
Cavitation Abatement Techniques

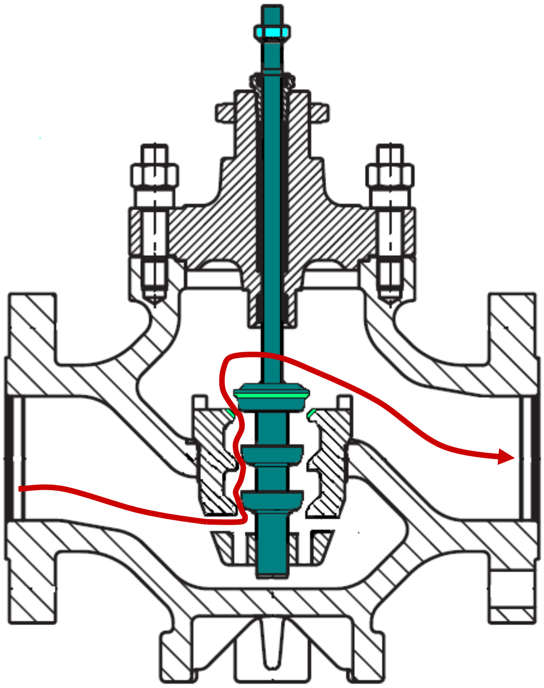


Let's look at some possible solutions to the cavitation problem. This is a globe valve style that has been around for many years, called a “Cage Trim” globe valve. These valves have triangular openings in the cage to characterize the flow. As an additional cost option we can get a cage that instead of the triangular windows has many holes drilled it.

This cavitation abatement method is called **Flow Division** because the flow is divided up into many small flows. It is effective for reducing cavitation damage because the size of the cavitation bubbles is partly a function of the size of the hole the flow is going through. Smaller holes make smaller bubbles which give up less energy when they collapse, which means less noise and weaker shock waves which cause less damage.

Cavitation Abatement Techniques





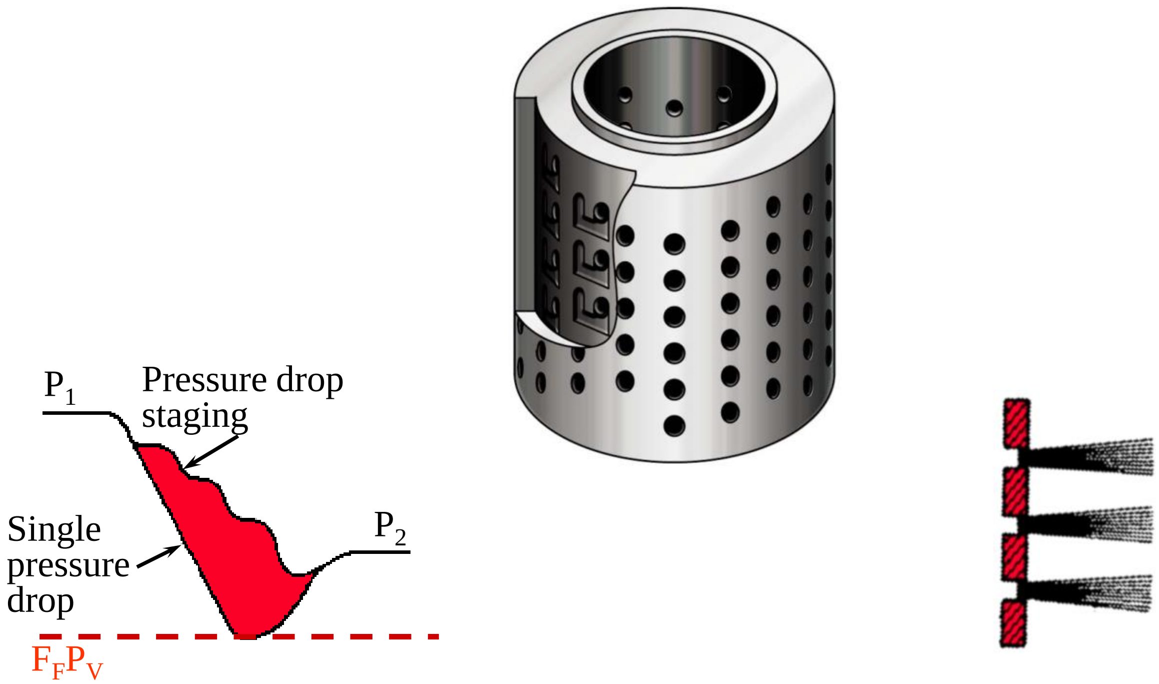
PRESSURE DROP STAGING

The other method used to reduce or eliminate cavitation problems is the called **Pressure Drop Staging.**

This valve is really three valves in series (see the three plugs and seat rings) which breaks one big pressure drop up into several small pressure drops. The lowest pressure dip will be much smaller than the dip resulting from taking the pressure drop in a single step.

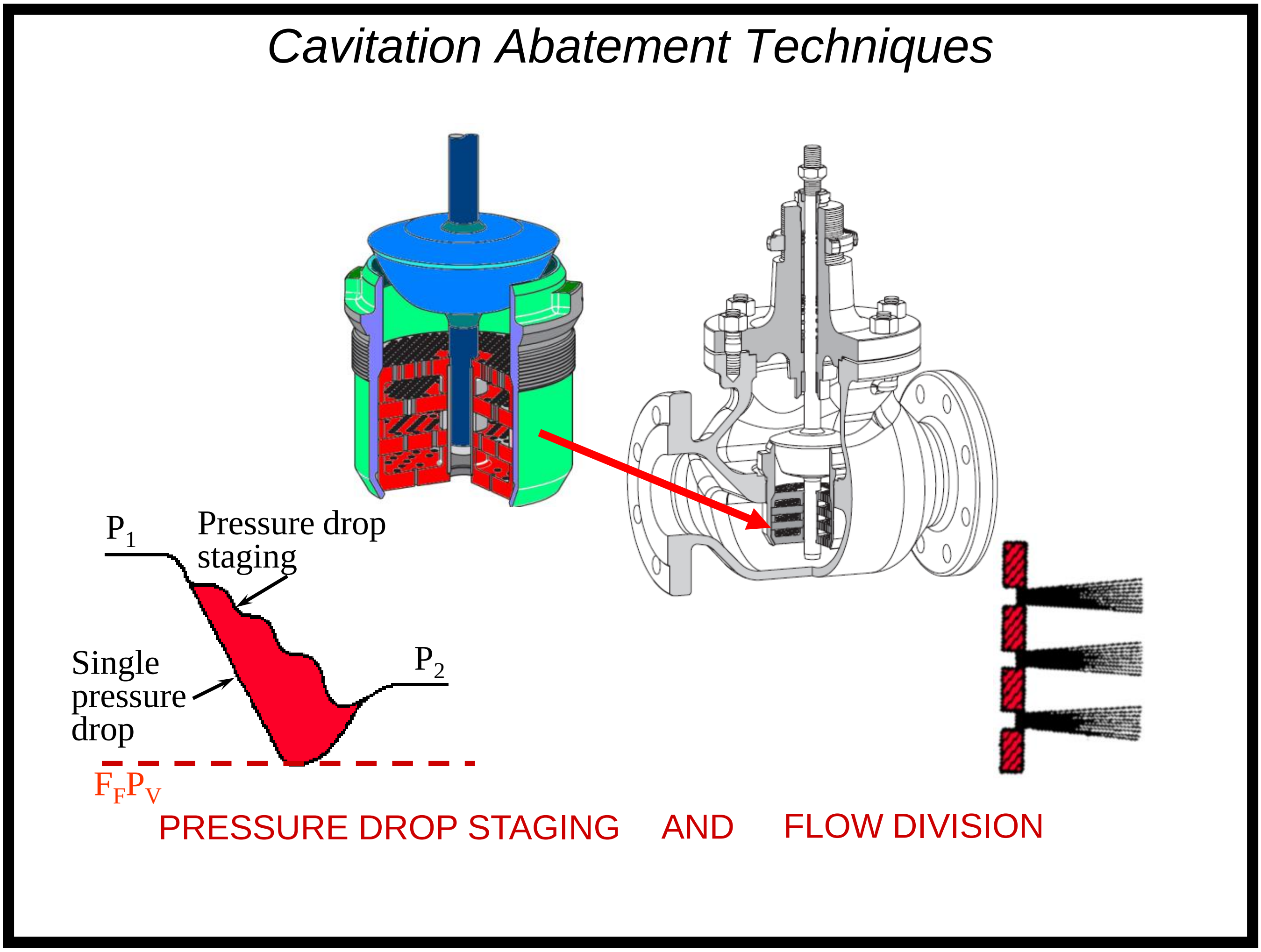
The photograph in the upper left is of a 5 stage plug and seat of the same type as the 3 stage configuration in the drawing.

Cavitation Abatement Techniques



PRESSURE DROP STAGING AND FLOW DIVISION

This shows one way that combines the principles of flow division and pressure drop staging in a globe valve. This is a globe valve cage consisting of three concentric cages, each with holes drilled in it. Pressure drop occurs as the liquid passes through the holes of each successive cage and the holes divide the flow into many small flows.



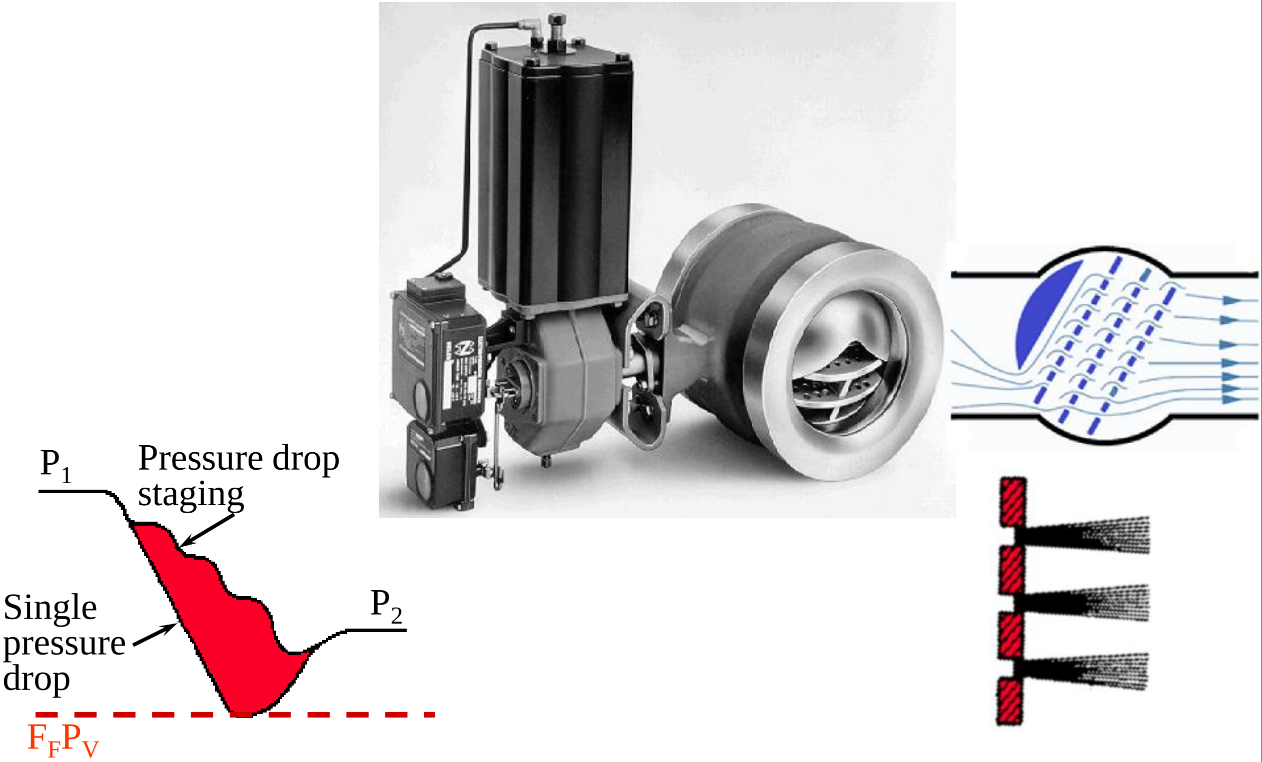
This valve also combines pressure drop staging and flow division.

Where the previous valve only took the total pressure drop in three steps, this one takes it in five smaller steps. That makes this valve more effective and also more expensive.

The flow enters from the bottom and passes upward through four sets of holes, each stage taking a portion of the total pressure drop before the final pressure drop across the plug and seat ring.

Notice that each successive stage has more holes. This is so that the earlier stages take a larger portion of the total pressure drop and the later stages take a smaller portion. The reasoning is that where the pressure is high, even a fairly large pressure dip will still not come near the $F_F P_V$ line. In the later stages where the pressure is lower, the pressure drops and resulting pressure dips need to be smaller to make sure the pressure dip does not drop to the vapor pressure.

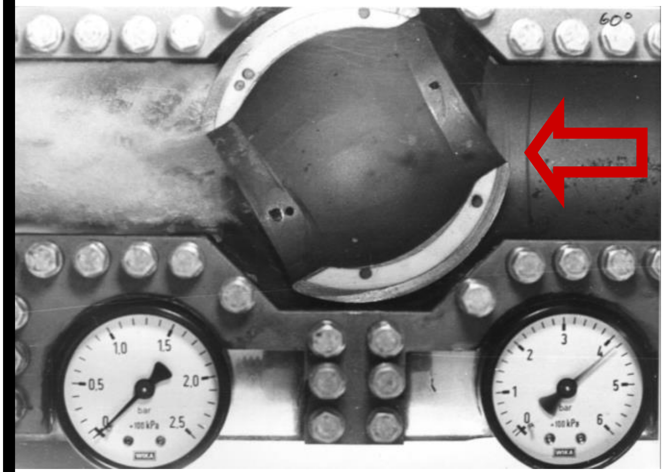
Cavitation Abatement Techniques



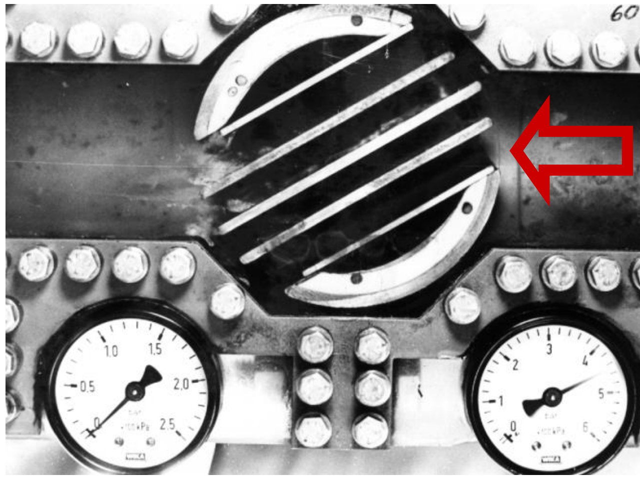
PRESSURE DROP STAGING AND FLOW DIVISION

This is a segment ball valve with cavitation abatement trim. It uses both of the cavitation reduction methods just mentioned. (1) The holes in the attenuator plates break the flow up into many small flows, which results in smaller bubbles and thus less noise and damage, and (2) it stages the pressure drop into four small drops, with the pressure at the last dip not getting as close to the $F_F P_V$ line it would with a single pressure drop.

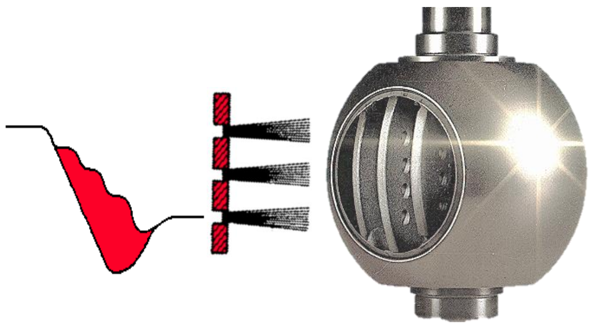
Effectiveness of pressure drop staging and flow division



86 dB(A)



76 dB(A)

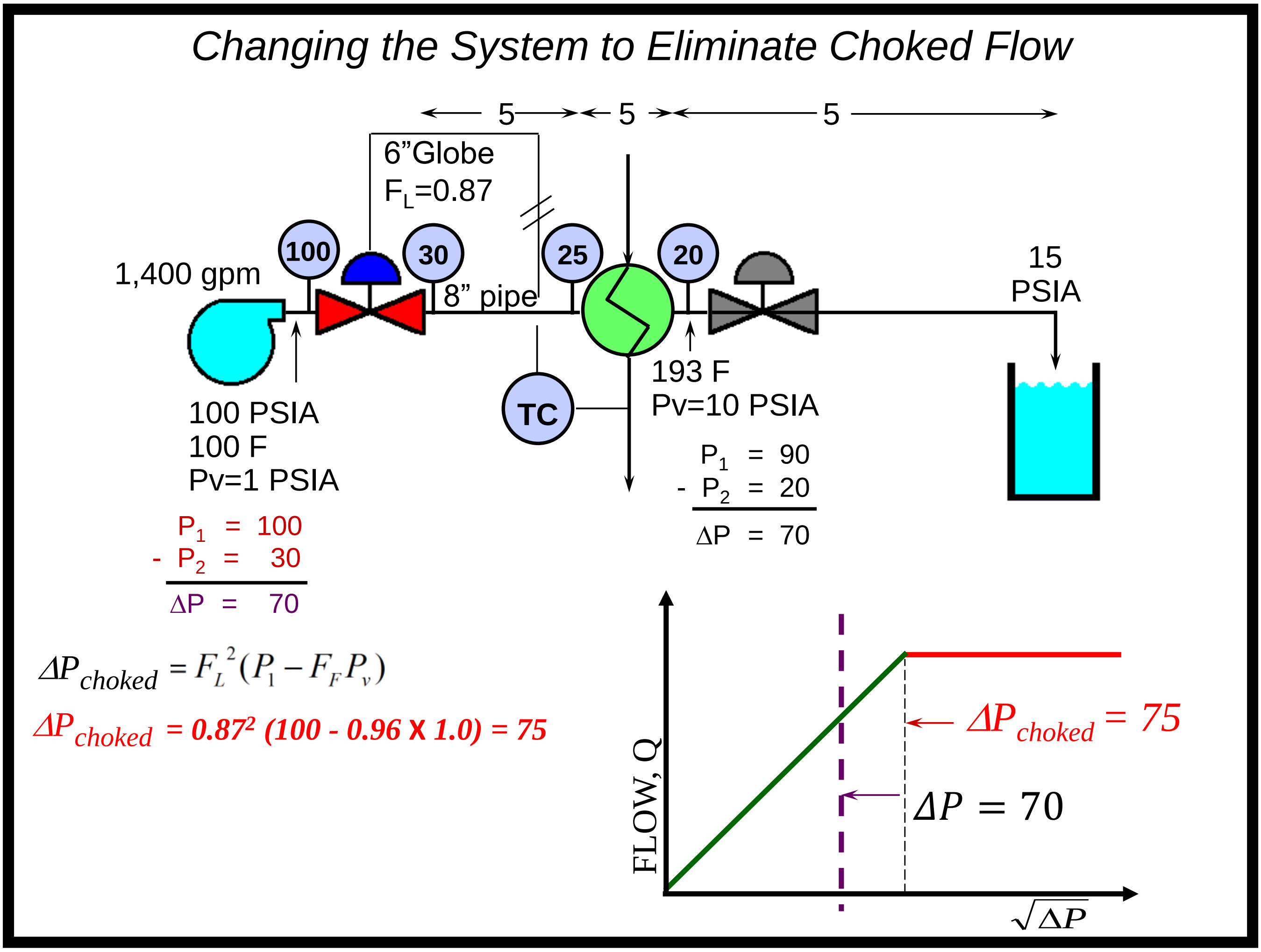


This demonstrates how well the combination of pressure drop staging and flow division works. These are two 2 inch ball valves that have been cut in half and covered with clear plastic to keep the water in and to let us see what is going on inside.

The inlet pressure is about 4 bar (about 60 psig) and the outlet pressure is zero bar (0 psig).

The one without three plates with holes in them is generating a large number of cavitation bubbles. The one with the metal plates that combine the techniques of pressure drop staging and flow division, which is taking a slightly higher pressure drop and only a very few cavitation bubbles. The measured noise level is 10 dB(A) lower.

A sound pressure level increase of 10 dB(A) is perceived by humans as being twice as loud.



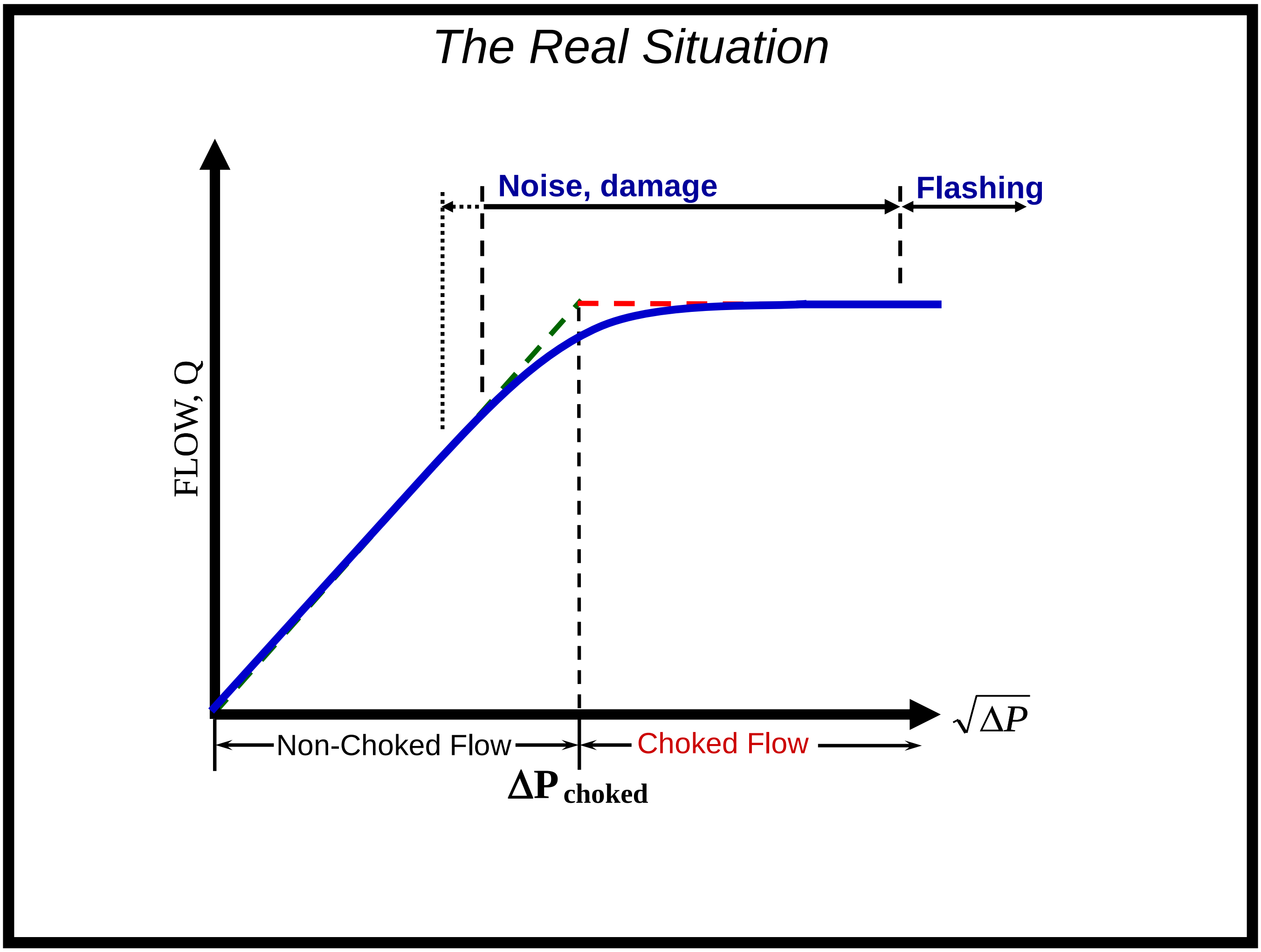
Let's go back to our system and see if there is some way we can solve the cavitation problem without spending any extra money on special valves.

Let's try moving the valve upstream, closer to the pump, and see what happens. When we do that the inlet pressure, P_1 increases by 10 psi to 100 psia, and we have also moved to the cool side of the heat exchanger where the vapor pressure drops from 10 psia to 1 psia. Examining the choked pressure drop equation we can see that increasing P_1 causes the choked pressure drop to increase, and that decreasing P_V also causes the choked pressure drop to increase.

This time when we do the math we get a choked pressure drop of 75 psi . (NOTE that we are still using a standard globe valve with the same F_L of 0.87.)

NOTE that moving the valve does not affect the actual pressure drop, because both P1 and P2 increase by 10 psi, so ΔP is still 70 psi.

This time when we compare actual ΔP with ΔP_{choked}, we see that the actual pressure drop of 70 psi is well to the left of the choked pressure drop of 75 psi and we have solved the choked flow problem. Later in this section we will see that it is possible to have cavitation damage without choked flow.

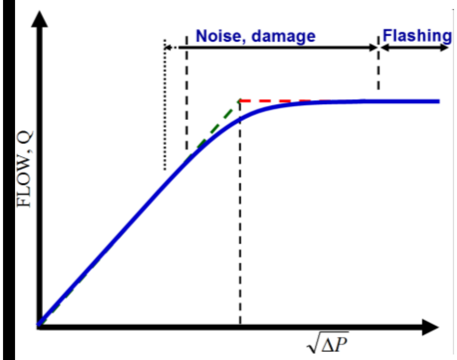


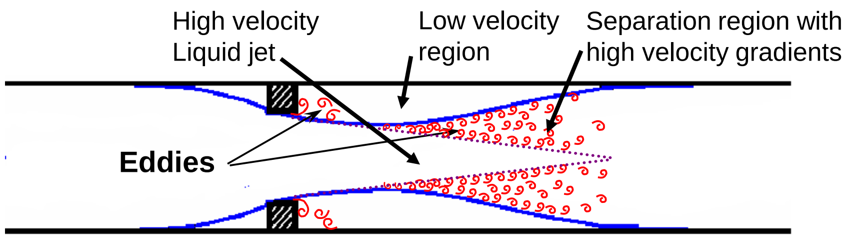
Now we will complicate the cavitation prediction problem. In the previous example we pretended that flow increases linearly with the square root of ΔP until ΔP reached ΔP_{choked} and then immediately became fully choked with no further increase in flow. In reality there is a certain amount of rounding out of the graph at the ΔP_{choked} point. For years people have used the same method we did in the previous example to predict cavitation. In recent years, though, most users and manufacturers have recognized that operating too close to the ΔP_{choked} point is very likely to result in cavitation damage. (See the next pages for a discussion of more refined methods for predicting cavitation damage.)

The rounding of the flow graph makes predicting cavitation damage more complicated than simply comparing the actual pressure drop with the calculated choked pressure drop (which assumes the classical discussion of a sudden transition between non choked flow and choked flow). Even before the non-choked portion of the flow vs. square root of pressure drop line starts to round off, and flow starts to choke, there is likely to be both noise and cavitation damage.

Next we will see how both noise and damage can begin even before the pressure in the main flow stream at the vena contracta drops to the vapor pressure of the liquid and causes flow to choke.

The Real Situation





Eddies create localized pressure reduction and recovery

- Localized cavitation
  - Vena contracta pressure still above $F_F P_v$
  - Flow is not choked
  - Noise and damage are likely

The first stages of cavitation begin when the average pressure in the main flow jet at the vena contracta is still above the vapor pressure of the liquid and flow is not choked. At points of abrupt increase in flow area the streamlines that are attached to the physical boundaries of the valve can separate and when they do, they form vortices or eddies. The rotational velocity in the eddies can be high enough that the local pressure inside the eddy drops below the vapor pressure of the liquid and vapor bubbles form. As the eddy's rotational velocity decreases, the pressure surrounding the vapor bubbles increases and the bubbles to collapse.

Vortices (and cavitation) also form in the shear layer adjacent to the main jet where high velocity gradients exist.

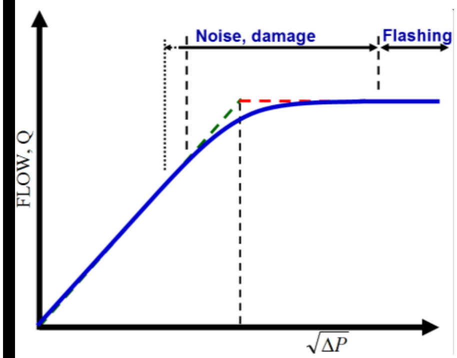
This level of cavitation has the potential for generating noise and causing damage even before the flow curve starts to deviate from a straight line.

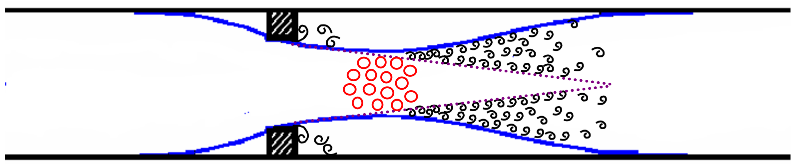
Note that this topic is discussed in Section 1.4 of "*Cavitation Guide for Control Valves*" published by the US Nuclear Regulatory Commission.

Try pasting this URL into your browser to download the book:

https://www.osti.gov/servlets/purl/10155405

The Real Situation





Vena contracta pressure drops to $F_F\ P_v$

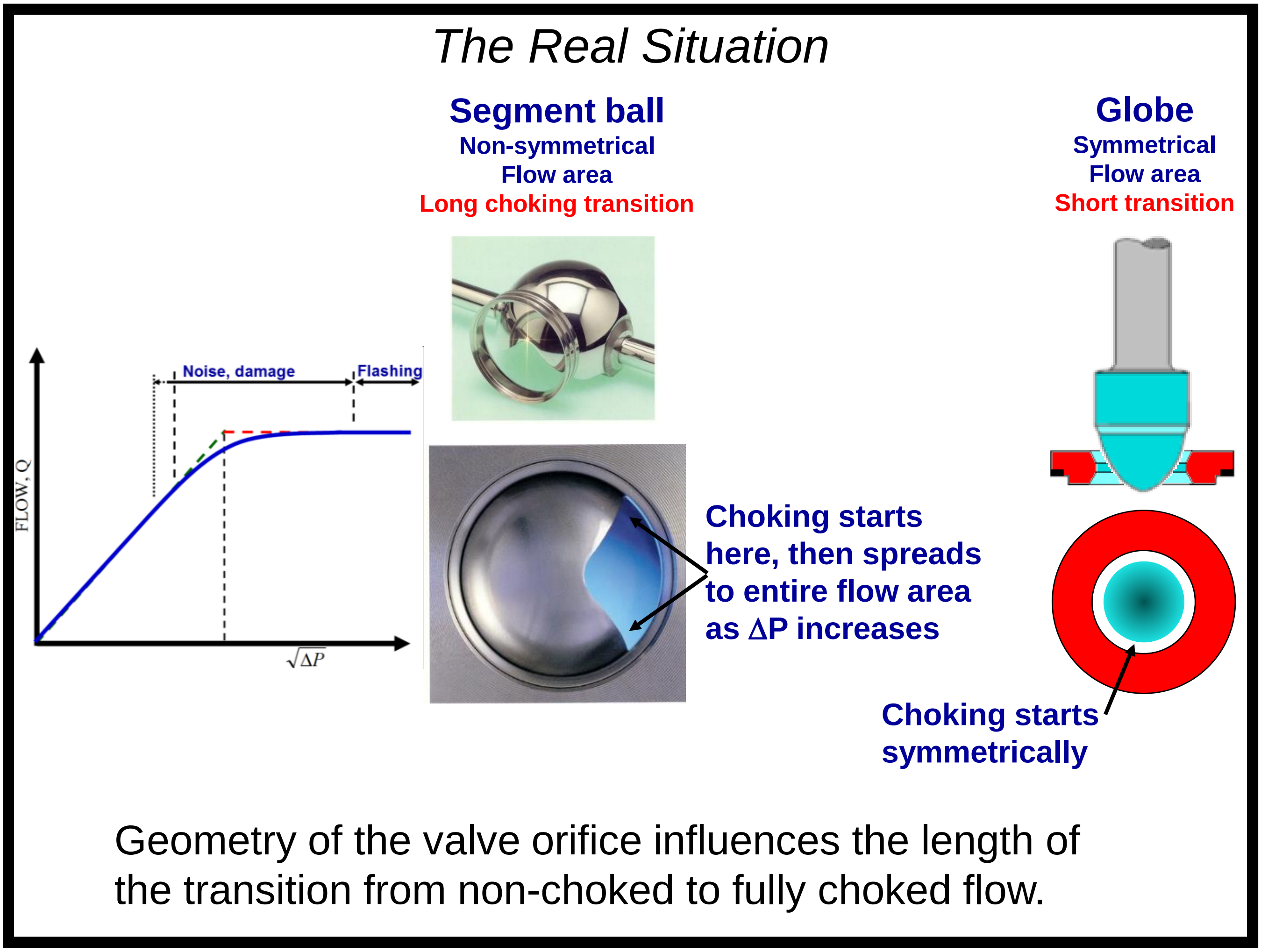
- Vapor bubbles form
- Pressure at vena contracta determines flow
- Pressure at vena contracta cannot be less than $F_F\ P_v$
- Flow chokes
- Severe cavitation damage very likely

As the pressure drop across the valve is increased, the velocity at the vena contracta increases and finally the pressure at the vena contracta drops to F_F times the vapor pressure of the liquid and vaporization begins.

The flow through a control valve depends on the pressure at the vena contracta. Since it is not possible for the pressure at the vena contracta to be less than F_F times the vapor pressure of the liquid, the flow becomes choked, that is, further decreases in downstream pressure do not cause the flow to increase further.

Operating a valve at or beyond the choked point calculated from F_L (Δp_{choked}) is almost sure to result in excessive noise and cavitation damage.

On the next page we will see that depending on the geometry of the valve the transition from non choked flow to fully choked flow can be gradual or fairly quick.



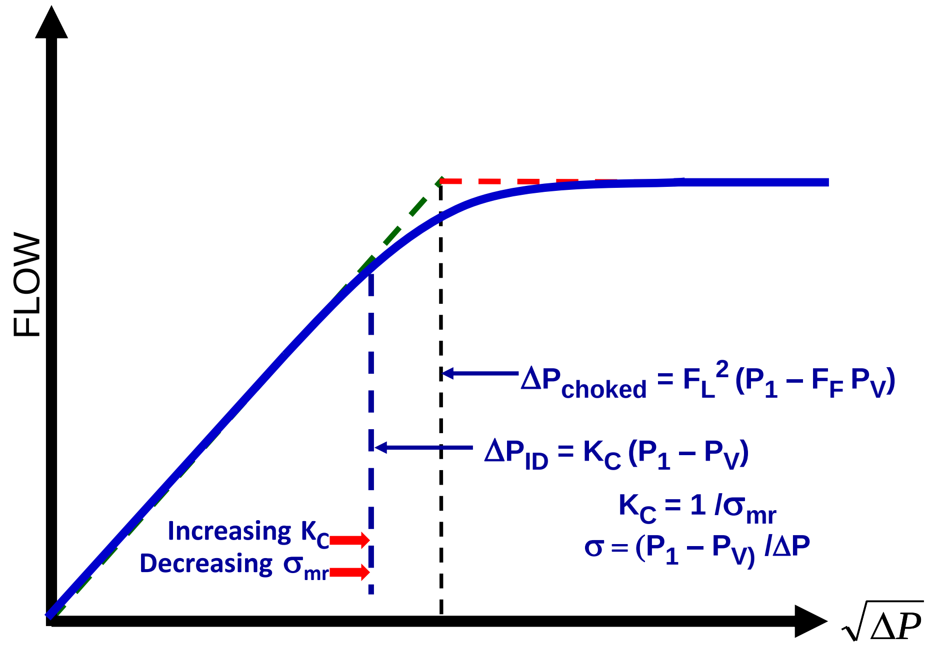
The shape of the choking curve is influenced by the geometry of the valve orifice.

As an example, look at the picture of the segment ball valve which will have a significant choking transition curve . For the segment ball valve, the reason why the choking range has a significant transition region is due to the configuration of the flow passage. The segment valve has an irregular shaped flow area. The restricted flow at the narrow ends of the irregularly shaped area creates locally higher shear stresses causing cavitation (and eventually choking) to first occur in these regions. Eventually the entire area will choke as the pressure drop across the valve increases. As a result of the non-uniform distribution of cavitation potential, choking occurs in different locations inside the valve at different flow rates. This causes the choking transition region.

In contrast to the segment ball valve, the globe valve with a parabolic plug has a very symmetrical flow area and as a result, choking will start at approximately all points in the flow path at the same time, resulting in a much shorter transition region between non choked flow and fully choked flow.
This subject is also discussed in ISA-RP 75.23-1995 "*Considerations for Evaluating Control Valve Cavitation*" Section B.8.7.

Also beginning with the last paragraph on Page 41 of "*Cavitation Guide for Control Valves*" published by the US Nuclear Regulatory Commission https://www.osti.gov/servlets/purl/10155405 discusses this topic as it applies to butterfly valves. The figure on Page 42 is based on a 10 inch butterfly valve at from top to bottom shows the shape of the flow vs. square root of pressure drop for increasing degrees of valve travel.

Cavitation Damage Prediction



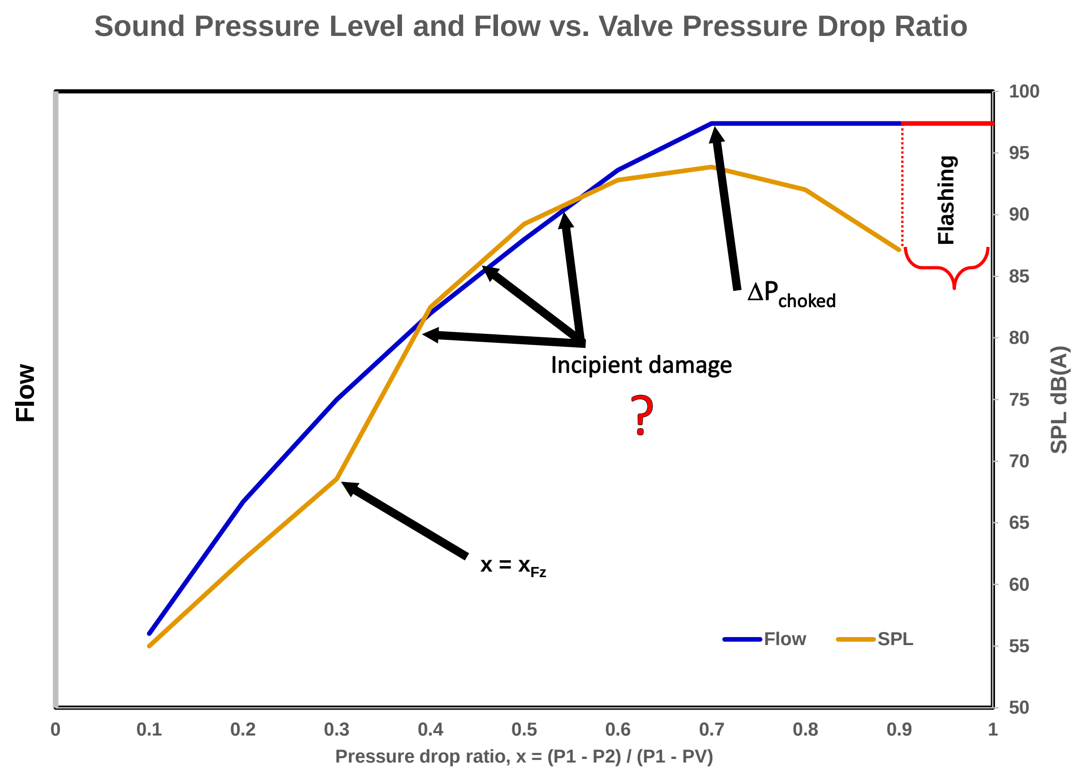
Some valve manufacturers predict the beginning of cavitation damage by defining an incipient damage pressure drop, which is sometimes referred to as ΔP_{ID} as shown in the formula in the figure above. These manufacturers evaluate actual application experience with cavitation damage and assign what they believe to be meaningful values of K_C to their valves. One manufacturer, for example, uses a K_C for stem guided parabolic plug globe valves that is equal to 0.7. There are other manufacturers who, based on the recommended practice, ISA–RP75.23–1995, use sigma (σ) to represent various levels of cavitation. These valve manufacturers publish values of either σ_{mr} (the **m**anufacturers **r**ecommended value of sigma) or σ_{damage}. Sigma is defined as "$(P_1 - P_V)/\Delta P$." σ_{mr} and K_C are reciprocals of each other and thus convey the same sort of information. Higher values of K_C move the point of incipient damage closer to ΔP_{choked}, where lower values of σ_{mr} do the same.

In the past, valve manufacturers determined K_C by simply noting the pressure drop at which the flow curve deviated from a straight line by 2%. Using this method is unreliable because experience has shown that unacceptable levels of damage can begin even before the flow curve deviates from a straight line.

It is also important to understand that F_L is NOT a cavitation parameter. It is a choked flow parameter and its only use is to determine the theoretical choked flow point based on the assumption that the choked flow point, ΔP_{choked}, is the intersection of the two straight dashed lines shown above in red and green. **Using F_L as a cavitation parameter is almost sure to result in unacceptable levels of cavitation damage.**

Cavitation Damage Prediction

Based on predicted noise



The method that Neles uses to predict cavitation damage is based on the fact that the same thing that causes damage also causes the noise, namely the collapse of vapor bubbles.

This slide shows a typical relationship between pressure drop (expressed as pressure drop ratio, x, which is defined as $\mathbf{x = (P_1 - P_2) / (P_1 - P_V)}$, where P_1 is the pressure upstream of the valve, P_2 is the pressure downstream of the valve and P_V is the vapor pressure of the liquid..

At low pressure drop ratios, the noise is low and the flow is turbulent with no cavitation bubbles forming or collapsing. The point labeled $\mathbf{x = x_{Fz}}$ is where cavitation bubbles start to form and above $\mathbf{x_{Fz}}$ noise starts increasing more rapidly. $\mathbf{x_{Fz}}$, the point where the first cavitation noise begins is a valve parameter based on noise tests, and published by valve manufacturers for each valve style and each ten percent increment of valve travel.

At first, as the pressure drop increases beyond $\mathbf{x = x_{Fz}}$ so does noise. Experience has shown that cavitation damage is not a problem at first. At some point the noise reaches a maximum. This point tends to be near where the pressure drop across the valve equals ΔP_{choked} which corresponds to fully choked flow, and where cavitation damage is almost sure to become unacceptable. The trick is determining the **Incipient damage** point, which involves time consuming tests. Consequently, there is practically no information on exactly where this **Incipient damage** point is.

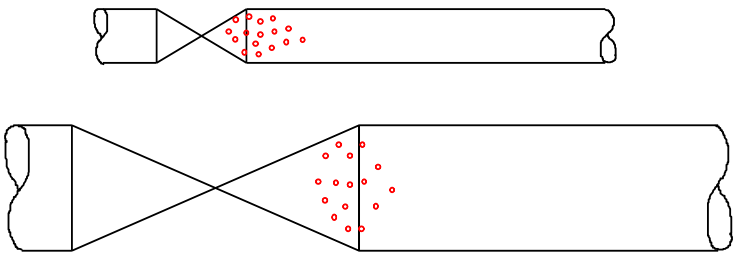
The idea of correlating noise with cavitation damage got its start in 1985 when Dr. Hans Baumann published an article in *Chemical Engineering* magazine where, based on some limited damage tests, he established a maximum SPL of 85 dBA as the upper limit to avoid unacceptable levels of cavitation damage in butterfly valves. To verify this premise, Neles did a study of many applications, where in some cases cavitation damage was minimal and in others it was excessive. The conclusion of the study was that it is possible to predict that damage will be within acceptable limits as long as the predicted noise level is below limits established in the study. In the case of a 6 inch valve, the limit turns out to be 85 dB(A).

Note that the graph, shown in the figure above, of flow from low pressure drop ratios up to the choked flow point is not a straight line. That is because, most commonly, flow in a valve vs. pressure drop is on a flow squared scale, and here the pressure drop scale is linear.

Cavitation Damage Prediction

Maximum Calculated SPL

UP TO 3” VALVE SIZE	80dBA
4” TO 6”	85 dBA
8” TO 14”	90 dBA
16” AND LARGER	95 dBA



To avoid cavitation damage:

- Calculated SPL must be less than above limits.
- The actual pressure drop must not exceed the choked pressure drop. ($\Delta P < \Delta p_{choked}$)

Because the same number of bubbles per second that result in a sound pressure level of 85 dBA and present a potential for cavitation damage in a 6 inch valve are more spread out and less concentrated in an 8 inch valve, we can allow more bubbles per second and hence a higher noise level in larger valves.

Applying the same reasoning, the number of bubbles per second that would be tolerable in a 4 inch valve would be more concentrated in a 3 inch valve, so to avoid cavitation damage in the smaller valve, the noise limit has to be lower.

The Nelprof control valve sizing program checks the calculated noise against the above recommendations, and issues a warning if the calculated noise level indicates the potential for cavitation damage.

Note that regardless of the noise calculation, the actual pressure drop must be less than the choked pressure drop, because experience has show that operating above the choked pressure drop is almost certain to result in damage. The Nelprof control valve sizing program also checks to see that the choked pressure drop is not exceeded. If it is, the program issues a warning.

The limits listed are based on noise calculations made with VDMA 24422 1979.

Recommended Velocity Limits

Liquid Service

- Erosion
- Corrosion
- Stability (butterfly)

Ball, Segment, Globe

32.8 fps (continuous) [10 m/s]

39 fps (infrequent)

Butterfly

23 fps (continuous)

27 fps (infrequent)

Neles also make recommendations on the maximum velocity of the liquid in a valve body. (The velocity is calculated at a point where the cross-

sectional area of the valve body equals the area of a circular pipe of the same diameter as the nominal valve size.)

The concerns are for EROSION, CORROSION, and STABILITY

If there are any erosive particles in the flow stream, high velocities can result in higher than desirable levels of erosion.

Stainless steel protects itself from corrosion by forming a “passivation” layer on its surface which is very corrosion resistant. High velocity fluids can wear away the passivation layer, exposing the unprotected metal to corrosion.

High velocities going through a partially open ball or over a butterfly valve disk can cause forces that attempt to move the valve.

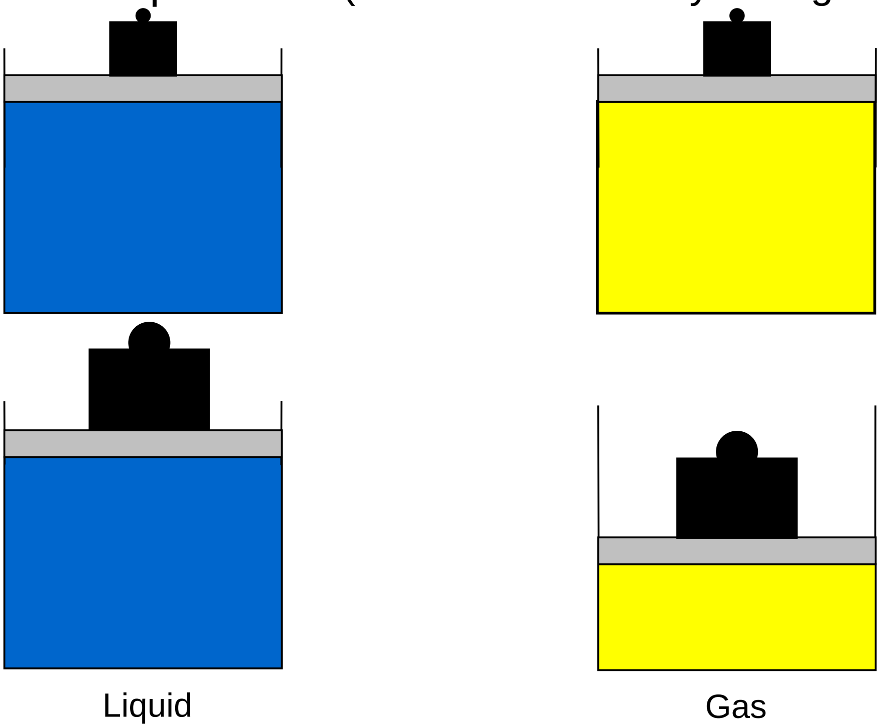
(The reason for selecting 32.8 fps is that the criteria was established by Neles’ research department in Finland where they measure velocity in meters per second. 10 meters per second (32.8 fps) is a nice round number)

If I had set the criteria, I probably would have chosen 35 feet per second which is 10.668 meters per second.

Compressible Flow

Gas vs. Liquid Flow

- Liquids are incompressible (volume and density are constant)
- Gas is compressible (volume and density change with pressure)



The main difference between gas and liquid is that gasses are compressible. As you increase the pressure, the volume decreases and the density increases.

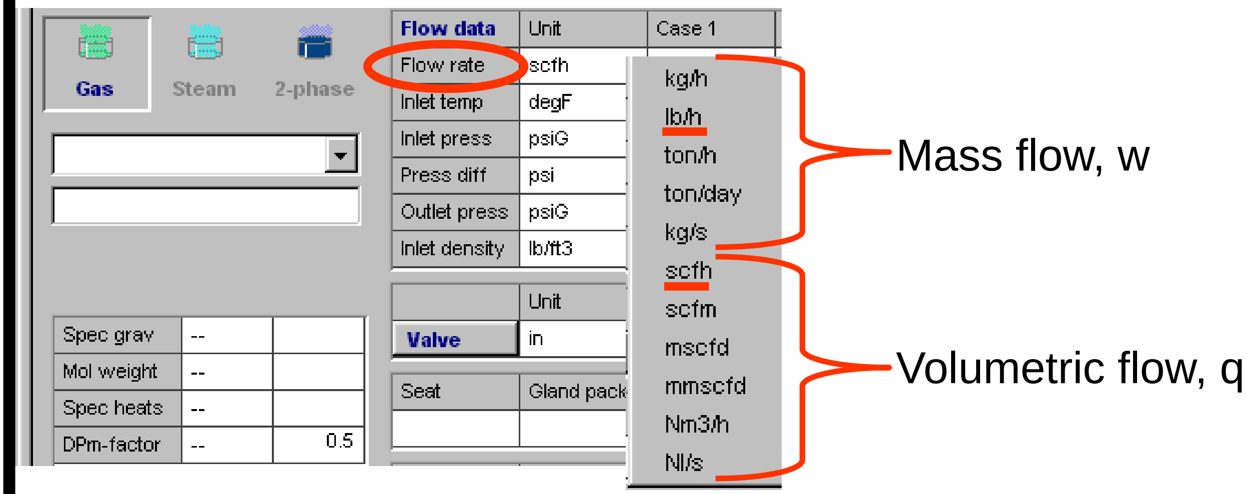
With liquids, for all practical purposes, the volume and density remains constant regardless of the pressure.

Gas Terminology

- Flow units
- Density
- Molecular weight
- Specific gravity
- Compressibility factor
- Ratio of specific heats
- Mach number

We will go over some of the terminology that applies to gas flow.

Flow Units



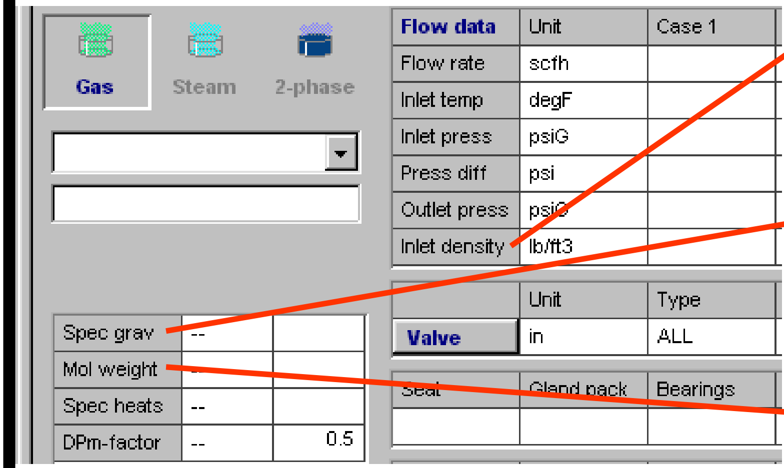
1 standard cubic foot of gas is the amount of gas that will occupy 1 cubic foot at standard conditions.
(14.7 psia, 60^o F)

Gas flow is measured in two types of units, volumetric flow and mass flow. Volumetric flow is nearly always in “Standard” or “Normal” units, that is the number of volumetric units if the gas were at standard or normal pressure and temperature. ("Standard" reference conditions are 14.7 psia, 60 deg. F. "Normal" reference conditions are 1.013 bar, 15 deg. C). Volumetric units are the most common.

Occasionally volumetric flow will be expressed in ACTUAL cubic feet or meters. For example, in the case of a positive displacement vacuum pump, each cycle of the pump removes the same volume of gas, but as the pressure decreases, the actual number of molecules with each cycle of the pump decreases. In this case the ACTUAL volume stays the same and the STANDARD volumetric flow decreases as the pressure decreases.

Mass flow units are most often used in cases where the fluid appears in the process sometimes in liquid form and sometimes in gas form. For example, in a boiler, where one pound of water becomes one pound of steam, when condensed again becomes one pound of water. It is also common practice to refer to gasses that are near the temperature where they can also exist as liquids, as “Vapors.”

Density, Specific Gravity or Molecular Weight



Density, ρ:

or

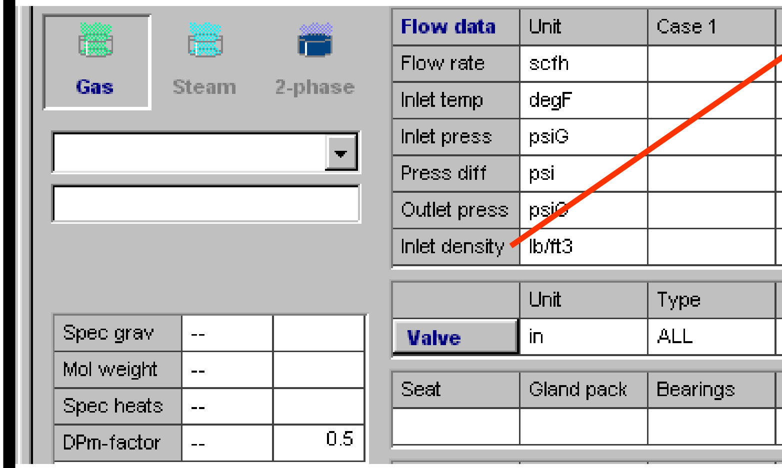
Gas specific gravity, G_g:

or

Molecular weight, M:

The gas sizing equations (and control valve sizing programs) require as an input density, specific gravity or molecular weight.

Density



Density, ρ: *

Weight per unit volume

- Most accurate when based on actual measurements at many pressures and temperatures
- Accurate tables are available for some common vapors such as steam and ammonia
- Most sizing programs contain density data for steam when pressure and temperature are given
- Density will be different for each case if the upstream pressure or temperature is different..

* (Sometimes called Specific Weight, γ)

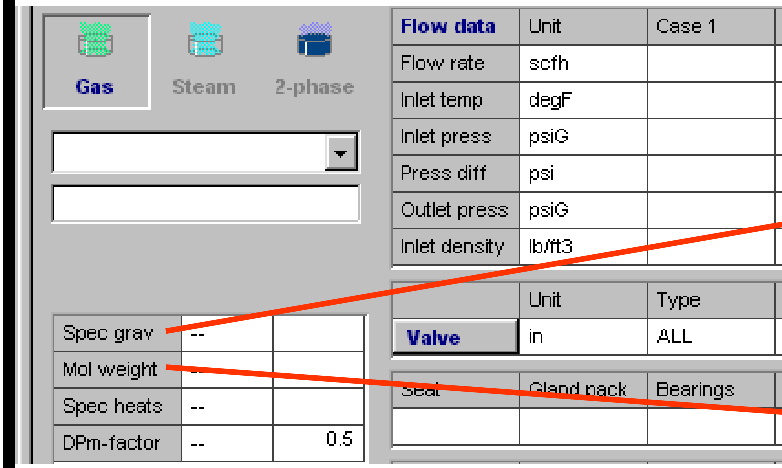
The basic information required by the sizing equations is density. The most accurate method of determining density is to actually measure the density at many pressures and temperatures and generate density tables. This has been done for some gasses where very accurate calculations are required. There are extensive tables for steam, ammonia and a few others.

Most computer sizing programs include steam density data, given the upstream pressure and temperature.

Keep in mind that if you have several cases where the upstream pressure or temperature are different, the density will also be different.

In some places Density is called Specific Weight (the same thing as Density, that is, weight per unit volume, with the symbol gamma).

Molecular Weight and Specific Gravity



If density is not known, the sizing equations approximate it from:

Gas specific gravity, G_g

$G_g = M/M_{air} = M/28.97$

or

Molecular weight, M ..

Usually, the actual density is not known, but the molecular weight or specific gravity is known and there are versions of the gas sizing equations that approximate density from molecular weight or specific gravity. There are extensive gas tables that list molecular weight and/or specific gravity.

The molecular weight of a gas is a property that is based on the mass of a molecule of that gas. The gas specific gravity is defined as the molecular weight of that gas divided by the molecular weight of air (which is 28.97). So, molecular weight and specific gravity convey exactly the same information.

SOME ADDITIONAL INFORMATION ABOUT MOLECULAR WEIGHT:

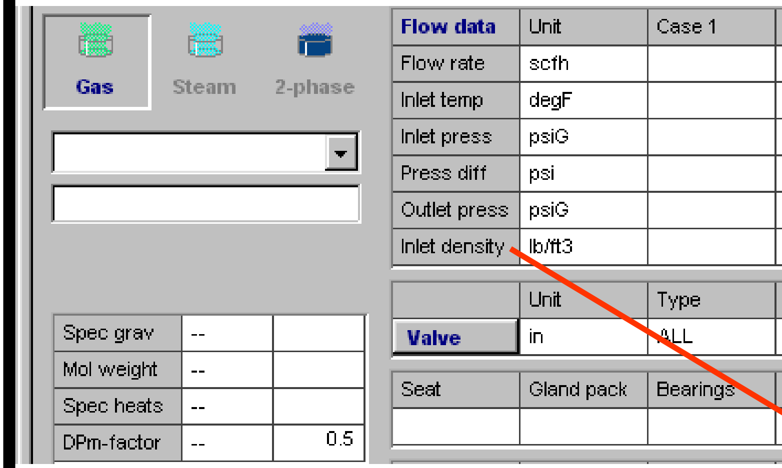
The molecular weight of a gas is the sum of the atomic weights of the atoms making up the molecule.

The atomic weight of an element is the relative mass of an atom of that element compared to the mass of a carbon-12 atom. (Carbon-12 is the most common isotope of carbon, and because the AMU is defined as 1/12 the mass of a Carbon-12 atom, Carbon-12, by definition, has an atomic weight of 12.)

Also, for those who are interested, 6.022… X 10^{23} (the **Avogadro constant**) molecules of a gas has a weight in grams equal to its molecular weight.

Another common term for molecular weight is molecular mass.

Compressibility Factor

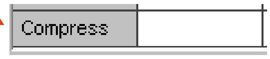


If **Specific gravity**

or

Mol. weight is entered,

Density field is replaced by “Compress” (compressibility factor, z) field.



Z = how much density differs from that of an ideal gas

- Accurate tables are available for a few gases
- z = 1.0 is a good estimate, for valve sizing purposes, for most (but not all) gasses..

$$\rho = \frac{p\ M}{10.732\ T\ Z}$$

When a molecular weight OR specific gravity is entered, the “Inlet density” field changes to the “Compress” (compressibility factor) field. Without the addition of a compressibility factor (symbol “z”) the calculation of density (and Cv calculation) will be an approximation based on the ideal gas equation (the parts of the density equation shown excluding the symbol “Z”). The ideal gas equation assumes that the gas molecules take up no space, while in fact, with real gasses, they do. Adding z (compressibility factor) to the density equation corrects for the difference between the behavior of the specific real gas and an ideal gas. (The density equation shown is for psia, Degrees R and Pounds per cubic ft.)

There are tables, based on tests, of compressibility for some gasses and also computer programs that can approximate the compressibility factor.

Neles’ “Nelprof” valve sizing and selection software can approximate the compressibility factor of some common gasses using the Van der Waals equation of state.

The Nelson Obert charts (readily found with a Google search) are also often used to determine compressibility factors.)

For most gasses used in industrial processes, and at the pressures and temperatures that they are normally used, for valve sizing purposes, assuming a compressibility factor of 1.0 is usually (but not always) sufficient.

Compressibility Factor

Gas Compressibility Factor (Z) for Control Valve Sizing Purposes

GAS: Carbon Dioxide   **TAG:**

		COND 1	COND 2	COND 3	COND 4
CRITICAL TEMP	**DEG F**	87.76	=========>	=========>	========>
CRITICAL PRESS	**PSIA**	1070.00	=========>	=========>	========>
UPSTREAM TEMP	**DEG F**	150.00			
UPSTREAM PRESS	**PSIA**	90.00			
COMPRESS. FACTOR (Z)		**0.98**			

Reduced Temperature (Tr)	1.114
	Tr within limits
Reduced Pressure (Pr)	0.084
	Pr within limits

Calculating Cv using Z = 1.0 would result in a Cv that is about 1% high.

Here is a screen shot of an Excel worksheet that approximates the compressibility factor based on the critical pressure and temperature of the gas and the operating temperature and pressure.

The worksheet calculation is based on the Nelson Obert Generalized Compressibility Charts. A number of points from the charts have been tabulated, and the worksheet performs two dimensional interpolation between tabulated points to find the compressibility factor for the specific process pressure and temperature points that are entered by the user. As shown, the worksheet is set up for temperature in degrees F and pressure in psia, however there is an area in the worksheet (not shown here) where conversion factors for other engineering units can be substituted.

This worksheet can be downloaded from the “Worksheets” page at:

www.control-valve-application-tools.com

Typical of most gasses at the temperatures and pressures we normally use them, assuming a compressibility factor of 1.0 would cause negligible error in the Cv calculation for carbon dioxide at 90 psia and 150 degrees F.

Compressibility Factor

Gas Compressibility Factor (Z) for Control Valve Sizing Purposes

GAS: **Carbon Dioxide** **TAG:**

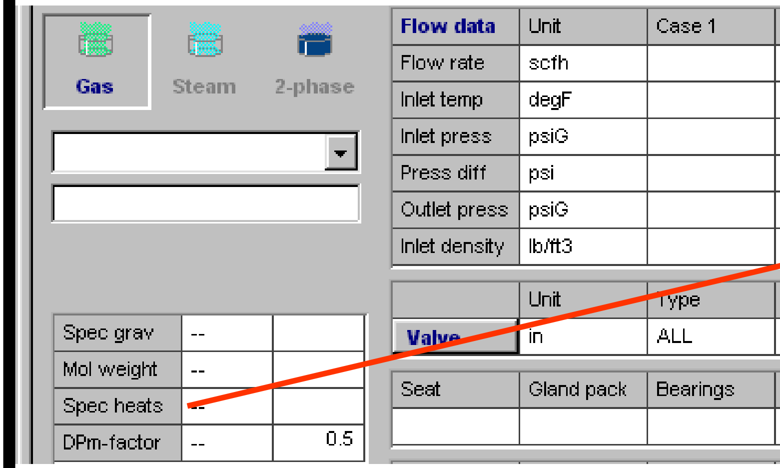
		COND 1	COND 2	COND 3	COND 4
CRITICAL TEMP	**DEG F**	87.76	=========>	=========>	========>
CRITICAL PRESS	**PSIA**	1070.00	=========>	=========>	========>
UPSTREAM TEMP	**DEG F**	100.00			
UPSTREAM PRESS	**PSIA**	800.00			
COMPRESS. FACTOR (Z)		**0.70**			

Reduced Temperature (Tr)	1.022
	Tr within limits
Reduced Pressure (Pr)	0.748
	Pr within limits

Calculating Cv using Z = 1.0 would result in a Cv that is about 20% high.

Occasionally, the compressibility factor will differ enough from 1.0 to have an effect on the valve sizing calculation. In this example we are looking at carbon dioxide very close to its critical temperature. Assuming a compressibility factor of 1.0 in this case where the compressibility factor is really 0.7 would result in a calculated Cv that is about 20% high.

Ratio of Specific Heats



Ratio of specific heats, γ

(Sometimes the letter **k** is used)

A gas property that effects the speed of sound in that gas.

$$v_s = C\sqrt{\frac{\gamma T}{M}}$$

Where:

C = a constant depending on the units used

γ = ratio of specific heats

= c_p/c_v (specific heat @ constant press. / specific heat @ constant vol.)

T = absolute temperature

M = molecular weight

Computer programs ask for the ratio of specific heats, which is one of the factors used in calculating the speed of sound in a particular gas and is therefore part of the choked flow calculation that we will discuss later.

The equation shown above is for an *ideal gas.* In non-ideal gasses there is a slight dependence on pressure, however the above equation is the only one I have seen in literature discussing control valve applications.

C is a constant that will have different values depending on the engineering units being used.

T is the absolute temperature

M is the molecular weight of the gas.

In some places, you may see the letter k used for the ratio of specific heats.

Mach Number, M_n

Describes the flow velocity of gas as a fraction of the sonic velocity of that gas

$$M_n = \frac{v}{v_s}$$

Supersonic, $M_n > 1$

Sonic velocity, $M_n = 1$

Subsonic, $M_n < 1$

Gas velocity is often measured in terms of the Mach Number and valve manufacturers often specify the maximum recommended valve outlet velocity in terms of the Mach Number.

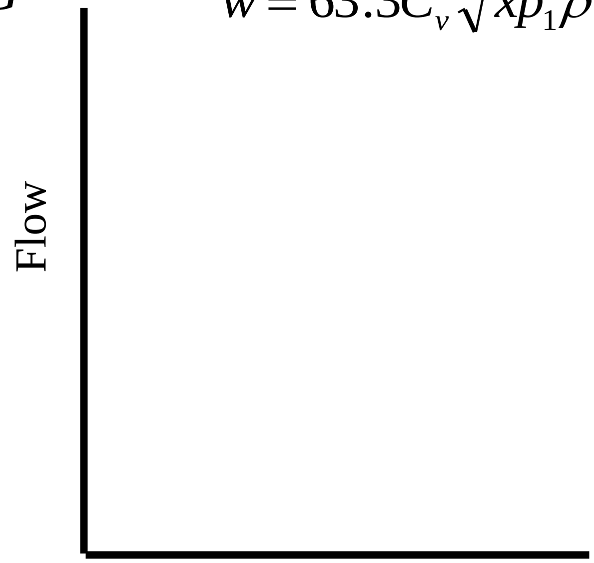
The Mach number is the ratio of the actual velocity to the velocity of sound (sonic velocity) in the same medium. Mach 1 means the flow is at sonic velocity. For flow velocity above sonic velocity, Mach number is greater than 1 and it is referred to as “Supersonic flow.” If the velocity is less than sonic velocity the Mach number is less than 1 and we say the flow is “Subsonic.”

Gas Flow in Control Valves

$$q = C_v \sqrt{\Delta p / G}$$

Water equation with conversions = gas equation (almost)

$$w = 63.3 C_v \sqrt{x p_1 \rho}$$



$$\sqrt{\Delta p / p_1} = \sqrt{x}$$ = Pressure drop ratio

$$\Delta p = x p_1$$

…

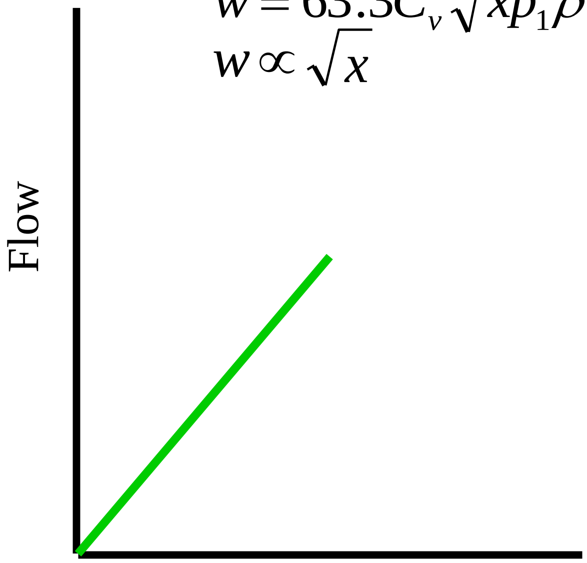
The gas sizing equations are actually very similar to the liquid equations. We will show how the gas flow equations describe the flow of gas in a control valve. We can start with the liquid equation (shown on the left) and make it suitable for gas (almost) by simply converting the units. We convert the flow units from volumetric (gallons per minute) to mass flow units (pounds per hour), and the liquid specific gravity is not appropriate for gasses, so it is converted to density.

Also, though not absolutely required, for the sake of convenience (and we will see later why this helpful) we change the horizontal scale from the square root of pressure drop, Δp, to the square root of *pressure drop ratio,* $\Delta p/p_1$. We then substitute the single character, x, for $\Delta p/p_1$ to make the expression simpler. Making this change, makes the expression "Δp" equal "$x\ p_1$"

The main point here is that the gas sizing equation can be traced directly to the simpler liquid equation. We will see later that one additional change will be needed to the equation to account for choked flow.

Gas Flow in Control Valves

$$w = 63.3 C_v \sqrt{x p_1 \rho}$$
$$w \propto \sqrt{x}$$



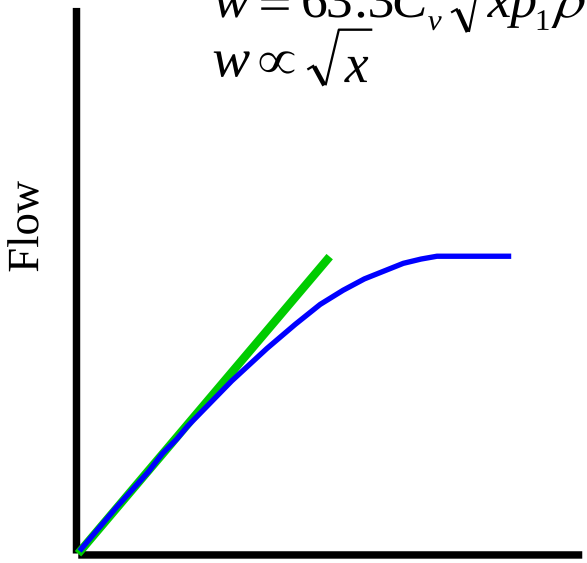
$$\sqrt{\Delta p / p_1} = \sqrt{x} = \text{Pressure drop ratio}$$

Now that we have shown that the gas flow equation is related to the liquid equation, we will clear it off the slide before we continue.

This modification of the liquid equation to make its units suitable for gas tells us that the flow of gas is proportional to pressure drop ratio, and if we graph it. we get a straight line who’s value increases in proportion to pressure drop ratio, x.

Gas Flow in Control Valves

$$w = 63.3 C_v \sqrt{x p_1 \rho}$$
$$w \propto \sqrt{x}$$



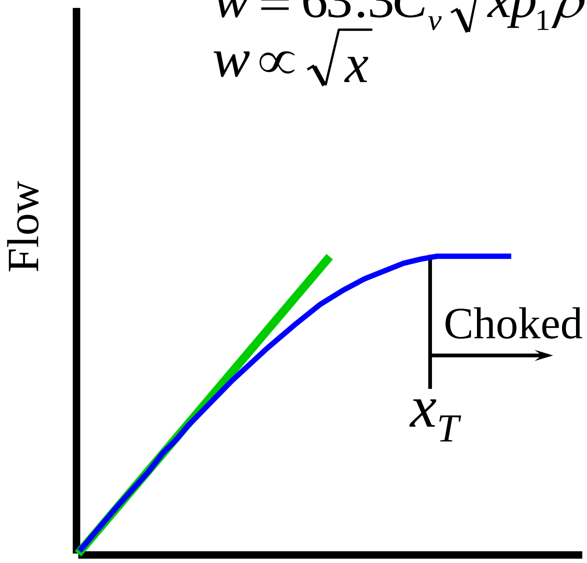
$$\sqrt{\Delta p / p_1} = \sqrt{x}$$ = Pressure drop ratio

As with liquids, the behavior of gas flow in a control valve is not quite as simple as the previous slide indicates.

If we were to conduct a flow test, the actual relationship between flow and pressure drop ratio would be as shown by the curved line, not the straight one. At low pressure drop ratios the flow follows the straight line, but then it deviates more and more until at last, further increases in pressure drop ratio do not yield any additional flow….. Continued on next page.

Gas Flow in Control Valves

$$w = 63.3 C_v \sqrt{x p_1 \rho}$$
$$w \propto \sqrt{x}$$



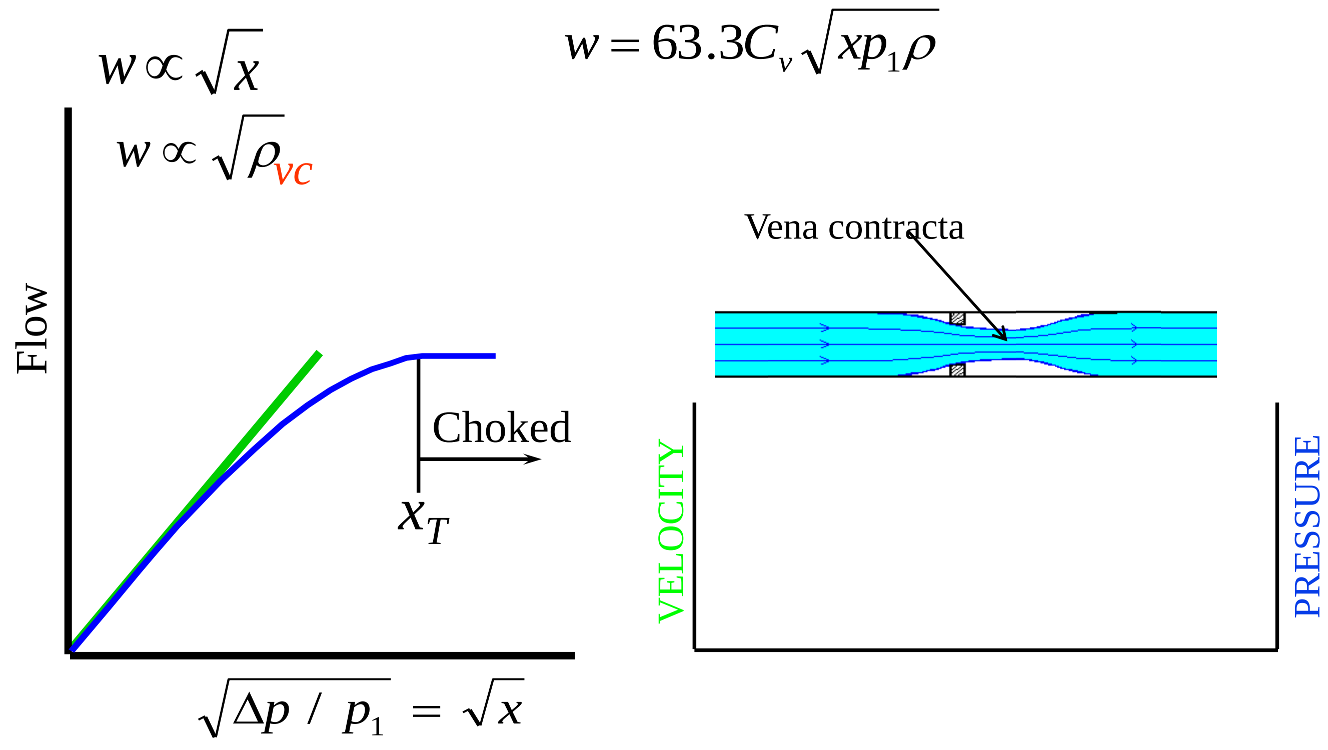
$$\sqrt{\Delta p / p_1} = \sqrt{x}$$ = Pressure drop ratio

x_T = Terminal Pressure Drop Ratio

Continued from last page…. At this point we say that flow has become choked.

Since for gas flow we have chosen to call the horizontal axis the “x” axis instead of the delta p axis, we define the pressure drop ratio at which flow becomes fully choked as the Terminal Pressure Drop ***Ratio***, and give it the symbol X_T

Choked Flow

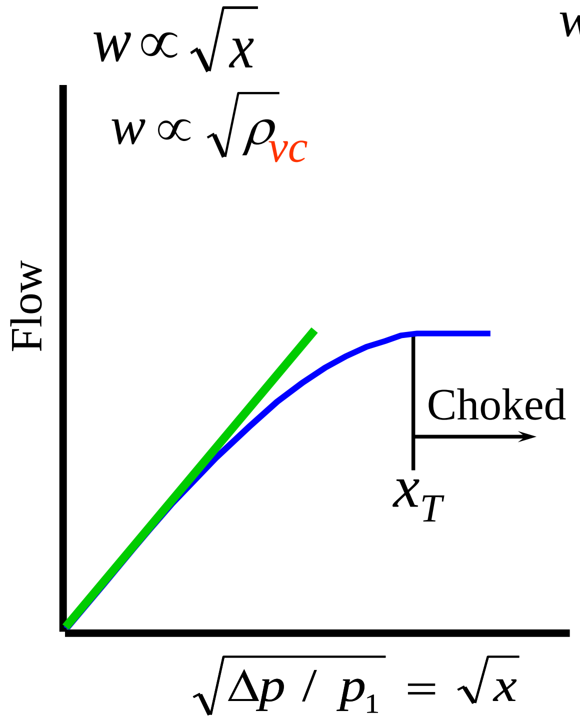


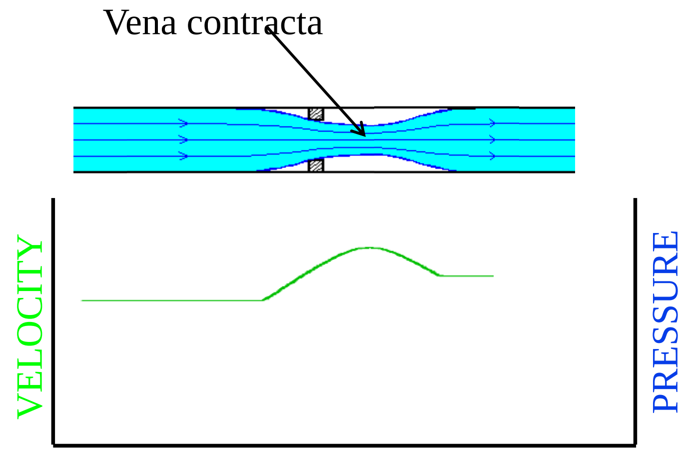
Let’s see what is going on inside the valve to cause this choking of the flow and to give the graph its shape.

At this point, I need to point out that it is actually the density at the vena contracta that influences the flow rate. This is true for both liquid and gas, but with liquids (which are incompressible) we don’t need to make an issue of the fact, because the density at the vena contracta is exactly the same as the density upstream of the valve. Also, with liquids, the density at the vena contracta does not change as the flow rate changes so the only thing we have to consider is that flow (which could be either q (for volumetric flow) or w (for mass flow) increases with pressure drop. With gas flow, the flow rate is proportional to the **pressure drop ratio, x,** but because gas density changes with pressure, flow is also proportional to the density at the vena contracta.

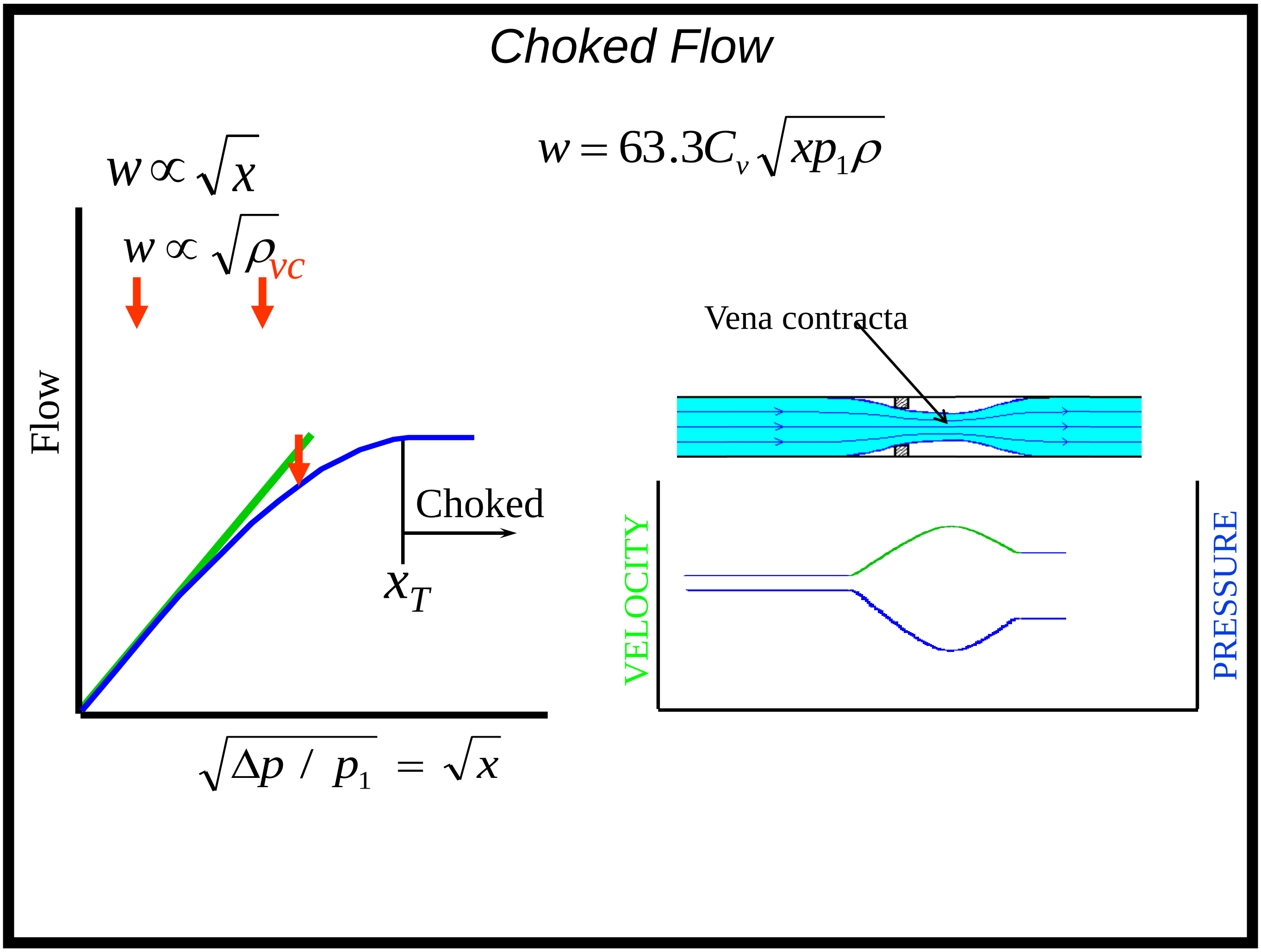
Choked Flow

$$w = 63.3 C_v \sqrt{x p_1 \rho}$$





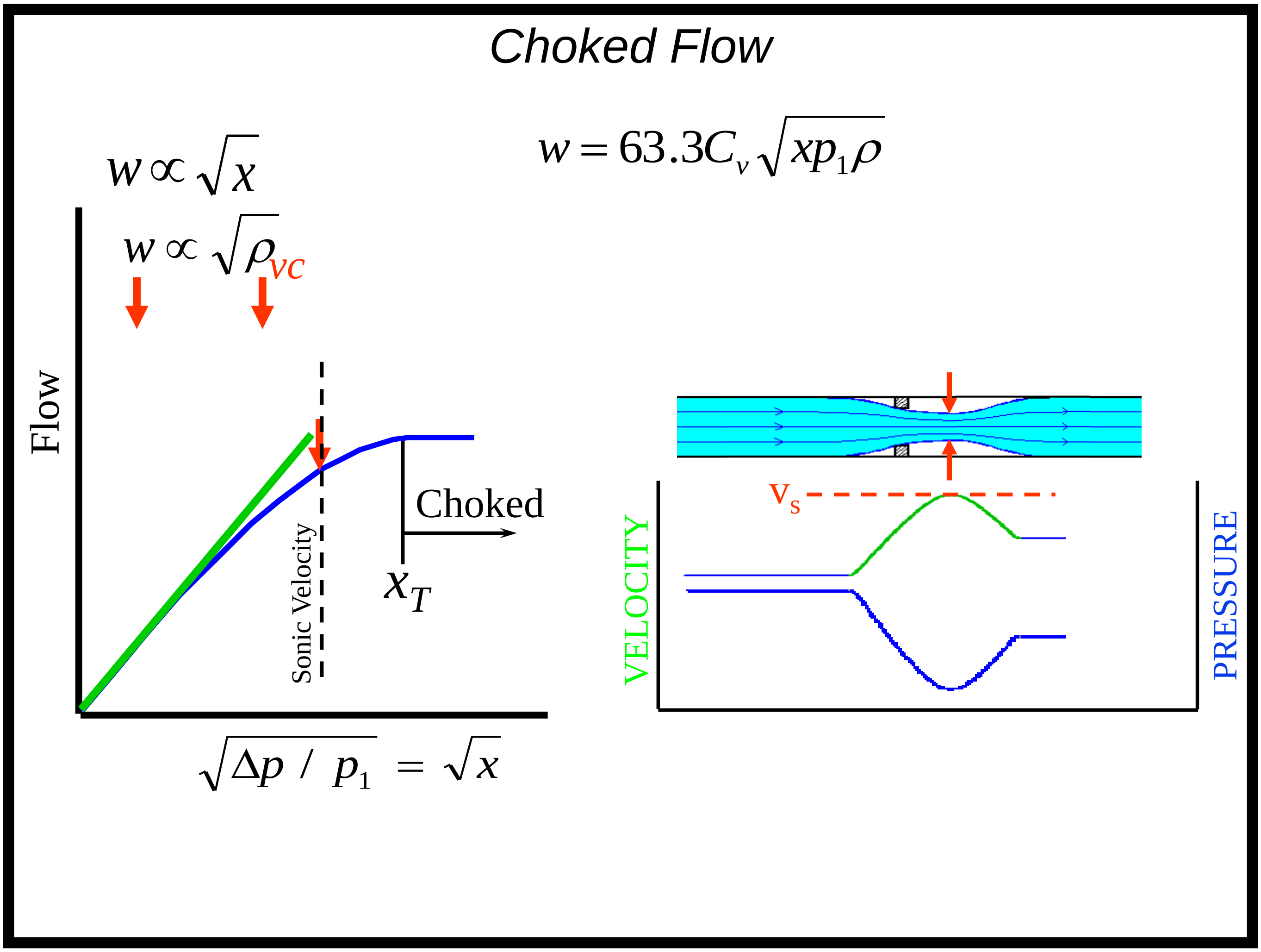
The velocity of the gas flowing through a valve reaches a maximum at the vena contracta.



Due to conservation of energy, as a result of the velocity increase, the pressure decreases to a minimum at the vena contracta. When the pressure decreases the gas becomes less dense.

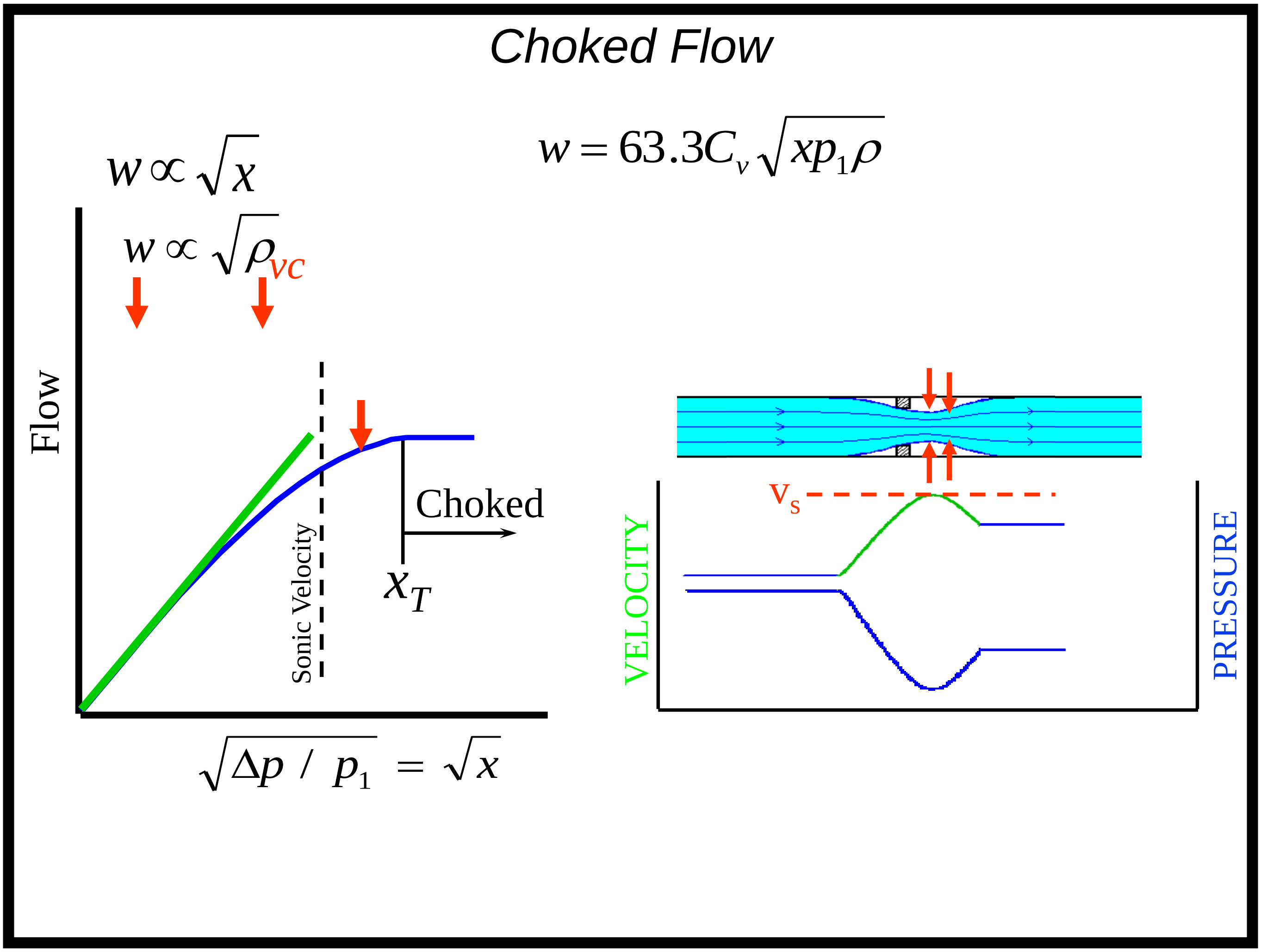
Since flow is proportional to the square root of density at the vena contracta, the decrease in density causes the flow to be less than it would be if gas were not compressible, accounting for flow starting to round off instead of following the straight line.

,



As we continue to increase the pressure drop ratio, the velocity at the vena contracta becomes greater and the pressure becomes less, resulting in an even lower density. Now the flow deviates even more from the straight line that assumes a constant density at the vena contracta as would be the case for a liquid.

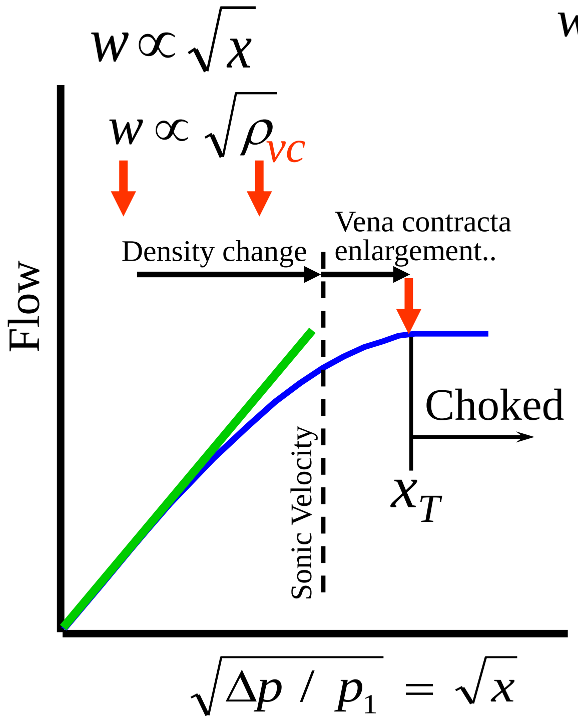
At some point, as the pressure drop ratio is increased and the flow rate increases, the velocity at the vena contracta becomes sonic. Because the vena contracta is downstream of the physical restriction and has a smaller cross-sectional area than that of the physical restriction, even though the velocity has reached the maximum velocity that is possible at a restriction, it is still possible for the flow rate to increase.

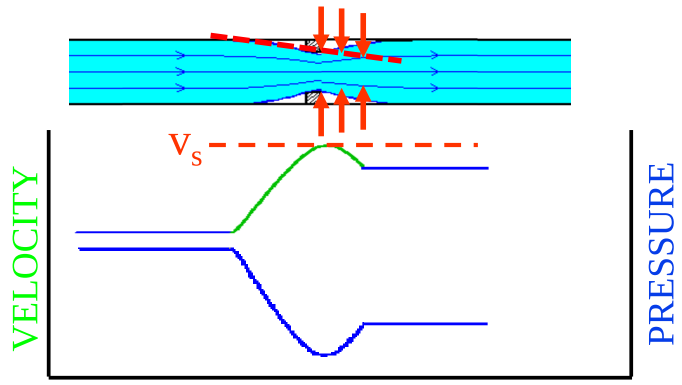


As the pressure drop ratio is increased further the vena contracta starts backing up toward the physical restriction the cross-sectional area of the vena contracta increases, so even though flow is sonic there is still some increase in flow because the area is larger.

Choked Flow

$$w = 63.3 C_v \sqrt{x p_1 \rho}$$



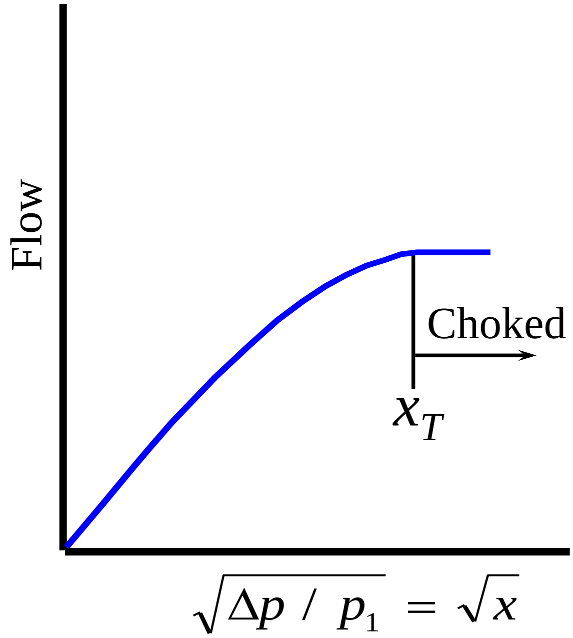


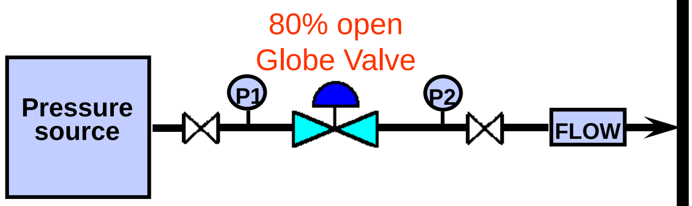
Finally, when the vena contracta backs up to the physical restriction, it can get no larger, and since flow is already sonic, no increase in flow is possible, and flow is fully choked.

Summarizing how the gas flow graph gets its shape:

At vena contracta velocities below sonic, the deviation of the flow curve from a straight line is caused by the density changing. Once sonic velocity is reached, the velocity and pressure at the vena contracta remain constant, but the vena contracta backs up toward the physical restriction, becoming larger and thus allowing flow to still increase. When the vena contracta finally reaches its maximum size (and since the velocity is already at the maximum possible) flow chokes.

Terminal Pressure Drop Ratio, x_T





P1 (PSIA)	100	200	1000
Δ P Choked	70	140	700
x_T	**0.7**	**0.7**	**0.7**

$$x_T = \Delta p \text{ Choked} / p_1$$

Now let's see why we plot flow against pressure drop ratio instead of just pressure drop.

Let's run three flow tests, using a typical GLOBE valve, with the inlet pressure, p_1, at first 100 psia, then at 200 psia, and finally at 1,000 psia.

With p1 at 100 psia and starting with delta p at zero and gradually increasing it we would find that flow would choke when the pressure drop was 70 psi.

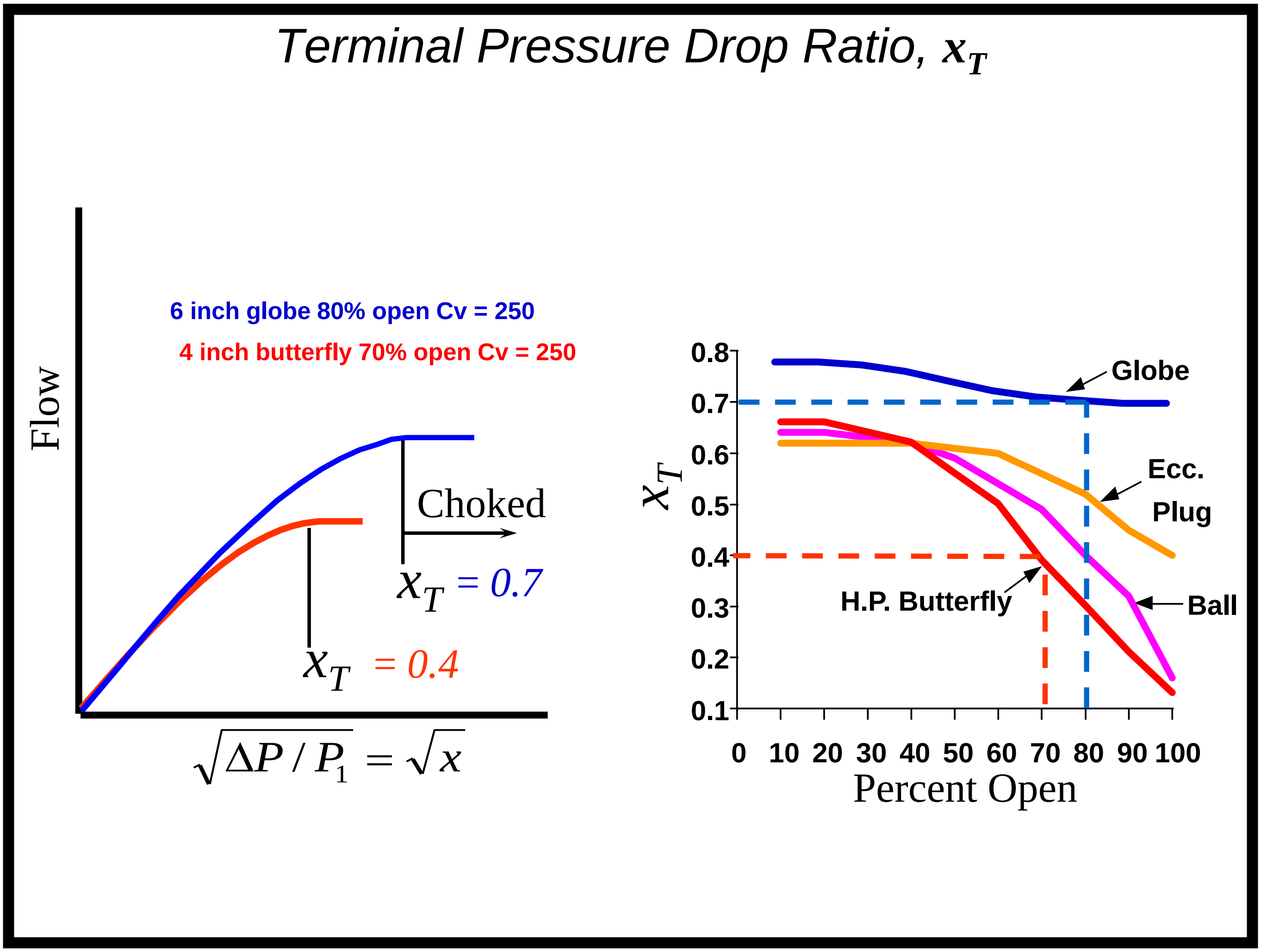
Repeating with p_1 = 200 psia, flow would choke at a pressure drop of 140 psi.

Finally, repeating the test with p_1 = 1000 psia, flow would choke at a pressure drop of 700 psi.

Now, if we calculate the value of x_T for each of the tests (since x is delta p divided by p_1, x_T, the choked value of x, is delta p Choked / p1) we see something very interesting. x_T turns out to be 0.7 in each case.

The point here is, that for a particular style of control valve (in this case a globe valve) the pressure drop ratio at which flow becomes choked is a constant. Knowing that the terminal pressure drop ratio for a typical globe valve is 0.7 we can now predict that if the inlet pressure was 300 psi, flow would choke at 210 psi pressure drop (300 X 0.7 = 210).

It is necessary to know at what pressure drop flow will choke in order to properly size a control valve.

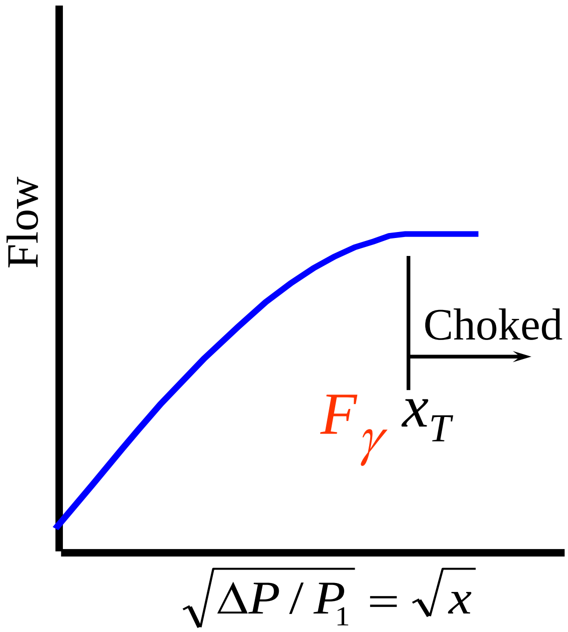


Different valve styles have different values of x_T , and for each valve type, x_T also varies with valve opening.

Valve manufacturers test their valves for x_T and then publish the results, making it possible to predict the point at which flow will choke and therefore properly size control valves.

The upper line on the left-hand graph represents the flow through a globe control valve, where x_T is 0.7. (That is flow will choke when the pressure drop is 70% of P1 (the upstream pressure). As an example, this upper line could represent a 6 inch globe valve at 80% open which would have a Cv of about **250.** A 4 inch butterfly valve operating at around 70% open would also have a Cv of about **250**. Although both valves have the same flow capacity (Cv), the graph of the butterfly valve (lower line on the left-hand graph) looks quite different. That is because it has an x_T of 0.4, meaning that flow chokes when the pressure drop is 40% of P_1. At lower pressure drop ratios, the flow is the same through both valves, but as the pressure drop ratio increases, the flow in the butterfly valve starts aiming toward choked flow before the flow in the globe valve does. Understanding this will help you understand why a sizing calculation may show that, with all the flow conditions the same, one style of valve will need a larger Cv than is required of a different style of valve.

F_γ *Corrects Choked Flow for Gasses Other than Air*



For gasses other than air, flow chokes when x equals $F_\gamma x_T$

$$F_\gamma = \frac{\gamma}{1.4}$$

Ratio of specific heats of the flowing gas

Ratio of specific heats of air…

$F\gamma$ is the Ratio of Specific Heats ***Factor*** of the gas and compensates for gasses with sonic velocities that differ from that of air…

Before we conclude by showing how the ISA/IEC control valve sizing equations accurately predict both the shape of the gas flow vs. pressure drop ratio curve and the point at which flow chokes, we need to introduce one more concept, that of the "Ratio of Specific Heats Factor," F_γ. The valve manufacturer's published values of x_T are based on choked flow tests using air as the test medium. Many gasses other than air have sonic velocities that differ from that of air, so to compensate for the sonic velocity of those gasses, the published value of x_T is multiplied by the Ratio of Specific Heats Factor, F_γ (F sub gamma), of the gas for which the valve is being sized. The Ratio of Specific Heats Factor is calculated by dividing the Ratio of Specific Heats, γ, by the Ratio of Specific Heats of air, which is 1.4.

$$F_\gamma = \gamma / 1.4$$

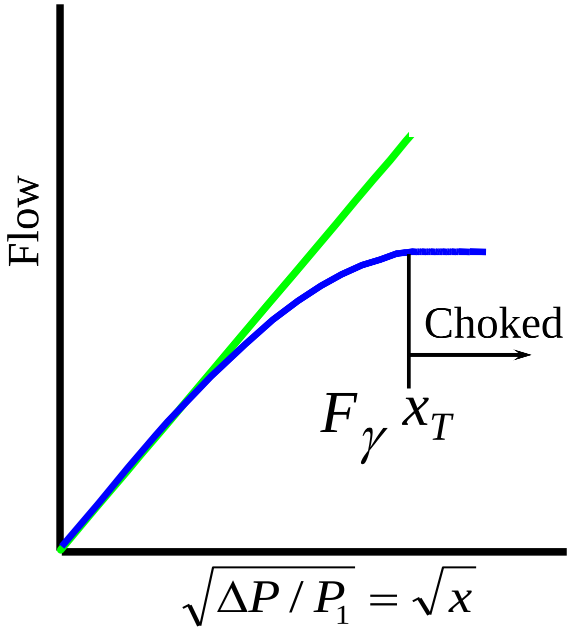
F_γ of air reduces to 1.0.

Most tables of gas properties include values of the Ratio of Specific Heats.

Expansion Factor, Y, Gives Graph its Correct Shape

by correcting for density change and vena contracta enlargement

$$W = 63.3\, Y\, C_v \sqrt{x P_1 \rho_1}$$



Y: Expansion Factor

$$Y = 1 - \frac{x}{3 F_\gamma x_T}$$

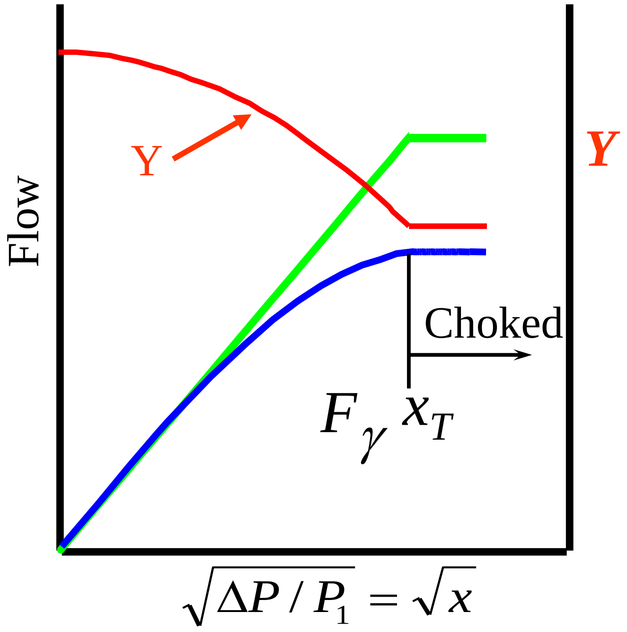
Because it is not easy to determine what the density of the gas is at the vena contracta (which varies with valve style, valve opening and flow rate), the ISA/IEC control valve gas equation uses the (easy to determine) upstream density (rho sub 1). Recall that earlier, when we discussed why the actual flow graph has the shape it does we said that the first part of the curved portion was the result of changes in density at the vena contracta, but the second portion (when flow at the vena contracta was sonic and density remains constant) was due to enlargement of the vena contracta as it backs up to the physical restriction. So, even if we could determine the density at the vena contracta, that wouldn't be enough to give the flow graph its correct shape.

Because the equation resulting from using rho sub 1 (density upstream of the valve) would graph as a straight line, we include an ***Expansion Factor (symbol Y)*** to the equation which corrects the calculated flow (and the graph) for both changes in density at the vena contracta and for vena contracta enlargement. The equation shown here for Y is based on experimental observation of what actually happens.

Expansion Factor, Y, Gives Graph its Correct Shape

by correcting for density change and vena contracta enlargement

$$W = 63.3\, Y\, C_v \sqrt{x P_1 \rho_1}$$



Y: Expansion Factor

$$Y = 1 - \frac{x}{3 F_\gamma x_T}$$

$$x_{max} = F_\gamma x_T$$

…

Y is a function of x (how far we have increased the pressure drop ratio from zero toward $F_\gamma\, x_T$, the pressure drop ratio at which flow will become choked). When plotted on a square root scale, the graph of Y looks like the line labeled “Y” on the graph.

Multiplying the straight line (the flow graph if there was no density change and choking) by Y results in the actual flow graph. It is important to limit the value of x used in sizing or flow calculations to the choked value ($F_\gamma\, x_T$). otherwise Y would decrease below 0.67 and the predicted flow, after reaching a maximum value at $x = F_\gamma\, x_T$ would then decrease, which we know is not the case.

ISA Gas Flow Equations

$$C_V = \frac{W}{63.3Y\sqrt{xP_1\rho_1}}$$

$$C_V = \frac{Q}{7320P_1Y}\sqrt{\frac{MT_1Z}{x}}$$

where:

$$Y = 1 - \frac{x}{3F_\gamma x_T} \qquad F_\gamma = \gamma / 1.4 \qquad x = \Delta P / P_1$$

and

$$x(max) = F_\gamma\, x_T$$ (The value of x at which flow becomes choked.)

The ISA/IEC gas flow equations can take several forms. The two most common forms are shown here.

The top equation is the one we have been using so far, but rearranged to solve for Cv instead of W. This form is appropriate for gasses and vapors (including steam) where flow is in mass flow units (Pounds per hour) and the upstream density is known.

The second equation, which is simply the first equation, with appropriate unit conversions for flow in volumetric units (scfh) and the density calculated from the molecular weight, absolute temperature, absolute pressure and compressibility factor.

For many years the ISA equations were published with flow as the dependent variable. (Which is the form we used in our discussion of gas flow since it was our purpose to understand how the flow of gas behaves as it goes through a control valve.) At one point the Standards published the equations with Cv as the dependent variable as shown in the figure above, but the current version of the Standards has reverted to presenting the equations with flow as the dependent variable, but pointing out that they can be rearranged for Cv or pressure drop ratio.

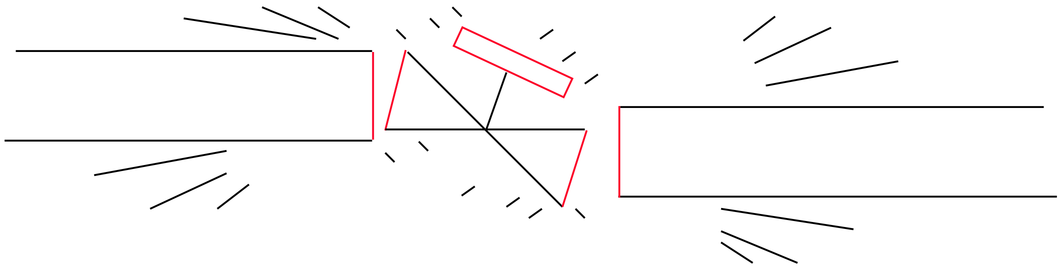
There is also a more widely distributed equation for gas flow that includes the gas specific gravity instead of the molecular weight in the equation in the figure, but that equation was not included in the current version if the IEC/ISA standards.

That equation is:

$$C_V = \frac{Q}{1360P_1Y}\sqrt{\frac{G_gT_1Z}{x}}$$

Valve Noise Concerns

- Hearing Damage - OSHA, insurance
- Community
- Valve and Equipment Damage
- Result --> Typical Valve Noise Specification is 85-95 dBA



We are concerned about valve noise for the reasons listed above.

Noise = Unwanted Sound

Topics:

- Basic acoustics
- Aerodynamic noise generation in valves
- Aerodynamic noise abatement methods

We will begin with a short discussion of basic acoustics. The basic acoustics pages apply to both aerodynamic and hydrodynamic noise.

The rest of this section on Noise generation in valves and noise abatement methods apply to aerodynamic noise (noise generated by gas and steam flow) since the section on Liquid Flow in Control Valves covers hydrodynamic noise.

Basic Acoustics

Sound Pressure Range:

0.0002 microbar to 500 microbar

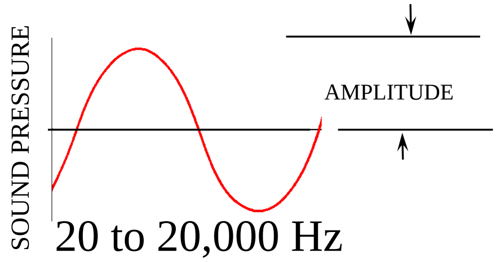
(2,500,000 : 1 ratio)

Sound Pressure **Level** (SPL):

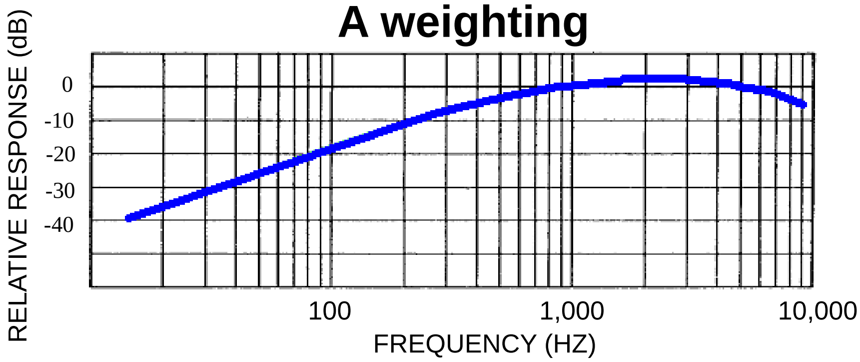
SPL = 20 log (Ps/0.0002) dB (decibels)

(8000 : 1 ratio)

Properties of sound



A weighting



Sound Pressure Level (SPL) measured with an "A" filter has units of **dB(A)**

What we perceive as sound is the effect of minute vibrations in the air pressure surrounding our ears. These vibrations have both frequency and amplitude.

The amplitude of these pressure fluctuations (called the sound pressure) is very small and falls in the range of 2 times 10 to the -4 microbar (the smallest sound pressure the average young human can detect also referred to as the "threshold of hearing") and 500 microbar above which we feel pain.

Because our ears respond in a logarithmic manner to changes in sound intensity we do not normally measure sound in units of sound pressure, but on a logarithmic scale based on the ratio of the sound pressure we are measuring to the sound pressure that represents the threshold of hearing. The units of this scale are decibels (dB) and we refer to measurements made in dB as Sound Pressure <u>Level</u> (as opposed to Sound Pressure).

The average young human being is sensitive to 20 to 20,000 Hz. but we are not as sensitive to all frequencies. We are the most sensitive to about 2,000 Hz. and the sensitivity of our hearing falls off at lower and higher frequencies. Because our main concern over noise is its effect on people, we put a filter in our noise meters that makes them have approximately the same frequency response as our ears. This is called the "A weighting" filter. Noise measured in decibles (dB) on a noise meter with the "A" filter is referred to as being measured in dB(A) (decibels on the "A" scale).

Combining Noise Levels

Logarithmic addition on noise sources

$$L_{total} = 10\log\left[10^{\left[L_1/10\right]} + 10^{\left[L_2/10\right]} + \ldots + 10^{\left[L_n/10\right]}\right]$$

Based on difference between two levels

DIFFERENCE	ADD TO HIGHEST
0 - 1 dB(A)	3 dB(A)
2 - 3 dB(A)	2 dB(A)
4 - 8 dB(A)	1 dB(A)
9 or more v	0 dB(A)

The noise generated by two nearby sources add logarithmically. Two 80 dB(A) valves generate a sound pressure level of 83 dB(A). Not 160 dB(A).

The formula at the top of the page shows how to add noise source.

The table above gives us a simple method of adding noises.

Two sources whose noise is equal, or at most 1 dB(A) different give a total noise of 3 dB(A) more that the loudest of the two sources.

When two sources differ by 9 dB(A) or more, then the total noise is simply the largest of the two. The lower of the two does not contribute significantly to the overall noise level.

Sound Comparisons

Decibels	Example
130	Rock band
125	Threshold of pain
95	Power mower (3 feet)
70	Vacuum cleaner (10 feet)
65	Cocktail party (second drink)
20	Electric clock (at 3 a.m.)
0	Threshold of hearing

Increase in SPL	Human Subjective Response
3 dB(A)	Just perceptible
5 dB(A)	Clearly noticeable
10 dB(A)	Twice as loud

The top table lists some common sounds and their approximate sound pressure levels.

The lower table describes how humans perceive changes in sound pressure level.

OSHA Noise Exposure Limits

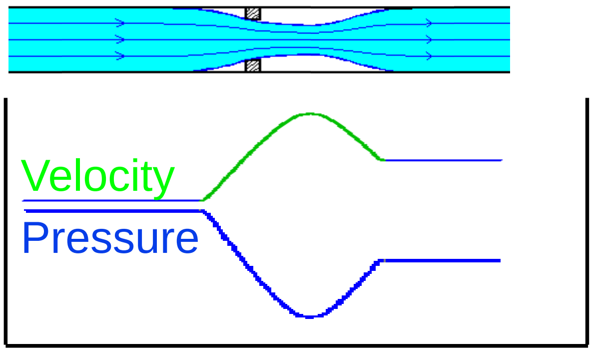
Duration of Exposure (Hours)	SPL (dB(A))
8	90
4	95
2	100
1	105
0.5	110
0.25	115

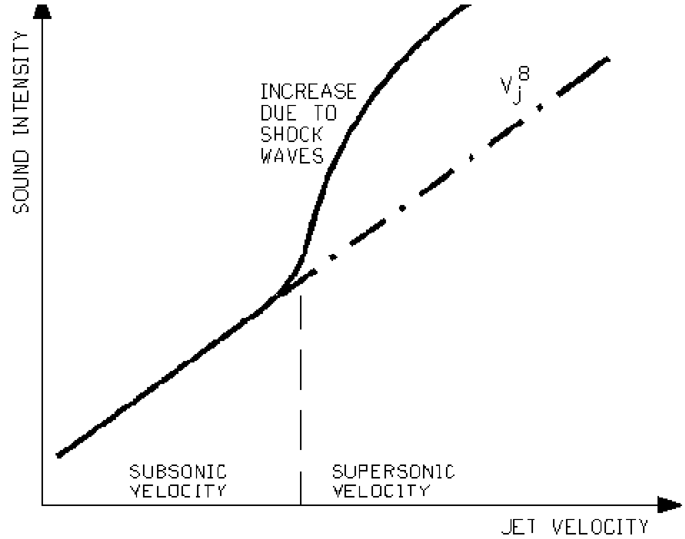
OSHA regulations define the maximum noise levels that workers can be exposed to, depending on how long the worker is exposed to the noise.

If workers are exposed to greater than 85 dB(A), they must be provided with hearing protection and an ongoing program of evaluation for damage to their hearing.

Most industrial plants specify that the maximum noise generated by process equipment (such as control valves) not exceed 85 or 90 dB(A).

Aerodynamic Noise Generation





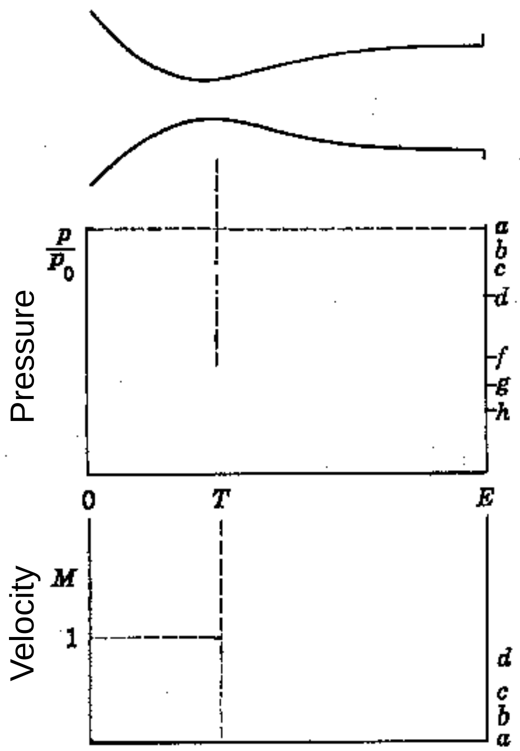
Sudden expansion and compression

When gas flows past the restriction in a control valve the velocity increases and reaches a maximum at the vena contracta. Because of the law of conservation of energy, when the velocity increases, the pressure must decrease. When the pressure decreases, the gas expands.

Any particular volume of gas, as it passes the vena contracta will expand rapidly, and then as it passes the vena contracta and the pressure increases re-compresses. This rapid expansion and compression causes sever turbulence which generates noise. The sound intensity is strongly related to the peak velocity. At subsonic velocities, the sound intensity if proportional to the jet velocity raised the the eighth power. At supersonic velocities the relationship between velocity and intensity is even greater.

We will see later that one method of reducing aerodynamic noise if to minimize the velocity in the valve.

Shock Waves



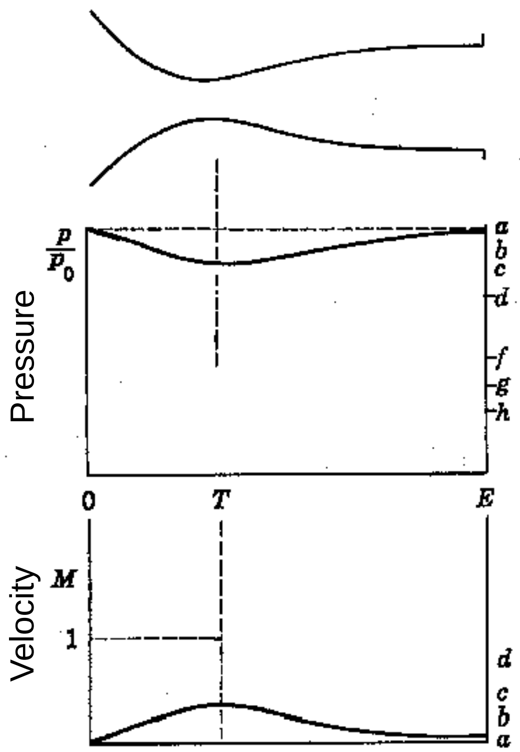
Shock waves caused by supersonic flow are a source of very high noise levels.

The top figure represents a venturi that has been installed in a pipe.

The middle figure is a graph of pressure at different points in the venturi at various flow rates.

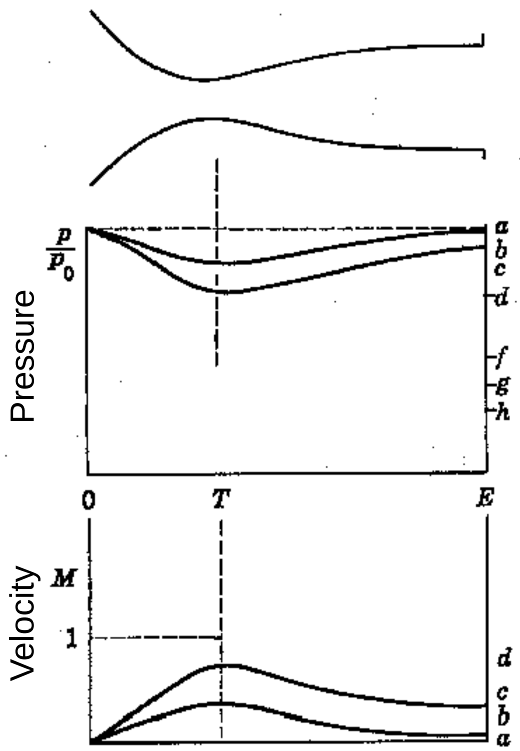
The lower figure is a graph of velocity within the venturi. The "y" axis of the velocity graph has units of Mach number, where a Mach number of 1 means that the flow is at the sonic velocity. Mach 1 is the highest velocity that is physically possible at the point of greatest restriction. (It is, however, possible for the velocity to be greater than Mach 1 downstream of the restriction.)

Shock Waves



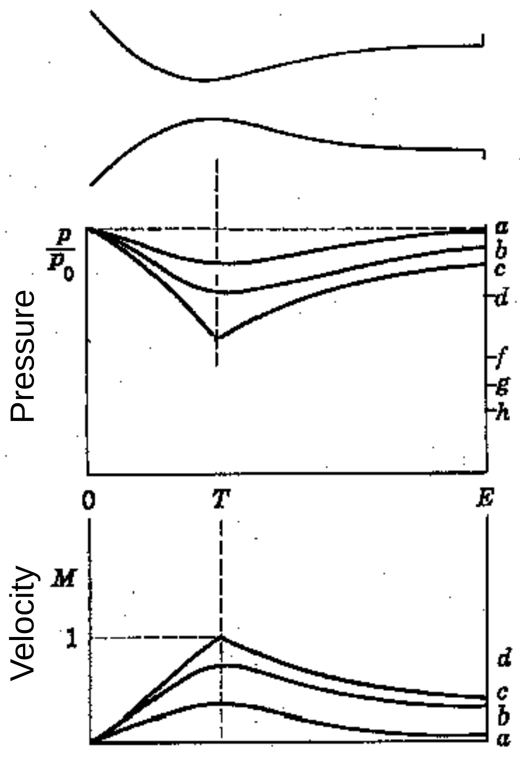
If the pressure downstream of the venturi was just slightly below the upstream pressure (say at Point “a”) there would be a small flow of gas through the venturi. At the throat of the venturi the velocity would reach a small peak and and the pressure would have a small dip, reaching a minimum at the throat of the venturi.

Shock Waves



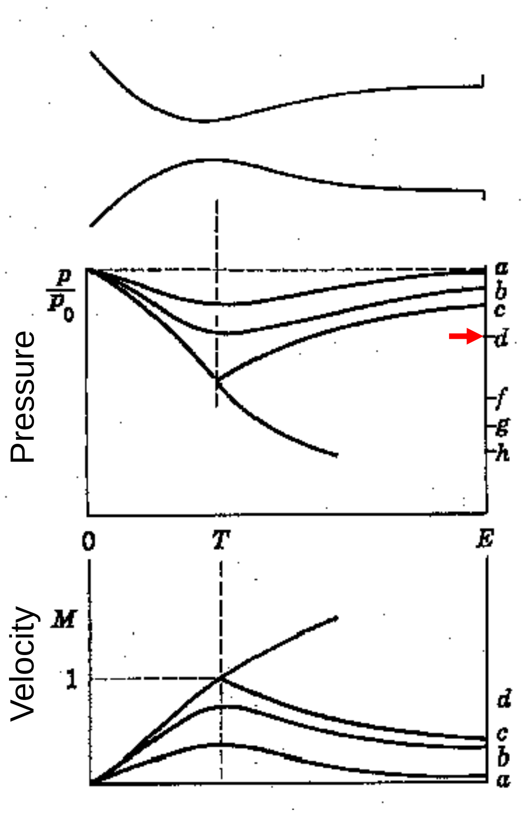
If the downstream pressure were lowered to Point “b” the flow rate would increase and the velocity at the throat would have a higher peak and the pressure a larger dip.

Shock Waves



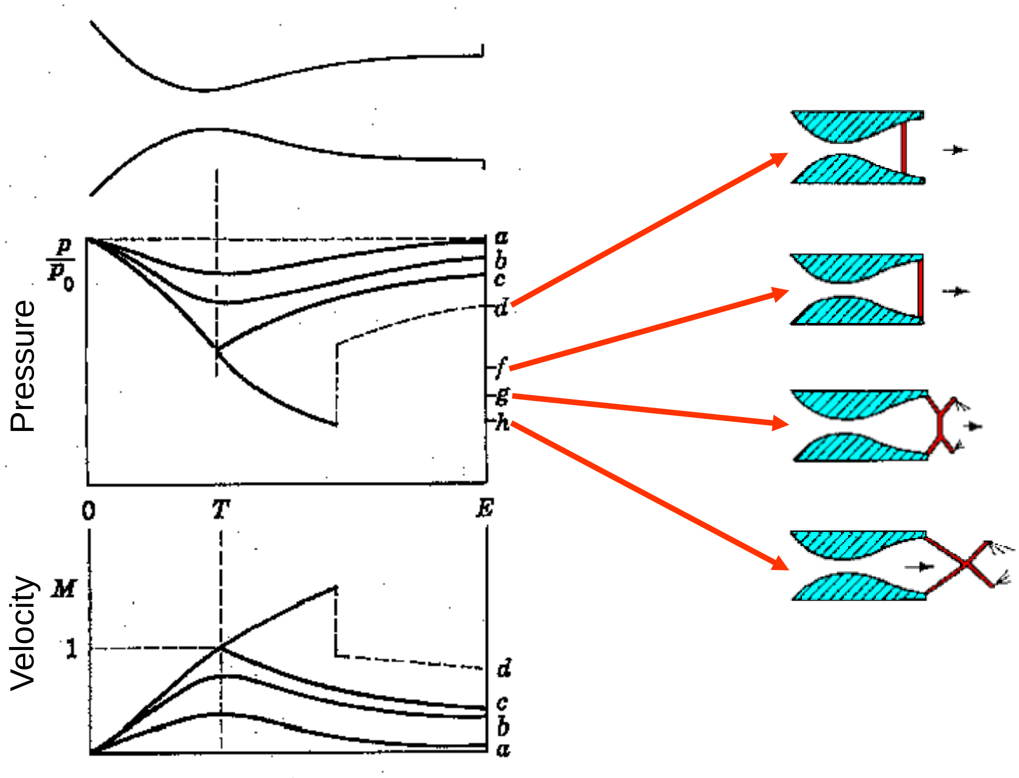
Lowering the downstream pressure to Point “c” would increase to flow to the point where the velocity at the throat just becomes sonic (Mach 1).

Shock Waves



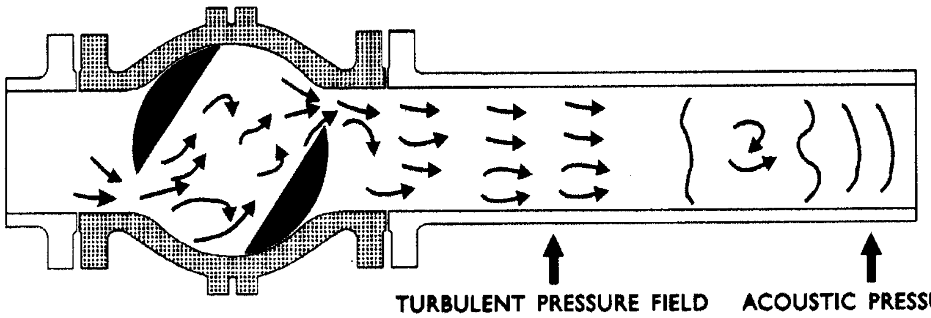
Now if the downstream pressure is lowered further to Point “d” the flow does not increase any more because the velocity at the throat is already Mach 1 and it is physically impossible for it to be any greater. However, the velocity downstream of the throat becomes supersonic. As the velocity increases, the pressure also continues to decrease. (Continued on next page.)

Shock Waves



(Continued from previous page.) Somewhere downstream of the throat the velocity suddenly drops to below Mach 1 and at the same time the pressure suddenly increases. This sudden change in velocity and pressure takes place over a space the width of only a few gas molecules and is called a shock wave. Because of the sudden change in pressure and density in a very small space, severe turbulence and high noise levels are generated. Further lowering of the downstream pressure causes the shock wave to move downstream. Shock waves tend to attach to physical boundaries and take on complex and interacting shapes, creating even greater turbulence.

Valve Noise Generation



- Sudden expansion and compression at vena contracta→Severe turbulence
- Supersonic velocity downstream of vena contracta → Shock waves and **very** severe turbulence
- Turbulence generates sound waves
- Sound waves propagate down the pipe and through pipe wall..

This summarizes the generation of aerodynamic noise in a control valve.

The sudden expansion and compression at the vena contracta generates severe turbulence.

If the pressure drop is high enough, supersonic flow downstream of the vena contracta generates shock waves, and these cause very severe turbulence.

The turbulence generates sound waves which propagate down the pipe, with some of the sound coming through the pipe wall where we hear it.

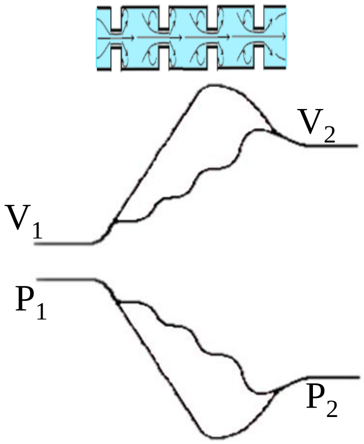
Valve Noise Reduction Strategies

- Source Control
- Path Control

The two strategies for reducing noise are:

Source control, that is doing something to the valve to make it less noisy, and path control, that is doing something to prevent the noise from reaching the people who would be bothered by it.

Source Control Concepts



PRESSURE DROP STAGING

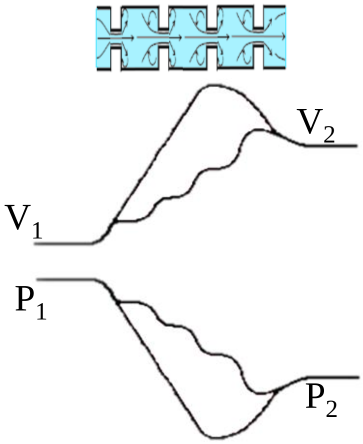
FLOW DIVISION

There are two concepts that are used for source control of aerodynamic noise:

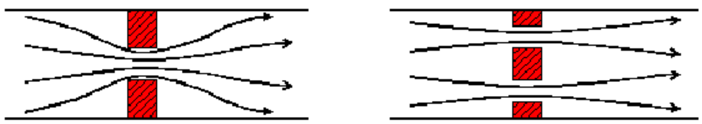
The first method of source control of aerodynamic noise is called “Pressure drop staging.”

Pressure drop staging means that instead of taking the entire pressure drop in one step, the pressure drop is divided up into several steps. When the pressure drop is taken in more than one step, the individual velocity peaks are smaller than the velocity peak that would result from a single stage pressure drop. Since noise is proportional to velocity to the sixth power, small reductions in velocity can have a large effect on reducing the noise.

Source Control Concepts



PRESSURE DROP STAGING



SPL_1 $SPL_2 = SPL_1 – 3dB$

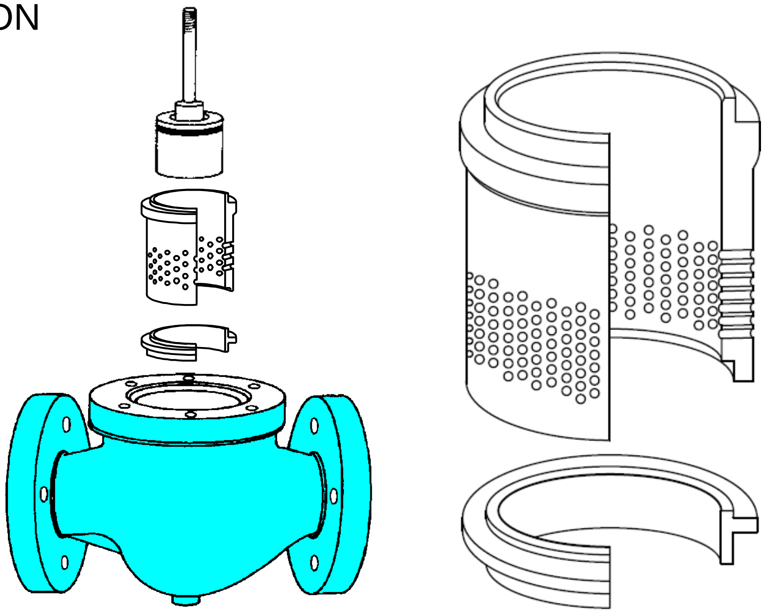
FLOW DIVISION

The second method of source control of aerodynamic noise is called “Flow division.”

Flow division means that instead of having the flow pass through a single opening, the flow is divided up so that it passes through several openings. Every time you double the number of openings, you reduce the noise by 3 dB(A). The reduction is due mainly to the fact that the smaller openings shift the noise to a higher frequency. The higher frequencies are attenuated more as the sound passes through the pipe wall than are the lower frequencies. Also, because the human ear (and the “A” weighting curve on sound meters) attenuate higher frequencies ,both the measured and perceived noise is decreased.

Globe Valve – One Stage

FLOW DIVISION



Small Holes

10-15 dBA attenuation

This is an example of how the principle of flow division can be applied to globe control valves.

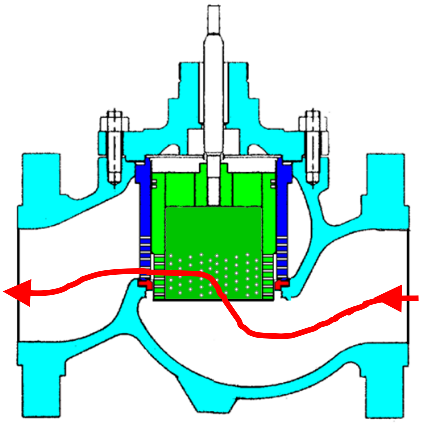
This is a cage with many small holes drilled in it, which divides the flow into many small flows and is the design used by Masoneilan and Hammel Dahl in their one stage noise trim.

Another valve manufcturer’s solution to the one stage aerodynamic noise reduction trim is to divide the flow using vertical slots instead of drilled holes.

Tests done by Neles indicate that both the drilled hole cage and the slotted cage give the same degree of nose reduction.

Globe Valve – Two Stage

FLOW DIVISION
and
PRESSURE DROP
STAGING

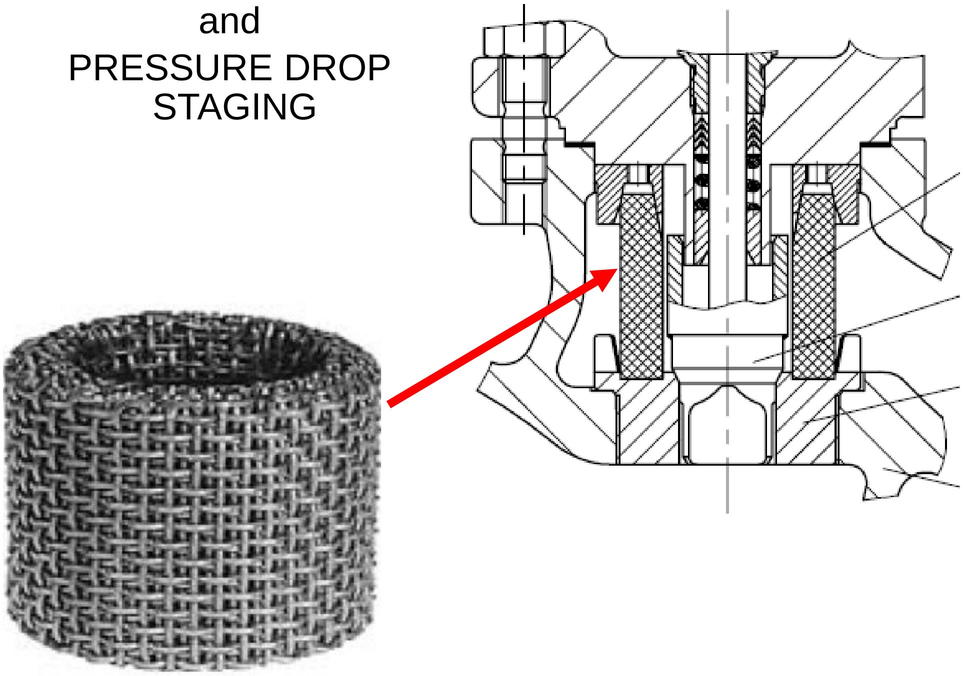


15-25 dBA attenuation

This is a two-stage globe valve that combines flow division and pressure drop staging. Flow enters from the right side of the valve, takes one stage of pressure drop going through the small holes in the lower part of the plug, expands in the space between the plug and the cage, then takes a second pressure drop as it goes through the small holes in the cage.

Globe Valve – Multi Stage

FLOW DIVISION
and
PRESSURE DROP
STAGING

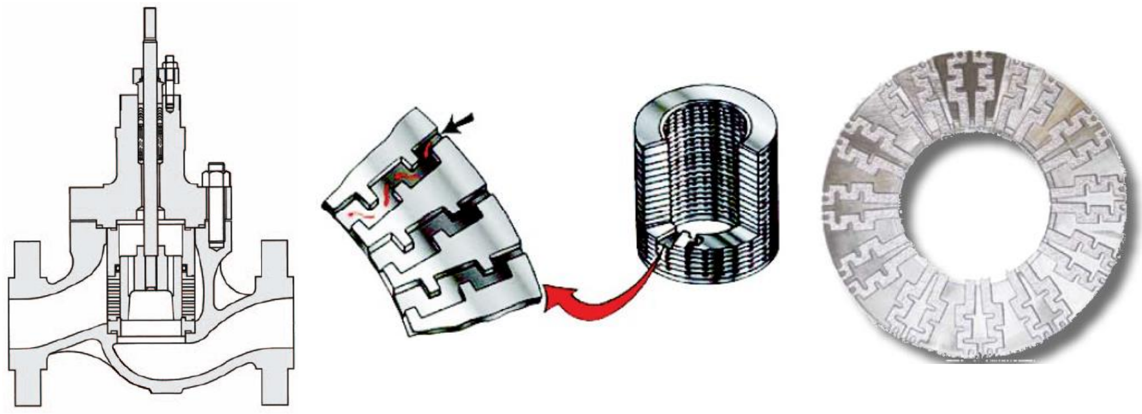


15-25 dBA attenuation

This is a multiple stage flow divider that one manfacturer can add to provide noise reduction to an otherwise standard valve.

Globe Valve – Multi Stage

FLOW DIVISION
and
PRESSURE DROP
STAGING



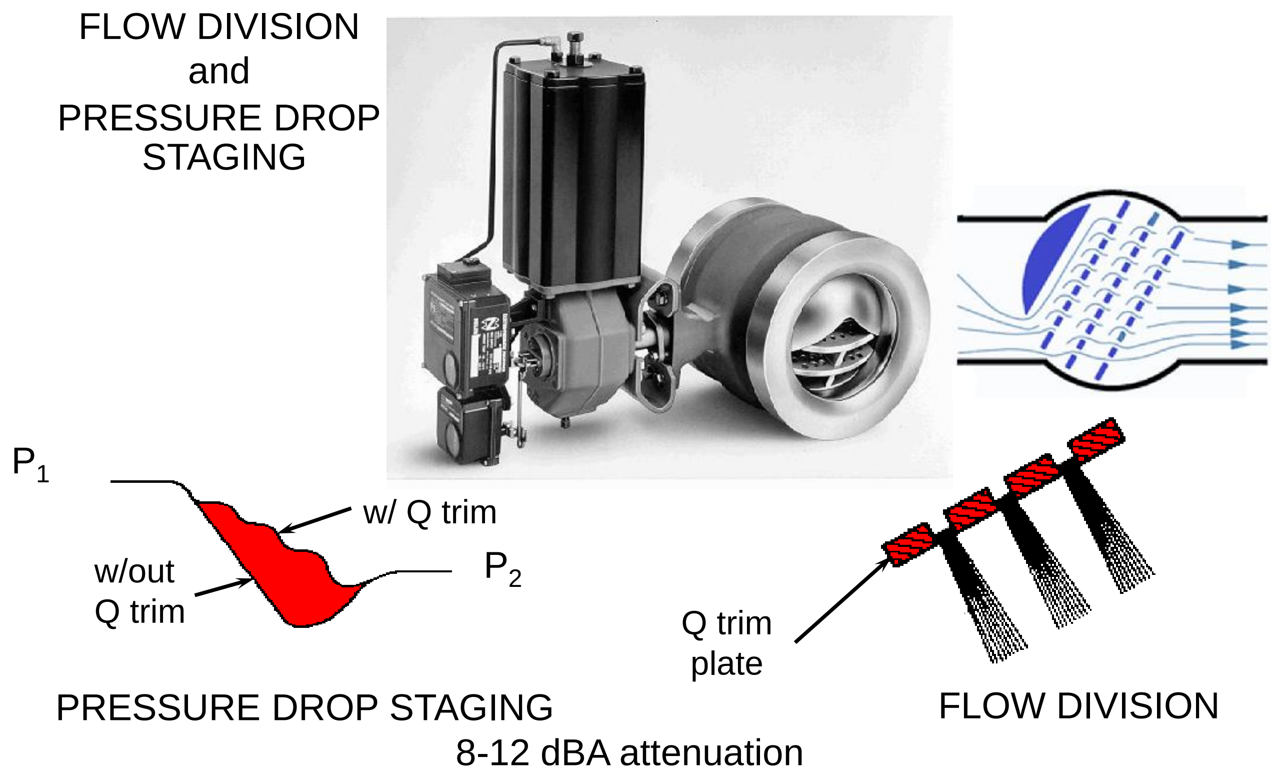
30+ dBA attenuation

This is the design used by the the Koso Hammel Dahl Vector trim.

The design is similar to a cage guided globe valve, except the cage is made of a stack of disks. Each disk has a pattern of multiple, multi turn channels etched into it. The multiple channels provide flow division, and the multiple turns in the channels provide pressure drop staging. It is possible to have a number of changes of direction, each taking a small portion of the overall pressure drop.

This design is extremely effective in reducing aerodynamic noise. It is also a very expensive valve.

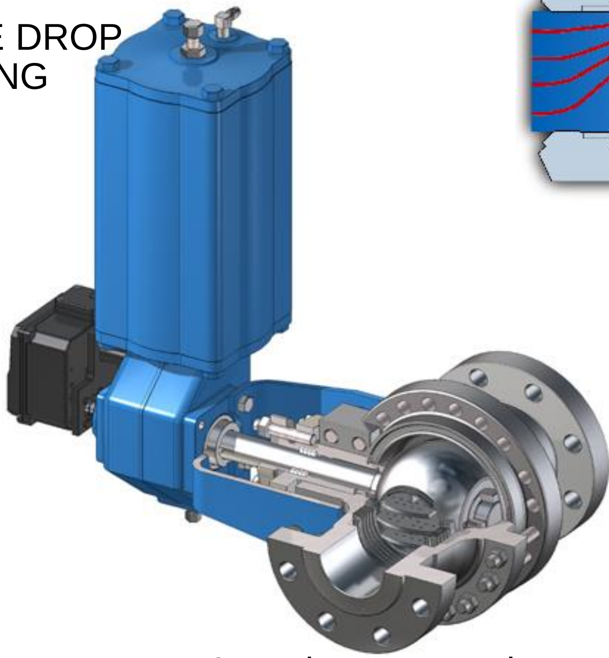
Segment Ball Valve With Q Trim

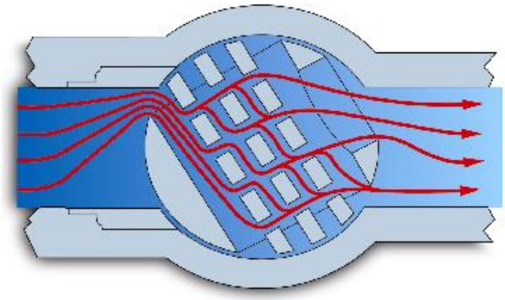


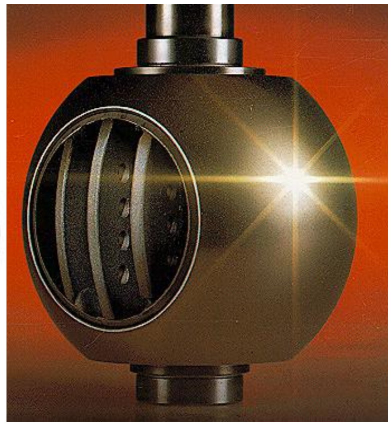
This is a segment ball valve with a special “Q Trim” that incorporates both pressure drop staging and flow division for aerodynamic noise reduction. There are three metal plates inside the ball segment with multiple holes drilled in them. The overall valve pressure drop is taken in four stages, and in the last three pressure drop stages the pressure drop is taken through the multiple holes which provide flow division.

Full Ball Valve With Q Trim

FLOW DIVISION
and
PRESSURE DROP
STAGING





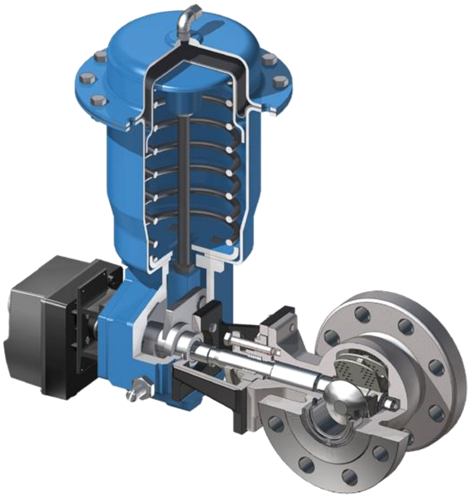


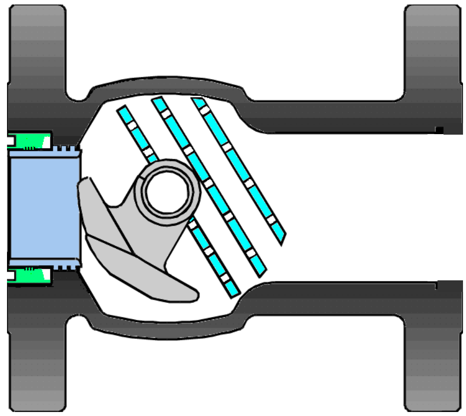
10-15 dBA attenuation

This pictures are various depictions of the multiple plate Q Trim installed in full ball valves.

Eccentric Rotary Plug Valve With Q Trim

FLOW DIVISION
and
PRESSURE DROP
STAGING

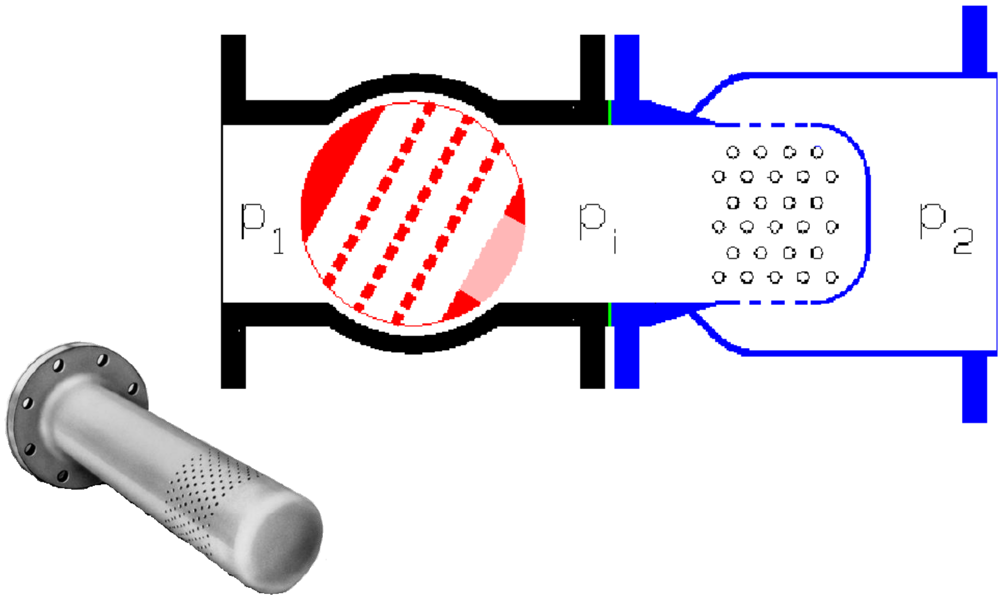




10-15 dBA attenuation

This is an eccentric rotary plug valve with rotary attenuator (“Q Trim”) plates similar to those in the segment ball valve on the previous page.

Low Noise Valve with Diffuser



10 to 15 dBA additional attenuation

A static resistance, such as a diffuser downstream of a valve reduces noise by absorbing part of the overall pressure drop, meaning that the control valve does not have as large a pressure drop across it. The diffuser does the same thing as a single hole orifice plate would do, so far as absorbing part of the pressure drop, but it does it using the flow division principle which is a low noise way of creating a pressure drop. A single hole diffuser would generate as much noise as it removes from the control valve.

Diffusers can reduce noise around 10 to 15 dBA

Vent Silencer or Diffuser

Up to 50 dBA attenuation

15-20 dBA attenuation

A vent diffuser is simply a diffuser that vents to atmosphere. It gives 15 to 20 dBA noise reduction

A vent silencer is a combination of a diffuser, and a muffler venting to atmosphere.

Path Control

- Distance
- Heavy Wall Pipe
- Thermal or Acoustic Insulation
- Buried Valve
- Sound Enclosures

Path control of noise is simply doing something to the space between the noisy valve and the people who would be bothered by it.

We will discuss distance, heavy wall pipe and insulation.

Noisy valves can also be buried on placed in sound enclosures, however we will not discuss these solutions here.

Distance Effect - Types of Noise Radiators

- Point source
  - 6 db reduction per doubling of distance from source
  - Examples: voice, atmospheric vent
- Line source
  - 3 db per doubling of distance
  - Examples: long pipe, busy highway
- Area source
  - 0 db per doubling of distance
  - Example: exterior wall, close to large pipe

The effect of distance in reducing sound pressure levels depends on the type of noise source is being considered.

Distance Effect – Line Source

Distance from Valve or Pipe	Attenuation
1 meter	0 dB
2 meters	3 dB
4 meters	6 dB
8 meters	9 dB

For each doubling of the distance from a line source of noise, the Sound Pressure Level decreases by 3 decibels.

Most valve noise behaves as a line source. Most of the noise is radiated from the downstream piping and the intensity of the noise remains constant for long distances downstream. The table illustrates how the noise from a line source decreases with distance.

The reason the reference point in the table is 0 dB at 1 meter is because the noise calculations given by the various control valve sizing programs is the noise that will be measured 1 meter away from the pipe, therefore at a distance of 1 meter from the pipe there will be no reduction due to distance.

Pipe Wall Effect

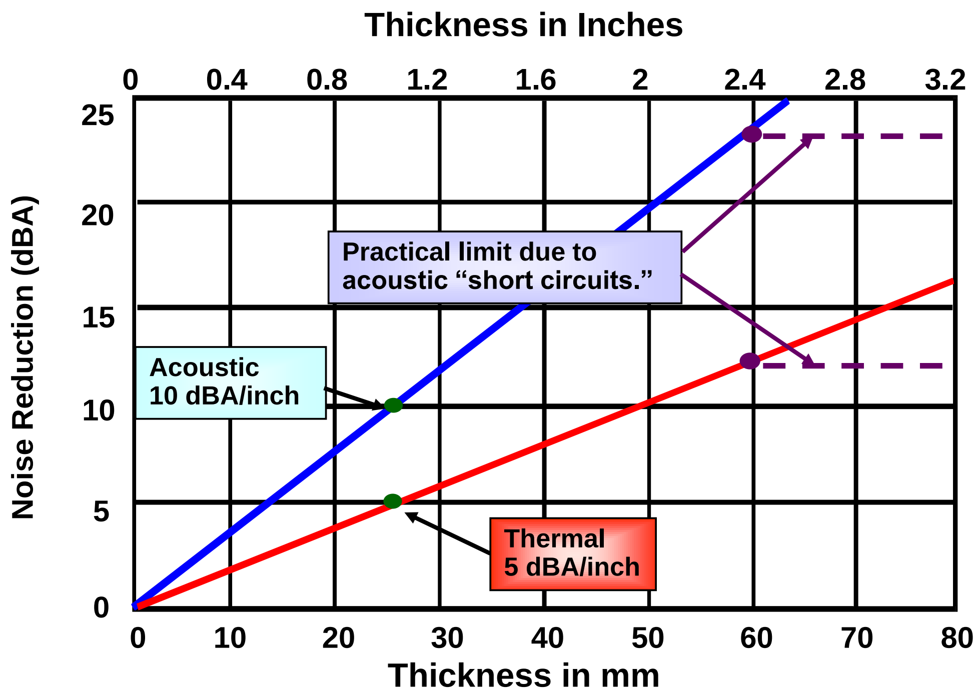
Relative pipe wall attenuation (dB)

Nominal Pipe Size	SCH 40	SCH 80	SCH 160
2	0	-4	-10
4	0	-4	-10
8	0	-4	-11
12	0	-5	-12
16	0	-6	-13
Approx cost/ft for 6" pipe	**$16**	**$26**	**$50**

The thicker the pipe wall, the more the noise is attenuated as it passes through the pipe wall to where we hear it. The table summarizes how much more attenuation is provided by schedule 80 and 160 than by schedule 40. Because valve noise propagates for long distances in the pipe downstream of the valve, if you want to rely on heavy wall pipe for noise reduction, you must use the heavy wall pipe for as far downstream as you want to have the lower noise level. When there is a lot of pipe downstream, using heavy wall pipe can be an expensive solution.

The approximate cost is an estimate I got from a pipe distributor for ASTM A106 carbon steel pipe several years ago.

Insulation



Using thermal or acoustic insulation on the piping downstream of a valve can reduce the sound pressure level that people are exposed to. As with heavy wall pipe, the pipe must be insulated for as far downstream as you need to have the reduced noise level.

Since steam lines are usually insulated anyway to conserve energy, the 5 dBA per inch of thickness is essentially free.

Special acoustic insulation is available that can reduce noise by as much as 10 dBA per inch of insulation thickness.

Insulation over about two to two and a half inches does not provide significant additional attenuation, due to acoustic "short circuits" caused by voids in the insulation and pipe hangers.

Before claiming noise reduction credit for insulation, it is important to discuss the application with the insulation supplier.

Recommended Limits for Gas

Maximum recommended SPL:	110 dBA
Maximum recommended outlet velocity:	0.5 Mach continuous 0.7 Mach intermittent

…

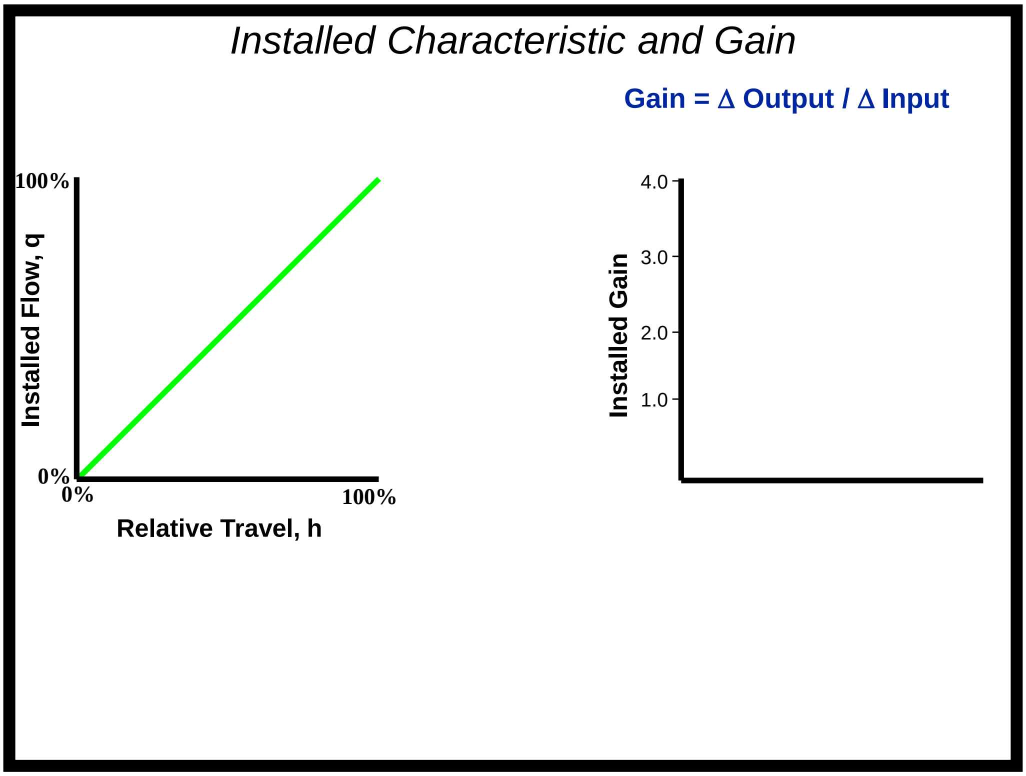
You should never apply a control valve where the calculation for schedule 40 pipe exceeds 110 dBA regardless of where the valve is located. This is because the vibration levels inside the pipe will be so high as to virtually ensure damage to valve parts.

It is also recommended that the gas velocity in the valve outlet (calculated based on the valve's outlet diameter) not exceed 0.5 Mach in continuous service. This is because high velocities in the valve outlet will generate high noise levels that are not included in the noise calculations for the valve trim.

Some manufacturers recommend even lower outlet velocities, especially with low noise valves.

Installed Gain

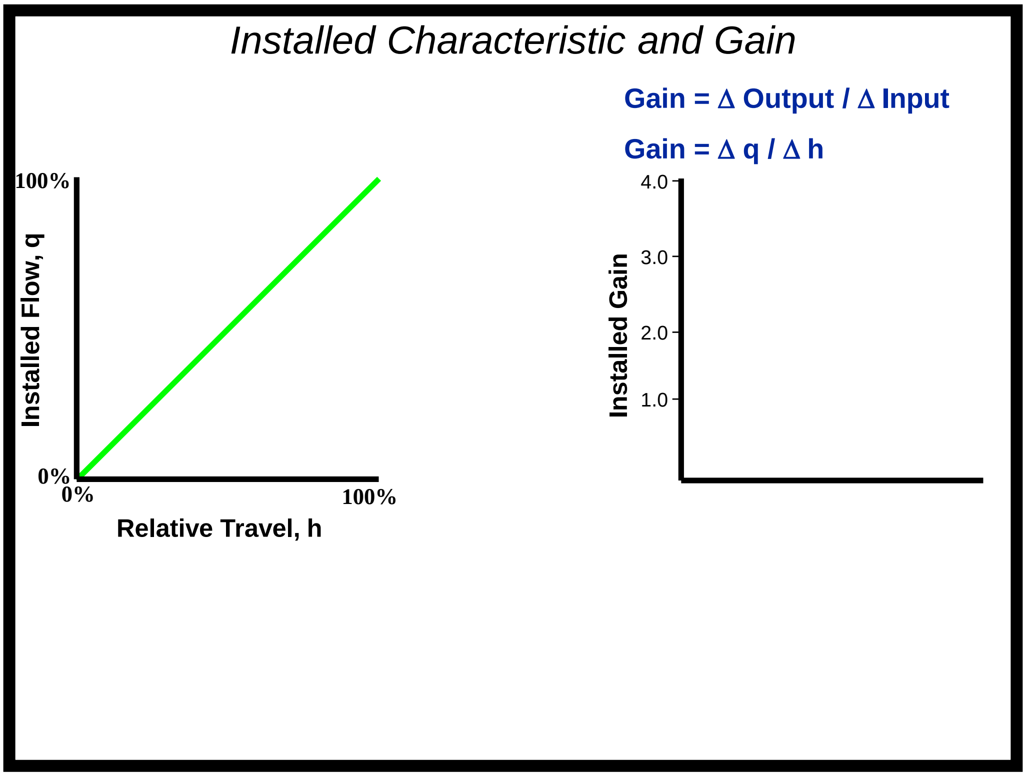
In the section on flow characteristics we saw that, for most systems, in order to get good control with stability throughout the full range of required flow rates, we need to use a control valve that has an installed flow characteristic that is linear, or at least as close to linear as possible. It is often difficult to compare the control capability of two valves with less than perfectly linear installed characteristics by simply studying their installed characteristic graphs, and you can learn more about how well they will control a particular system by examining their installed gain.



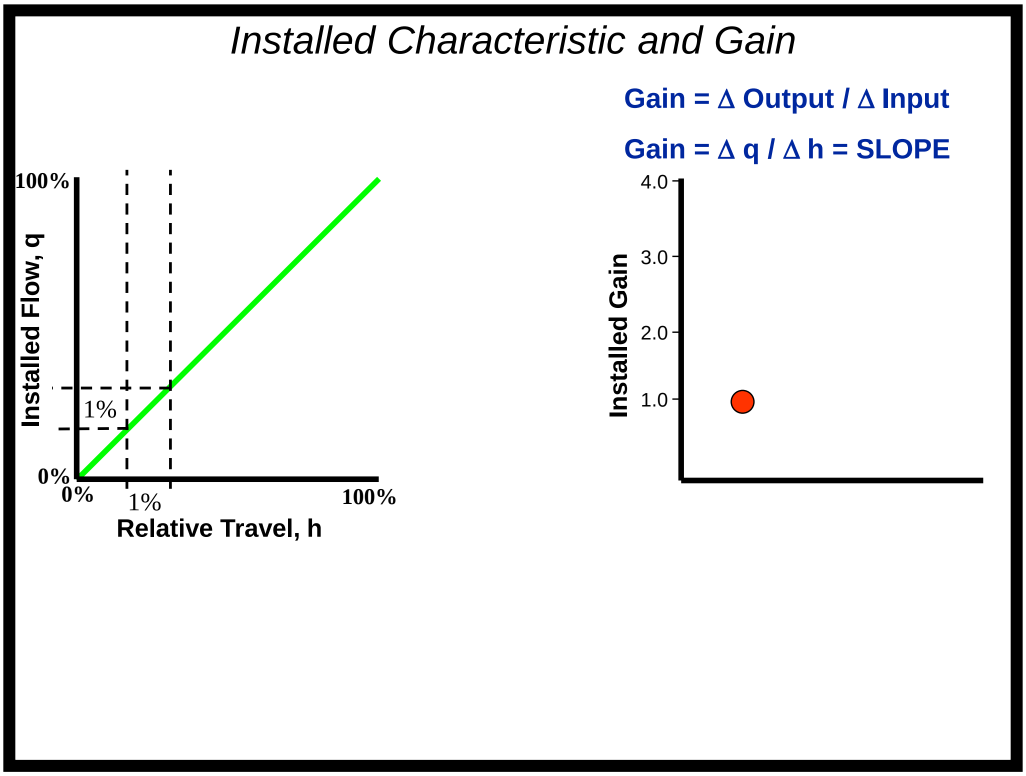
In this section we will use a lower case "q" to represent flow, which could be liquid or gas in either volumetric or mass flow units.

The left hand graph on the next several pages is an Installed Characteristic and the right hand graph is the corresponding Installed gain.

The gain of a device is defined as the ratio of changes in output to corresponding changes in input.

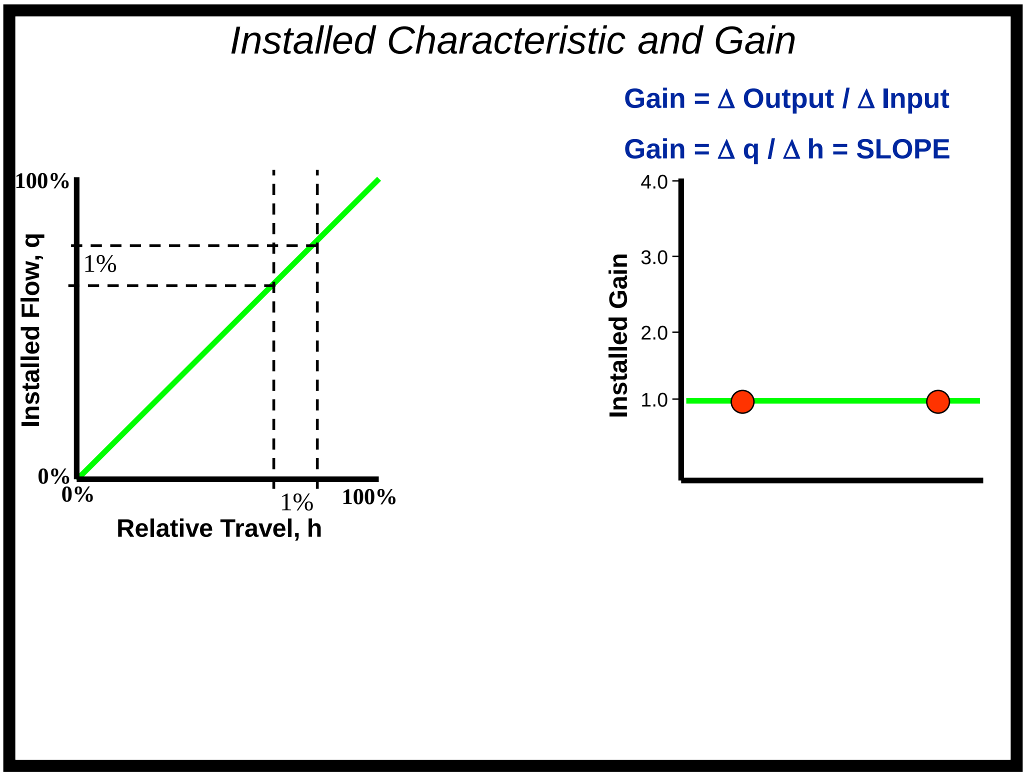


In the case of a control valve, the output is the flow in the system (q) and the input is valve position (h).



A graphical interpretation of the GAIN is the SLOPE of the installed characteristic.

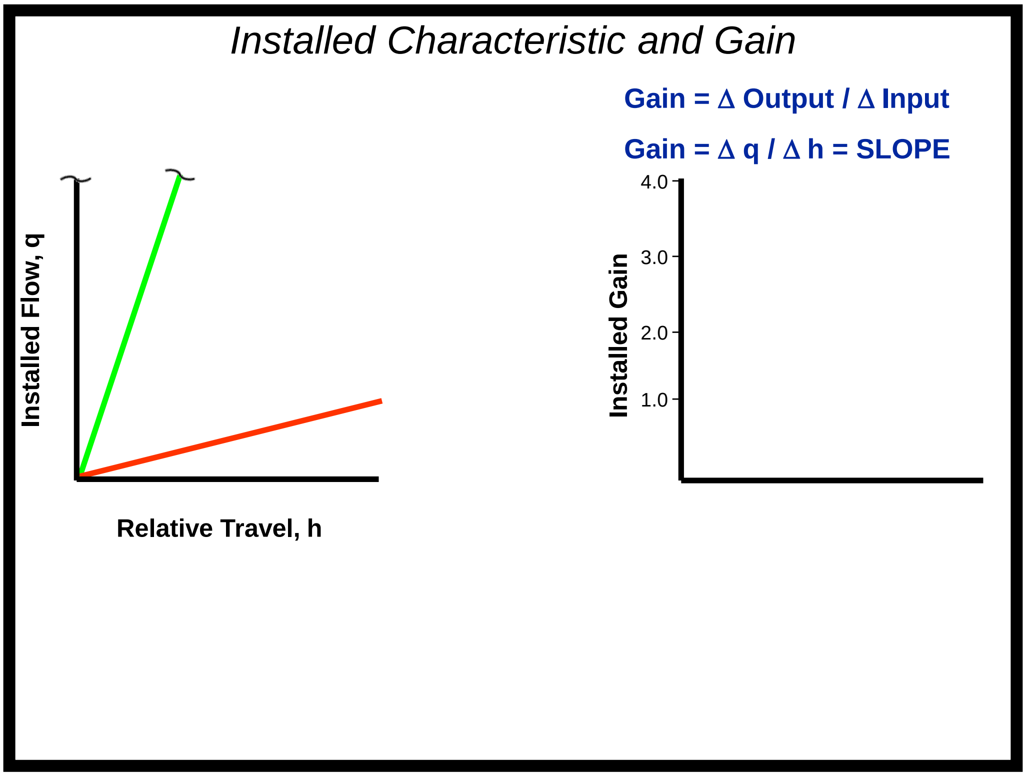
In the eighth grade we learned that the slope of a line is defined as the RISE divided by the RUN. In the case of a control valve, the RISE is the *change in flow* and the RUN is the *change in stem position.* For an ideal linear installed characteristic, as shown on the left hand graph, and any arbitrary point, a change in relative valve travel of one percent will result in a change in flow of one percent. So at this point the gain is 1.0 (1 divided by 1,So we have put a dot on the gain graph showing a gain of 1.0.



Because by definition, we have drawn an ideal linear characteristic on the left-hand graph, at any other arbitrary point a change in relative valve travel of one percent will also result in a change in flow of one percent, so the gain there will also be 1.0. We have put a second dot on the gain graph and connected them with a line.

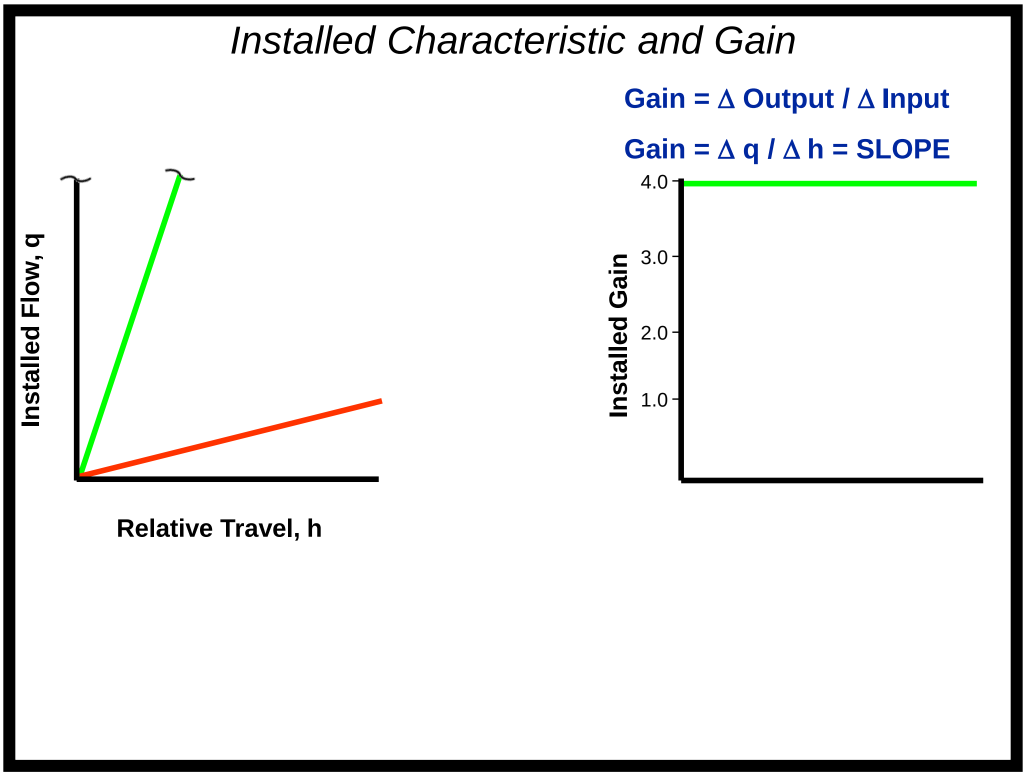
The point here is that for the ideal linear installed characteristic with a constant slope of 1.0 we get what we will call the ideal installed gain with a constant value of 1.0

We will never exactly get the ideal installed characteristic and installed gain: (1) because real valves do not have exactly linear or equal percentage inherent characteristics, and (2) because the interaction between the equal percentage inherent characteristic and the system characteristic do not exactly cancel each other, but we want to get as close as we can, so the perfectly linear installed characteristic and the constant installed gain of 1.0 are the benchmark we always aim for.

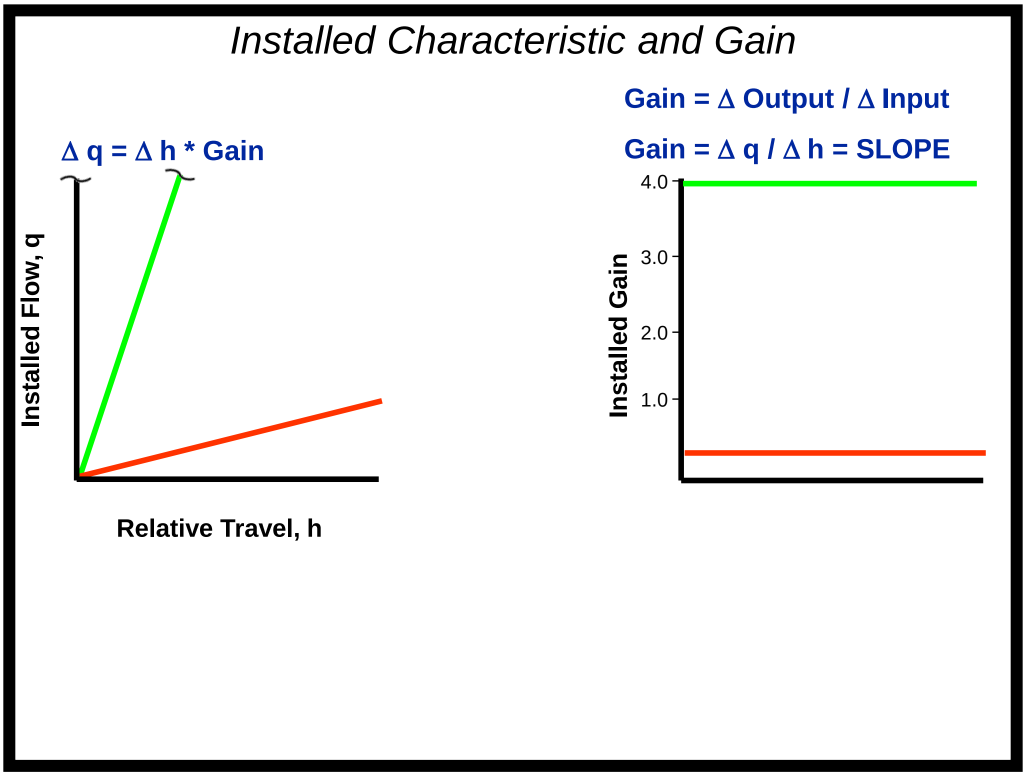


Here we have two valves with straight line <u>installed</u> characteristics, one with a very steep slope and one with a very shallow slope.

We are only showing a portion of the flow vs. opening graph of the valve with the very steep slope. The rest of it goes off the scale of the graph.



The valve whose graph is the one with the steep slope is a very sensitive valve and has a steep (but constant) slope so its gain graphs as a constant but high number .



The valve whose graph has the shallow slope is a very insensitive valve. Its gain graphs as a constant, but small number.

It turns out that neither of these valves would make a very good control valve.

The low gain valve would not make a good control valve, because when the valve stem moves, the flow hardly changes at all.

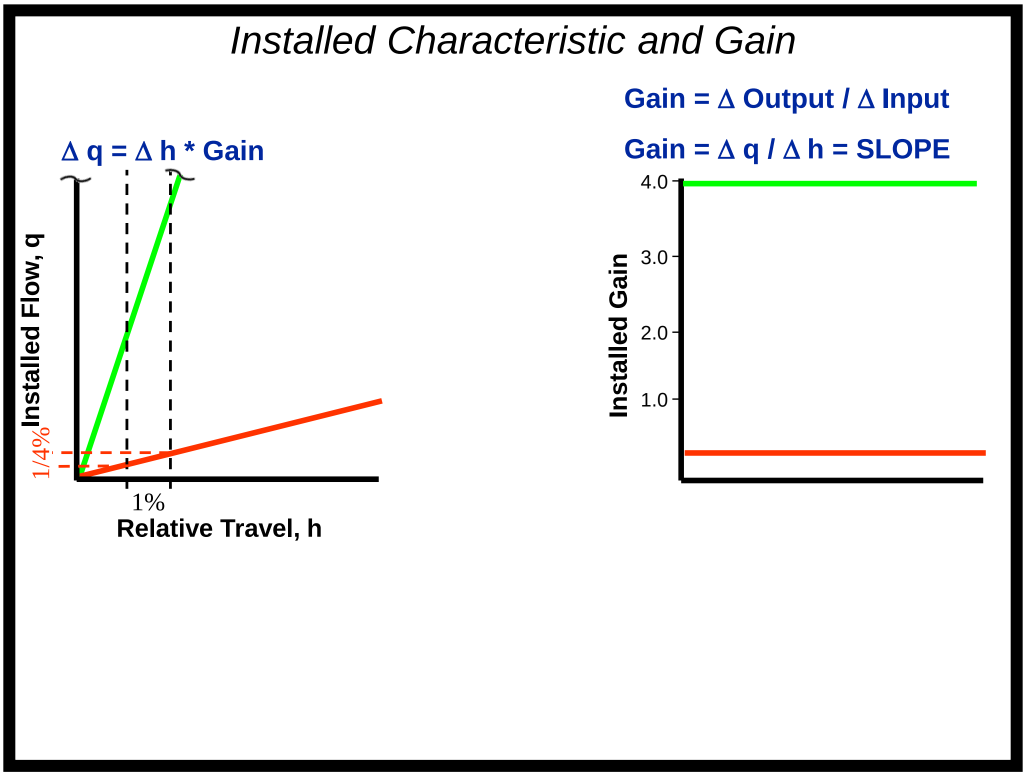
A control valve that when it moves does not change the flow is not much of a control valve!

The valve with the steep slope has a very high gain, meaning that small changes in valve position cause very large changes in flow. It is less obvious why this valve would not be a good control valve. If it was an electronic amplifier, it may be that the higher the gain the better, but a valve is a mechanical device. When two parts (such as a ball and a seat, or a valve shaft and packing) are in contact with each other they exhibit two kinds of friction. When the parts are not moving, they tend to stick together and the friction is high (static friction). When they are moving the friction becomes much lower (dynamic friction).

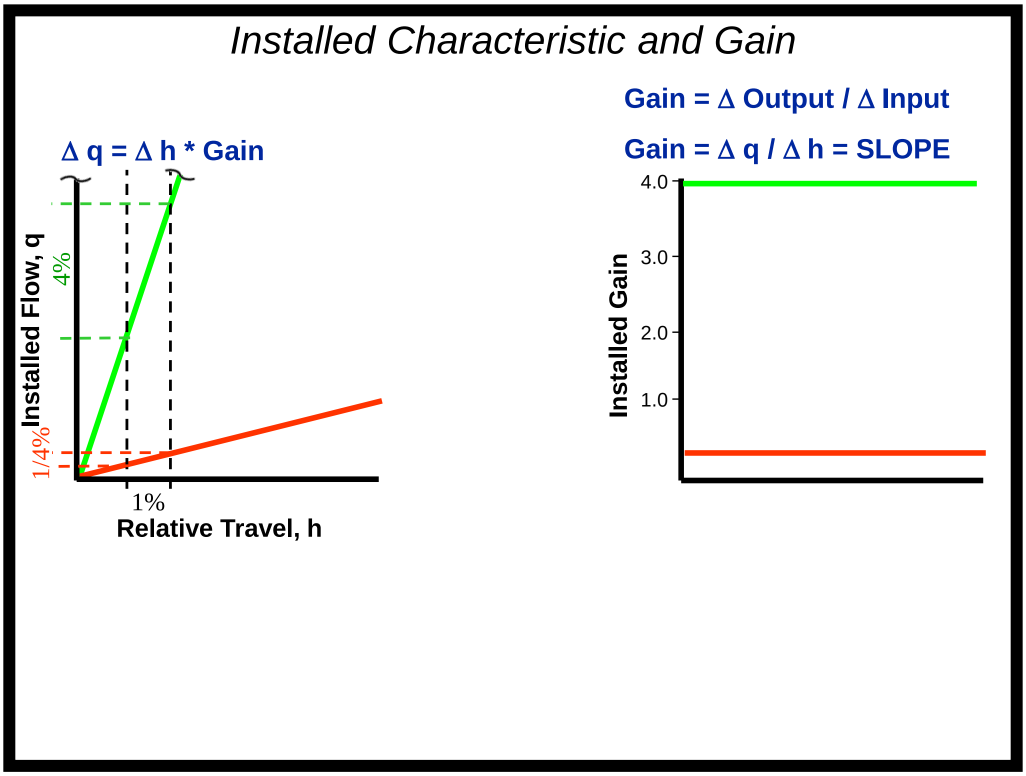
The interaction between static and dynamic friction makes it very difficult to position a valve exactly where you want it. (As an example, think of moving a refrigerator. It is easy to move the refrigerator a distance of about one foot. You push harder and harder until you generate enough force to overcome the static friction, and the refrigerator suddenly moves and continues moving fairly easily because the dynamic friction is much lower. But try to move the refrigerator precisely 1/8 th of an inch. You push and push, then suddenly it moves an inch or two. You then pull back on it, but is is stuck again, then suddenly moves back an inch or two.)

It is fairly common to find installed valves that you cannot position much more accurately than within one percent of where you want them.

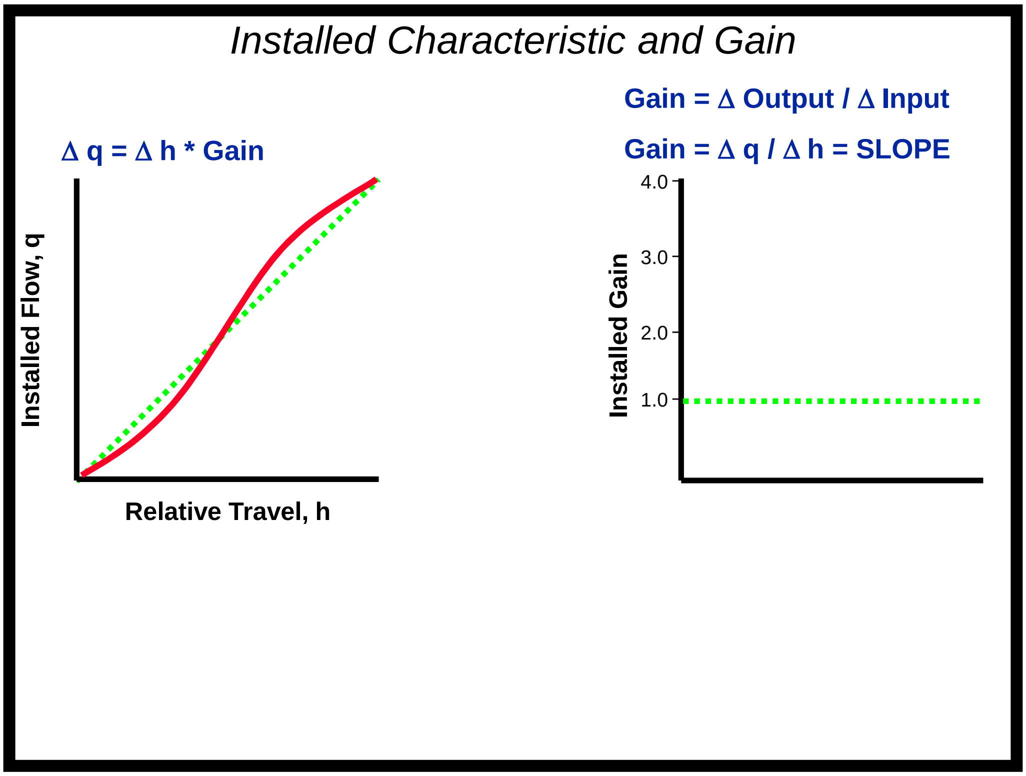
From the definition of gain, if the high gain valve with an installed gain of 4 had a position error of 1% it would result in a flow error of 4% which might not be acceptable.



As a further clarification of the meaning of gain, if the stem of the low gain valve with an installed gain of 1/4 moved by 1%, the flow would only change by 1/4%



The same 1% stem movement of the high gain valve with a gain of 4 would result in a flow change of 4%.

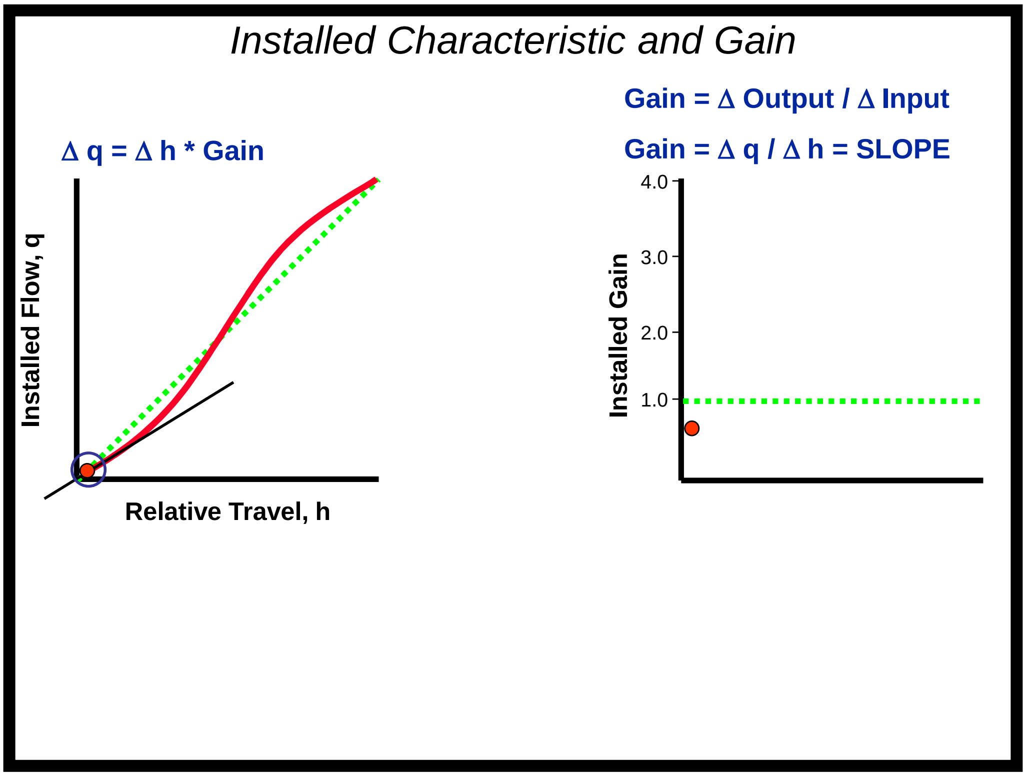


This brings us back to the real world of an equal percentage valve installed in a system with a lot of pipe, where the installed characteristic is nearly linear, but slightly “S” shaped.

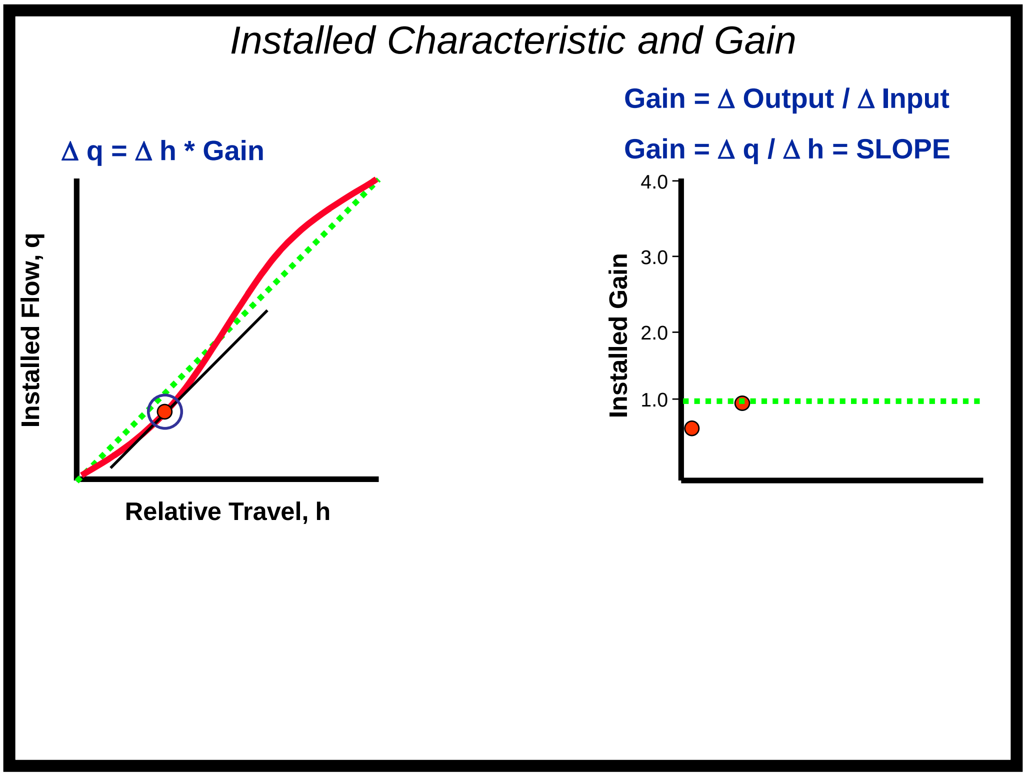
I have left dashed lines to represent the ideal linear installed characteristic and the resulting ideal installed gain with a constant value of 1.0.

Here the shape of the installed characteristic is constantly changing and its slope (and gain) is also constantly changing.

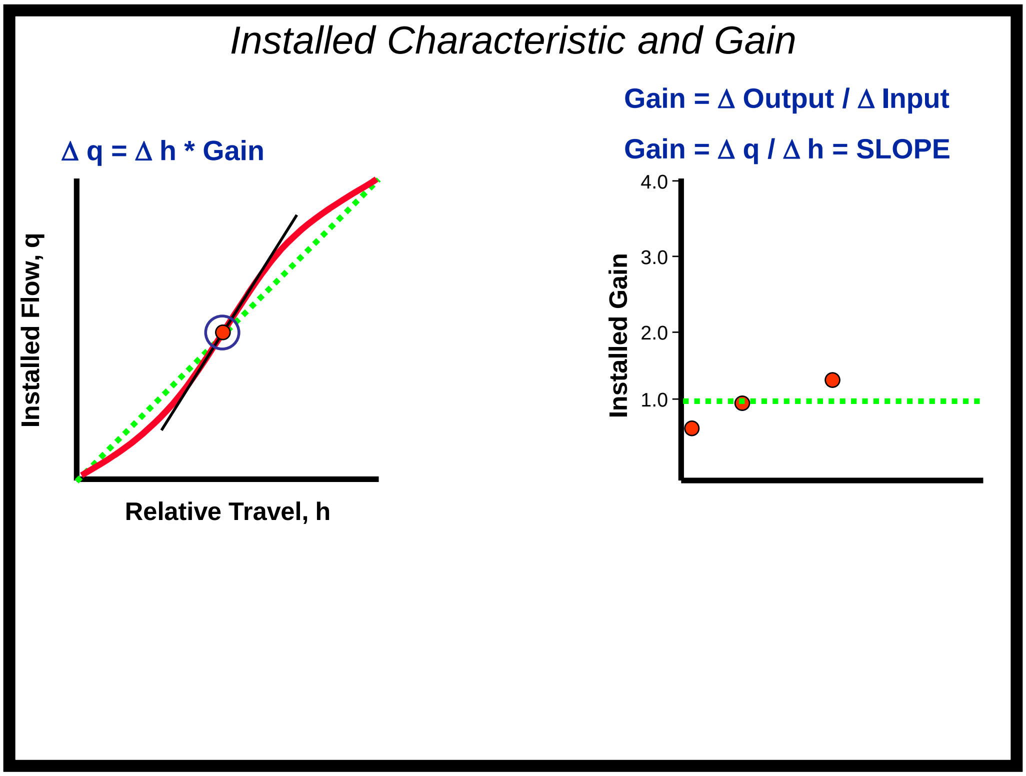
Let’s take a look at the instantaneous slope (and thus the gain) at several points.



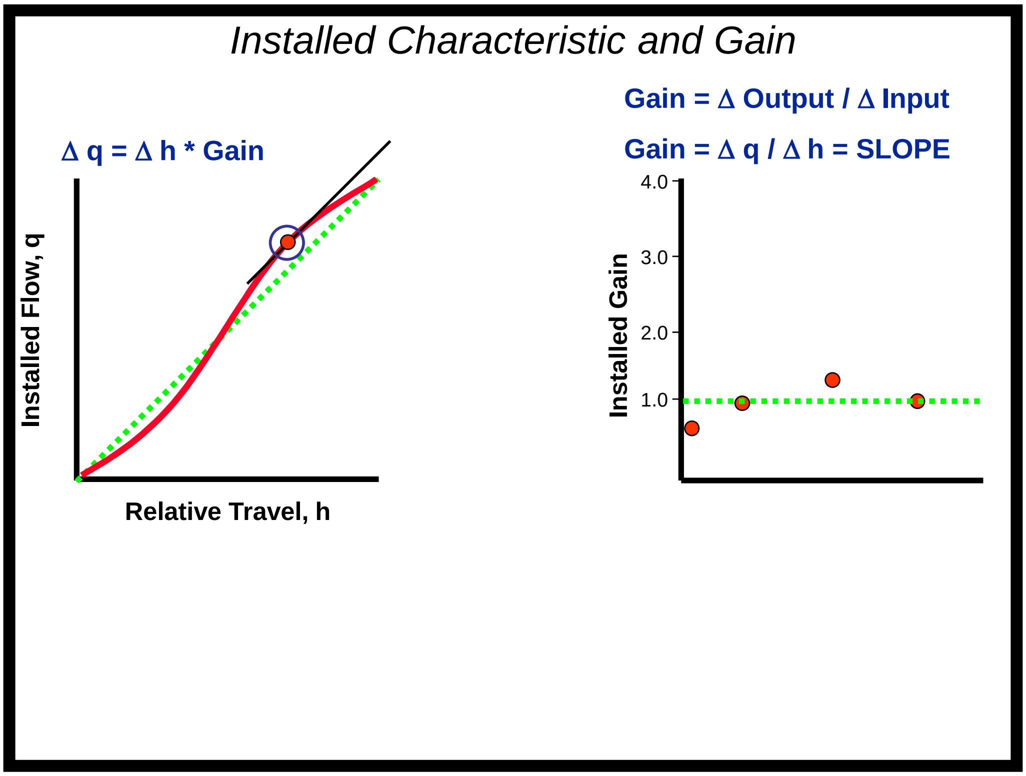
At this point, the instantaneous slope, and therefore the gain, is more shallow than the dotted line which represents a gain of 1.0, so the instantaneous gain plots as a number less than 1.0 on the GAIN graph.



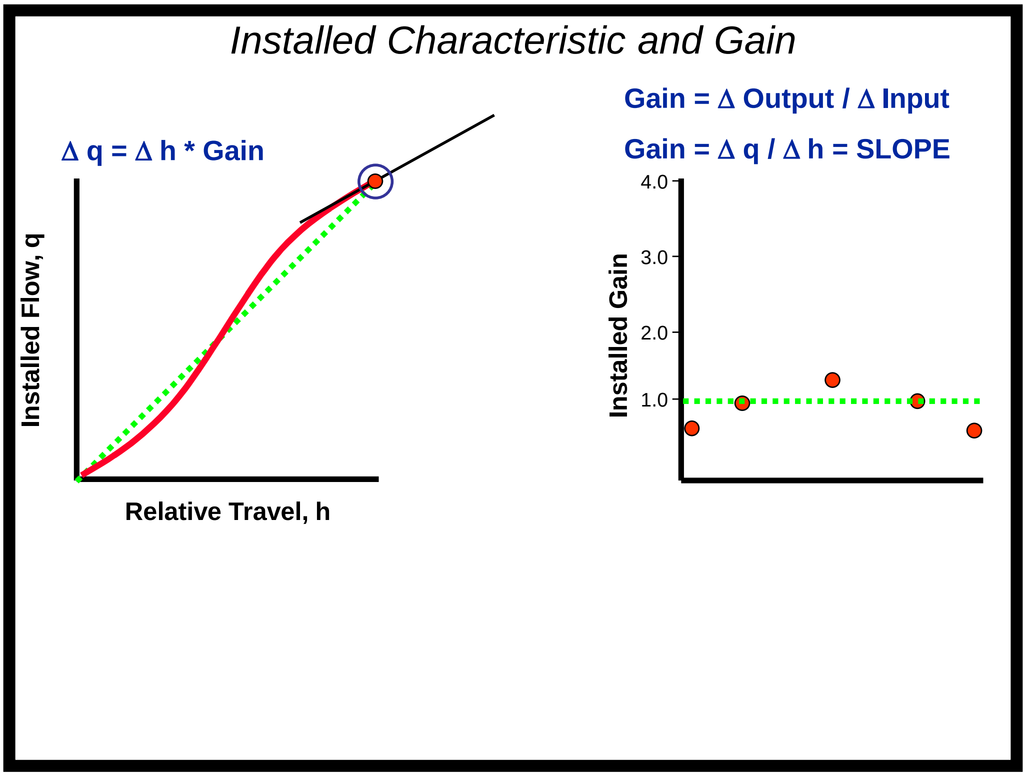
At this point, the instantaneous slope, and therefore the gain, is parallel to the dotted line which represents a gain of 1.0, so the instantaneous gain plots as a 1.0 on the GAIN graph.



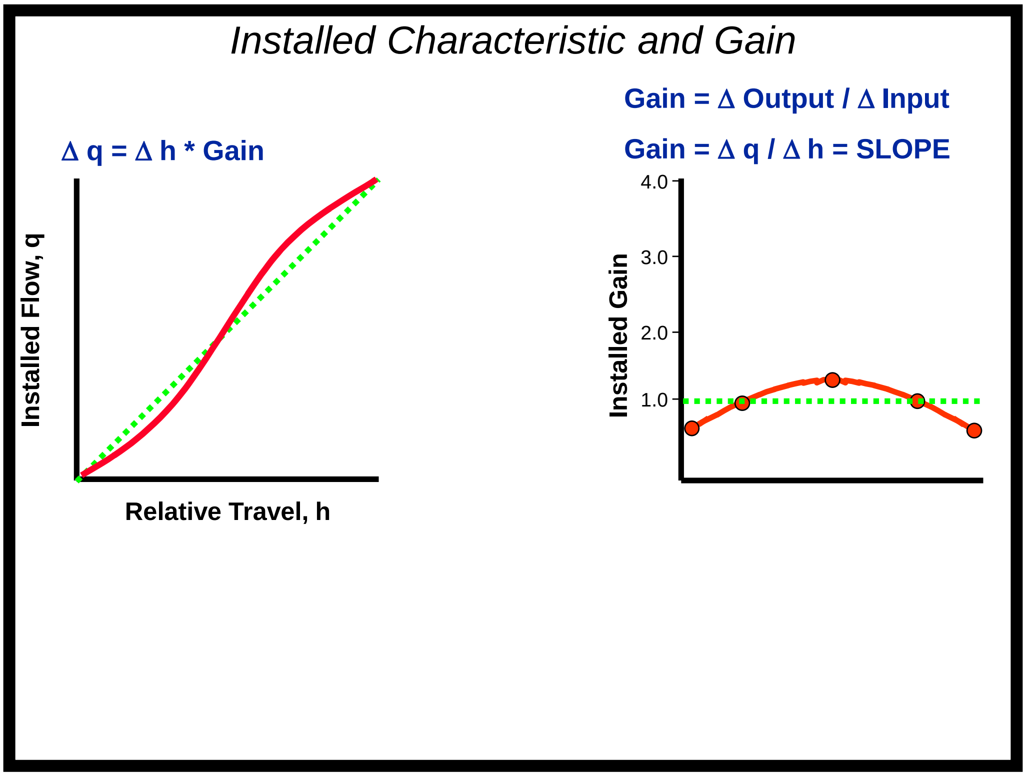
At this point, the instantaneous slope, and therefore the gain, is greater than the dotted line which represents a gain of 1.0, so the instantaneous gain plots as a number greater than 1.0 on the GAIN graph.



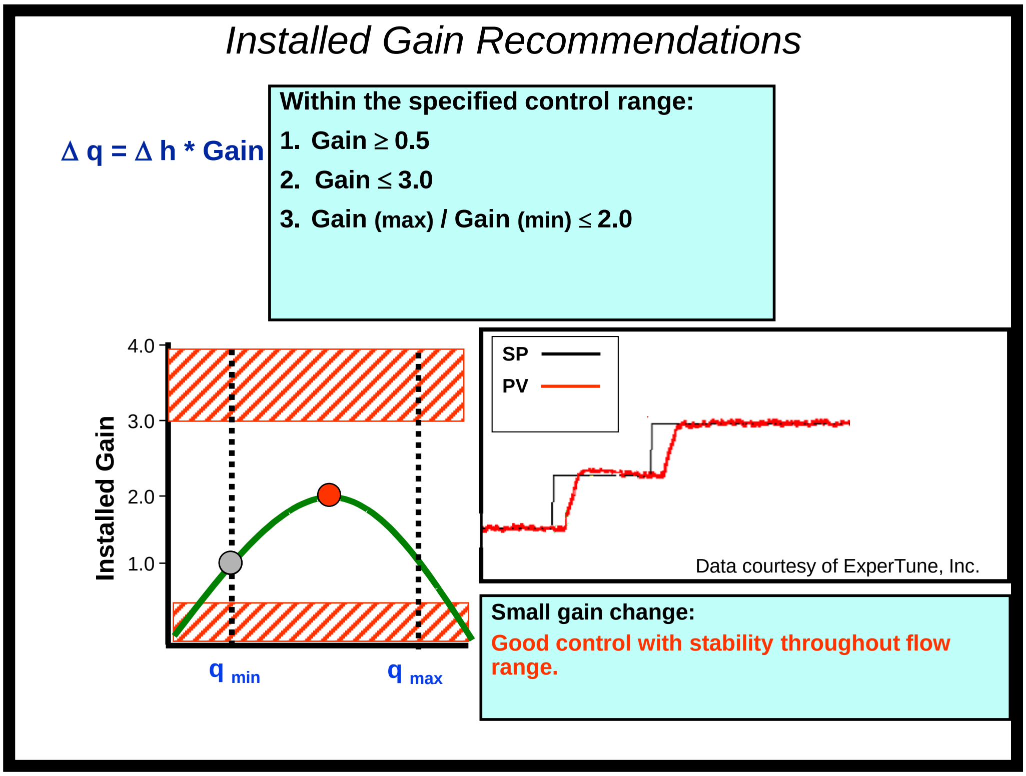
At this point, the instantaneous slope, and therefore the gain, is parallel to the dotted line which represents a gain of 1.0, so the instantaneous gain plots as a 1.0 on the GAIN graph.



At this point, the instantaneous slope, and therefore the gain, is more shallow than the dotted line which represents a gain of 1.0, so the instantaneous gain plots as a number less than 1.0 on the GAIN graph.



Connecting the points gives the typical graph of installed gain of an equal percentage valve installed in the typical system that has a significant amount of pipe.



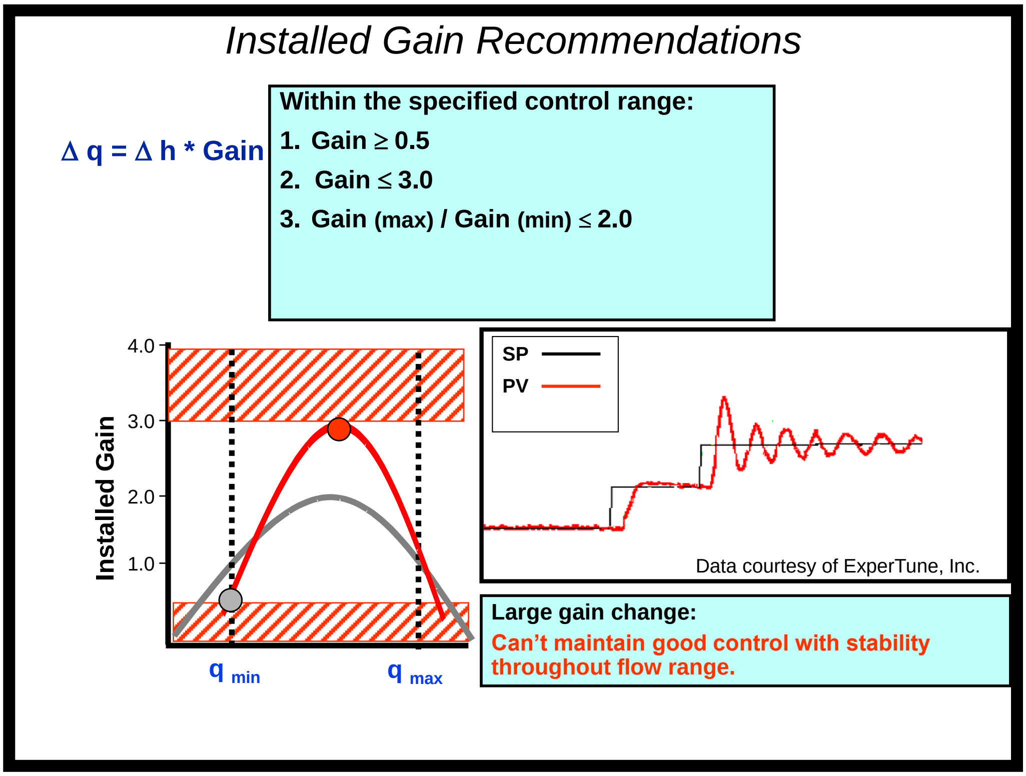
This slide shows our recommendations (and the rules that Nelprof uses when selecting the best valve size for an application) for gain magnitude and variation.

Within the specified control range (by definition we will not be controlling outside this range so we are not concerned with what happens there) that is between q min and q max, the gain should not be less than 0.5, or greater than 3.0.

Going back to the definition of gain, that is the change in flow equals the change in valve position multiplied by the gain, if the gain is too low, when the valve moves the flow hardly changes, which means the valve will not be effective in controlling flow. If the gain is too high, small errors in valve position will result in large errors in flow, making it impossible to control accurately.

Typically, if the gain changes by not much more than a 2 to 1 ratio, it will be possible to come up with one set of PID tuning parameters that will result in good control and stability throughout the required flow range.

The example shows the system response at the point of minimum gain and at the point of maximum gain in system where the gain change is 2 to 1.

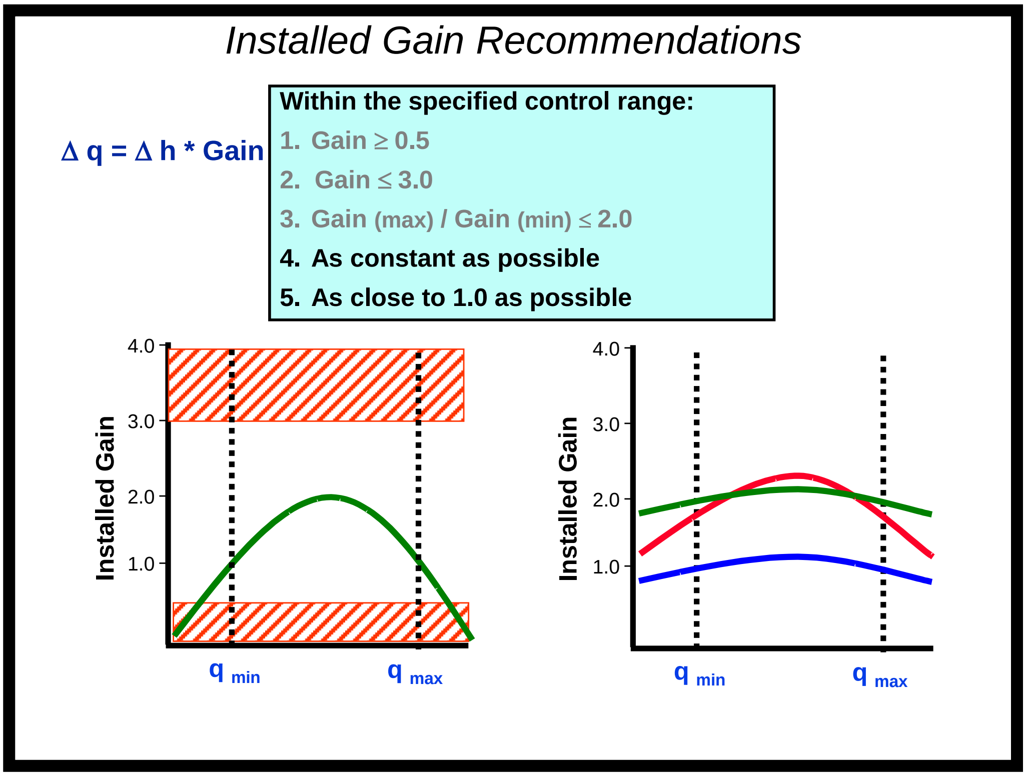


In this example, the gain is changing by almost a six to one ratio.

If the PID controller is tuned when the system is operating at the point of minimum gain, then the set point is stepped we get a quick stable response (as it should, since this is the point where the loop was tuned).

Later, when the system is operating at the point of maximum gain, if we step the set point we get an oscillatory response because the PID parameters that were suitable for a low gain system are way too aggressive for a high gain system.

With large gain variation, if the loop is tuned where the gain is low, the loop becomes unstable where the gain is higher. If it is tuned for stability at the point where the gain is highest, control will be slow and sluggish when operating in the lower gain regions.

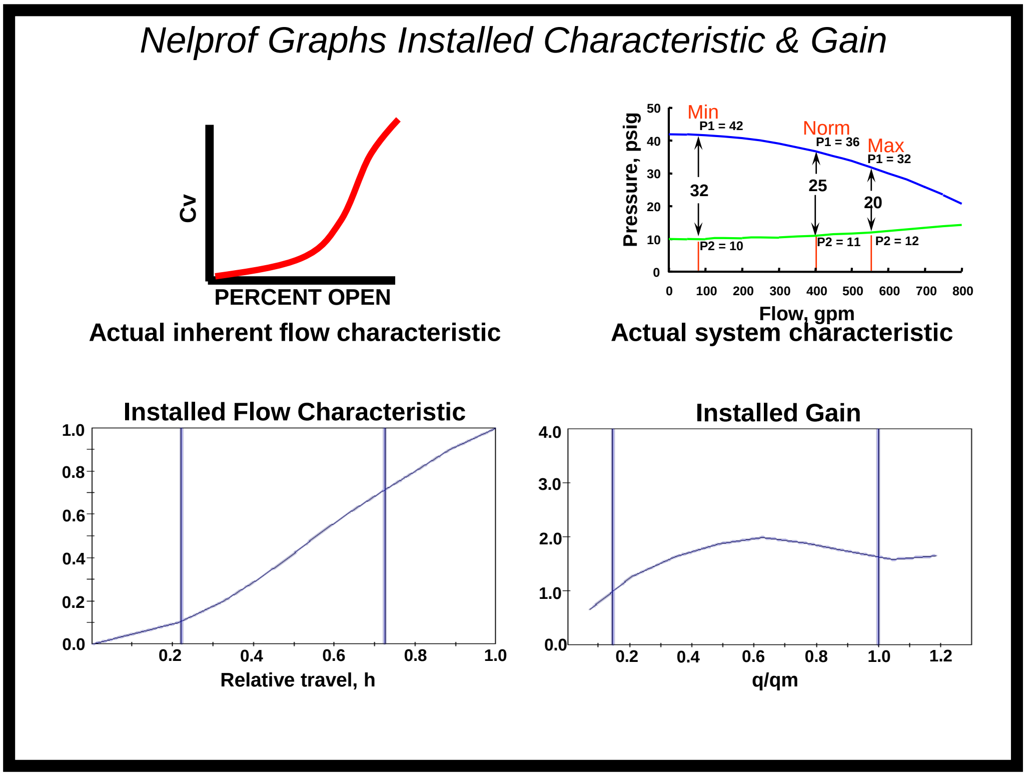


If you can't find any valve that meets the first three criteria, or if you want to select the best valve of several that all meet the first three, then use criteria 4 and 5.

The gain should be as constant as possible. The more constant the gain, the more aggressive can be the PID tuning without the danger of instability. If you had the choice between the red valve and the green valve, the green valve would be the best choice because the PID tuning could be more aggressive.

The gain should also be as close to 1 as possible. The green and blue valves both allow equally aggressive tuning, but the blue valve is a better choice.

For a valve position error of 1% the green valve would give a flow error of 2% and the blue valve would give a flow error of 1%.

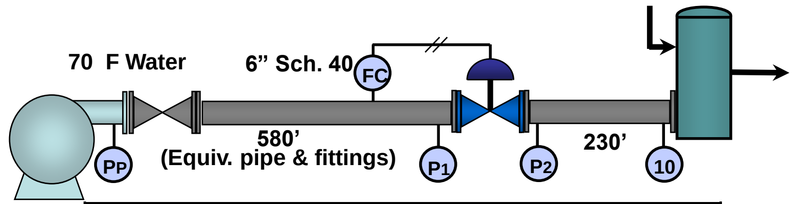


Next, we will look at a control valve sizing program that, based on a database of actual valve inherent characteristics, and with some information about how the system pressure drop changes with flow, can calculate the installed characteristic and installed gain of a particular type and size valve in the system it will be used in.

In order to define the process model, at least two flow points (maximum and minimum flow), along with the associated values of P1 and delta P are required.

It is also helpful to include the normal flow rate and its associated pressures.

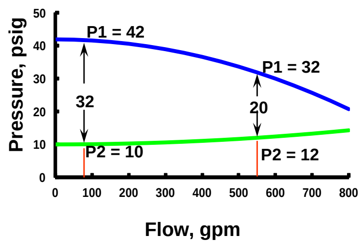
Nelprof Sizing Example

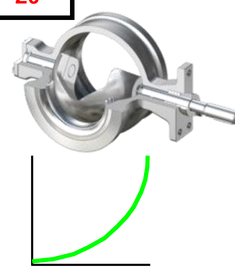


Flow Gpm (q)	P_P psig	Pipe Loss (up)	P_1 psig	Pipe Loss (down)	P_2 psig	ΔP
80	42	0.1	42	0.05	10	32
550	37	5	32	2	12	20

$$Cv = q\sqrt{\frac{G}{\Delta P}}$$

q = Flow, gpm
G = Specific Gravity
$\Delta P = P_1 - P_2$





=% Inherent characteristic

COMPUTER DEMONSTRATION using NELPROF to select the best valve by considering the installed characteristic and installed gain.

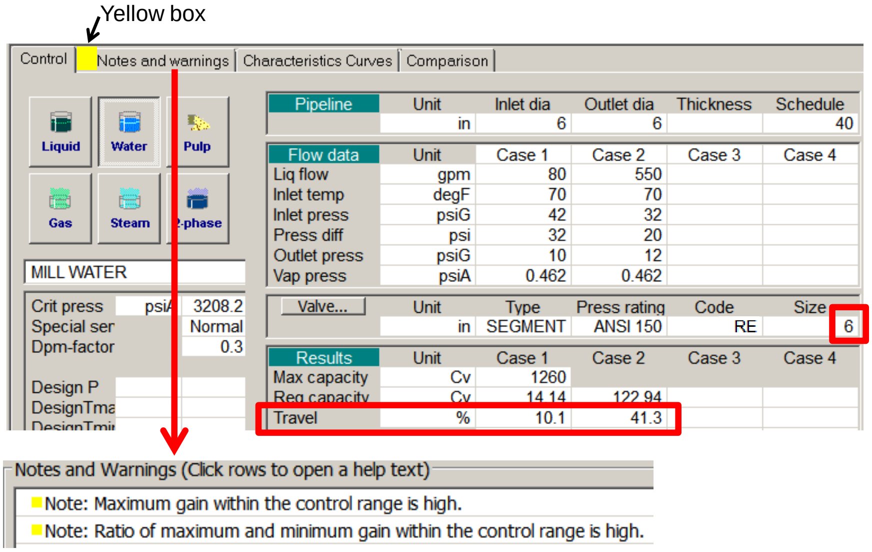
The task is to select a properly sized R SERIES SEGMENT BALL control valve. Besides flow rates, the computer program for needs to know the pressure drop across the control valve.

The data relating to control valve pressure drop was determined as follows: Start at a point upstream of the valve where the pressure is known, then at the given flow rate, subtract the system pressure losses until you reach the valve inlet, at which point you have determined P1. Then go downstream until you find another point where you know the pressure, then at the given flow rate work backward (upstream) adding (you add because you are moving upstream against the flow) the system pressure losses until you reach the valve outlet at which point you have determined P2. You can now subtract P2 from P1 to obtain ΔP.

.The graph shows how P1, P2 and ΔP vary with flow according to the rule that pressure drop in piping is approximately proportional to flow squared.

For this example things like choked flow, noise, and velocity do not affect the selection, allowing us to concentrate on installed characteristics and gain.

Nelprof Sizing Example



This is a screen shot of Nelprof, the Neles control valve sizing program where we have run a sizing calculation for a 6 inch R Series segment ball valve in the system on the previous page.

The process data for the example is entered into the upper portion of the screen.

Because we are presenting this example to demonstrate the use of the valve's installed characteristic and gain to select the properly sized valve, the example was chosen so that things like noise, flow velocity and terminal pressure drop (an indicator of choked flow) would not be important, allowing us to concentrate on valve opening and the installed characteristic and gain. For that reason, only the portion of the calculated results that are of interest for this example are shown.

This 6 inch valve would be operating between about 10% and 41% open as it controls between 80 and 550 gpm.

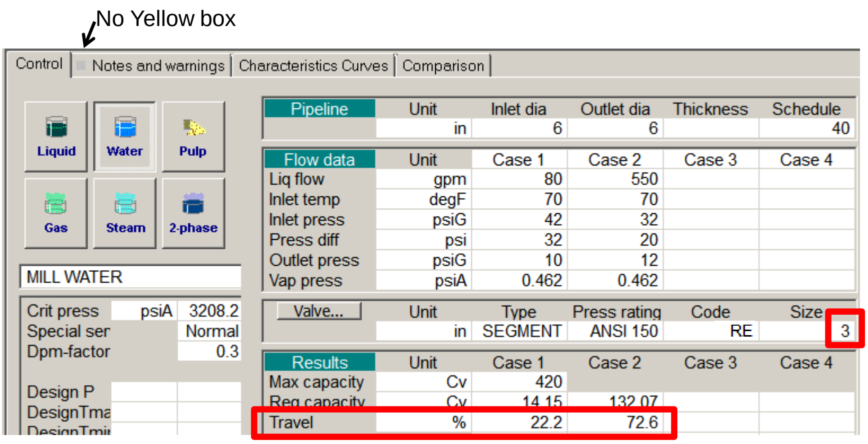
If you recall the popular rule of thumb that says a control valve should ideally control at minimum flow at not much less than 20% open and at maximum flow somewhere around 60% to 80% open you might say this valve is over sized for the application. It turns out that you would be correct, and we will see in a minute why that is by analyzing the installed gain of the valve in this particular system.

You may also recall that it is uncommon to find properly selected control valves that are the same size as the line they are installed in.

On the left side of the "Notes and warnings" tab there is a yellow box showing, which indicates there is something about this valve that the program doesn't like. Clicking the "Notes and warnings" tab takes you to the "Notes and warnings" screen. The notes from that screen are shown at the bottom of the slide.

You will see in a few slides why the program has generated these notes.

Nelprof Sizing Example



Here, the only change is that the calculation is now for a 3 inch R Series segment ball valve.

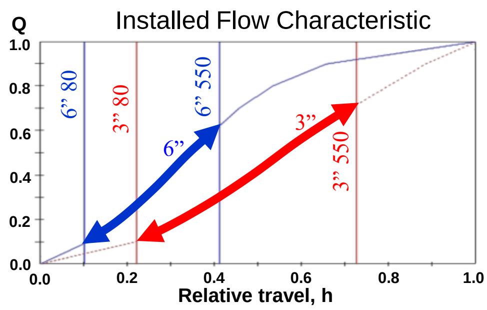
For the same process conditions, the 3 inch valve operates between about 22% and 73%, much closer to the rule of thumb.

On the next two pages we will see why a valve that meets the rule of thumb is really the best choice.

You may recall that it is very common to find properly selected ball and butterfly control valves that are two sizes smaller than the line they are in.

On this screen for the 3 inch valve, there is no yellow box on the left side of the “Notes and warnings” tab, which means that the program has not generated any notes about the three inch valve. This is because there is nothing about this valve that the program doesn’t like.

Nelprof Sizing Example



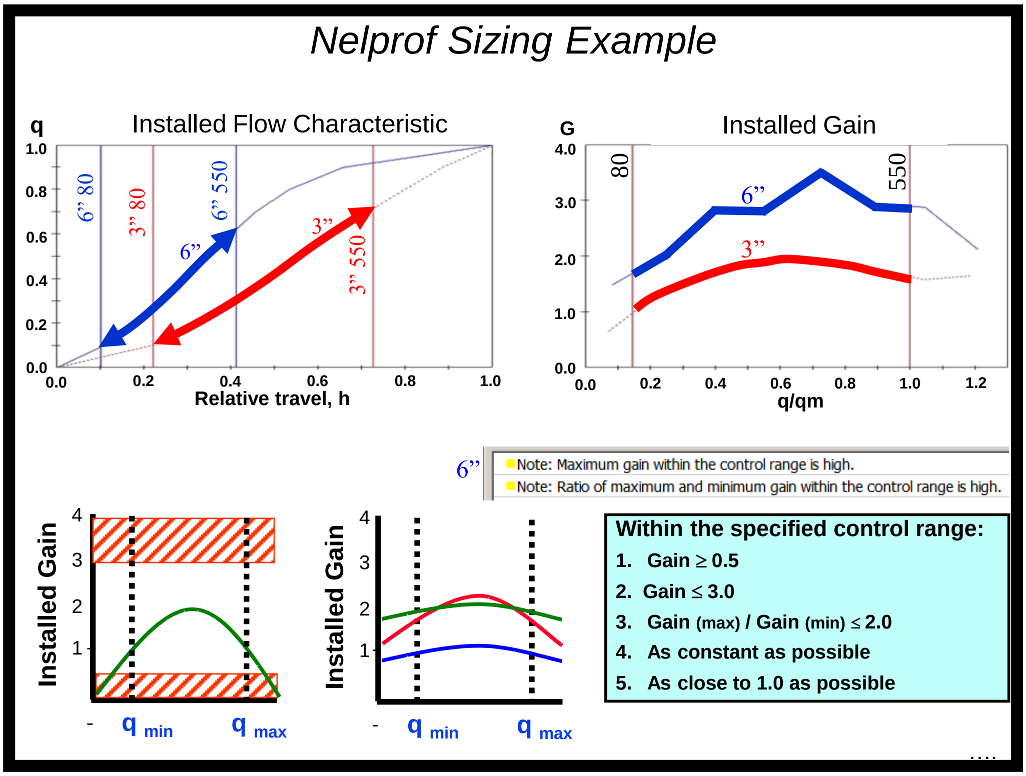
Here we see the installed characteristics of both the 6 inch valve and the 3 inch valve in this particular application. Since segment ball valves have an equal percentage inherent characteristic, and the pressure drop across the valve decreases as the valve opens to increase the flow in this application, we should not be surprised to see that the installed characteristic of both valves is quite linear within the specified control range.

The Nelprof program always draws two vertical lines on the graph to represent where the specified minimum and maximum flows (80 gpm and 550 gpm in this example) fit on the valves characteristic graph. (The first and third vertical lines go with the 6 inch valve, and the second and fourth go with the 3 inch valve.) On the computer screen the vertical lines are the same color as the valve curve they go with.

With the 6 inch valve there is a lot of wasted flow capacity (that we don't need) beyond the maximum flow of 550 gpm. There is also not very much safety factor on the low end.

The 3 inch valve is using a much greater portion of its total stroking range and the minimum and maximum specified flows are symmetrically placed on the valve's installed flow characteristic. There is about the same amount of safety factor at each end of the specified control range.

NOTE: The "x" axis for the installed characteristic is valve travel, so for smaller valves the travel will be greater and each size valve will have the vertical lines at different points. The "Y" axis scale (Q) is fractions of the flow through the valve when it is fully open. That is, Q is the actual flow at a particular opening divided by the fully open flow . Also note that the fully open flow, represented by 1.0 on the Q scale, will be different for each size valve. If the installed characteristics of these two valves were graphed on a gpm scale, the graph of the 6 inch valve would be much steeper than the graph of the 3 inch valve, and would end at a much higher point on the flow (Q) scale.



The real story of how well a valve will control the process is in the installed gain graph. The scaling of the "x" axis is in units of q/qm (q is actual flow and qm is the maximum specified flow, This means that at q/qm of 1 the flow is qm, or 550 gpm. At 0.5 on the q/qm scale the flow is 0.5 times 550 gpm, or 275 gpm.

Within the specified flow range of 80 to 550 gpm (between the two vertical lines) the gain of the 6 inch valve changes quite a bit. The more the gain changes, the harder it will be to find one good set of controller tuning parameters that will give both tight control and stable operation over the entire flow range. At about 70% of the maximum specified flow the installed gain peaks at about 3.5. From our study of installed gain you will recall that when the gain is 3.5, a 1% position error will cause a 3.5% flow error. Ideally the gain should be as close to 1.0 as possible.

The installed gain of the 3 inch valve is much more constant than that of the 6 inch valve and is closer to the ideal 1.0. This will make it much easier to tune the loop for fast, but stable control throughout the specified flow range. The peak of 2 means that a position error of 1% would give a flow error of 2% instead to the 3.5% error we would get with the 6 inch valve.

The Notes for the 6 inch valve were generated because the maximum gain is greater than 3.0, and the gain is changing within the specified flow range by more than 2:1. There were no notes generated for the 3 inch valve because it meets all of the criteria for installed gain. The conclusion from our analysis of the installed characteristics and gain is that the 3 inch valve would do a better job of control in this application than the 6 inch valve.

If we were to analyze a 4 inch segment ball valve, we would find that it would be better than the 6 inch valve, but not as good as the 3 inch valve.

The Nelprof program does not show the installed graph for two valves at the same time. For these pages I have combined two screen shots using Adobe Photoshop. The Nelprof program does make it very easy to compare the graphs of several valves with just a click of the mouse.

Graphs identify otherwise unseen problems

Customer's question: Does this look like a good application for a 6 inch Neles segment ball valve?

CONTROL VALVE SIZING: LIQUID				
Flow	gpm	Q	200	1000
Inlet pressure	psiA	P_1	90	60
Pressure drop	psi	DELTA_P	35	5
Specific gravity		G	1	1
Vapor pressure	psiA	P_V	0.5	0.5
Critical pressure	psiA	P_C	3208	======>
Recovery factor		F_L	0.94	0.8
Inlet pipe diameter	inch	D_1	8	======>
Outlet pipe diameter	inch	D_2	8	======>
Valve size	inch	d	6	======>
Valve style SPL code		VSC	2	======>
Pipe wall correction	dB(A)	DeltaLp	-2	======>
Required flow coef. (Cv or Kv)		Cv	33.81	458.78

METAL SEATED V-PORT SEGMENT VALVE (Nelprof® code RB, RE)

SIZE		Relative opening h									
DN	INCH	10 %	20 %	30 %	40 %	50 %	60 %	70 %	80 %	90 %	100 %
	6	13.9	34.1	67.2	115	178	258	352	484	719	1260

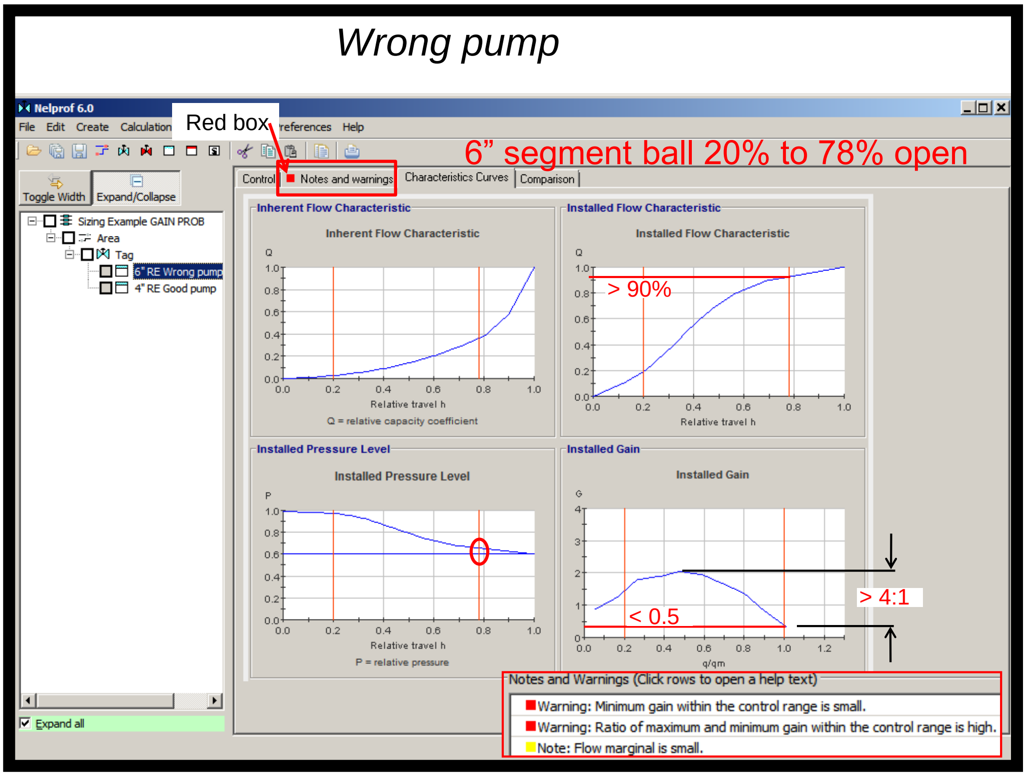
About 20% to 78% open

Here is an example of how the Neles Nelprof software identified and helped solve a potential misapplication of a control valve.

Applying the popular rule of thumb that selecting a control valve that will operate at not much less than 20% open at the minimum flow and somewhere between about 60% and 80% open at the maximum flow usually gives good results. However, just considering the valve's range of opening doesn't always tell the whole story.

A few years ago a customer sent us this print out of an Excel valve calculation spreadsheet and asked us if we agreed that a 6 inch segment ball valve was a good choice for the application. It certainly lines up with the rule of thumb.

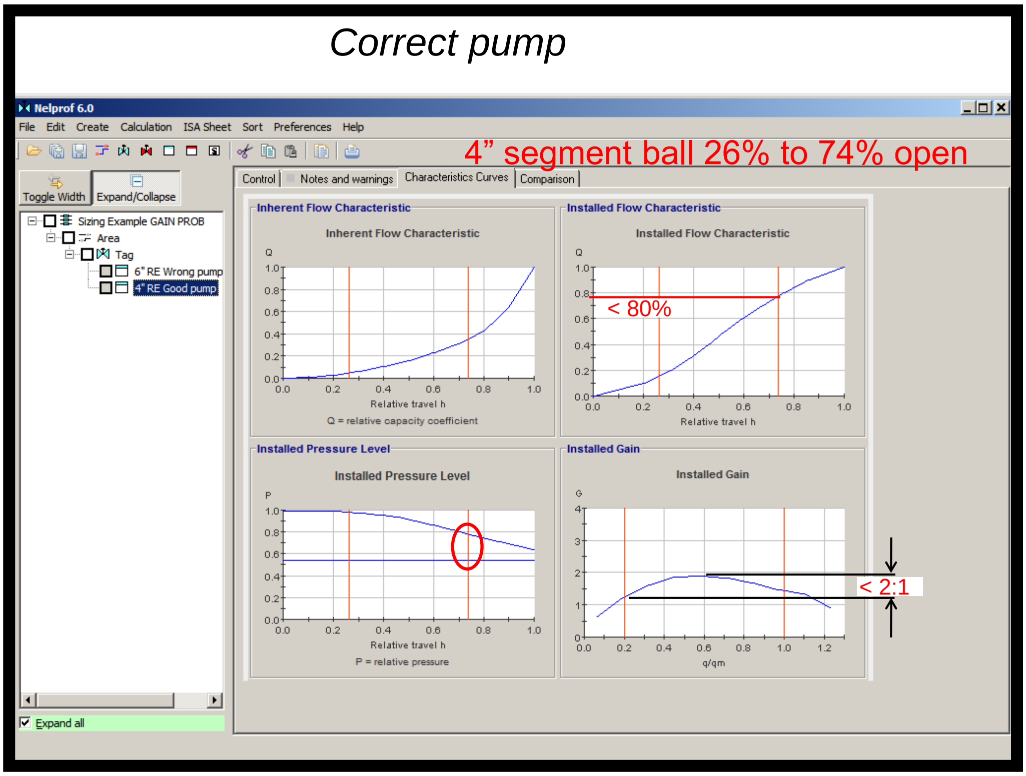
We entered the process conditions into Nelprof and then took a look at the graphs.



The installed flow characteristic graph shows that the valve will be operating between 20% and 78% open just like the customer’s sizing calculation said. What would have been difficult to know (without building the system) was that at 78% open the valve would be flowing more than 90% of its fully open capability. There is hardly any safety factor at the high end. Near the 70% open point the installed characteristic has flattened out and correspondingly the gain has taken a real nosedive, dropping to less than 0.5 at the maximum flow. Throughout the flow range, the gain is changing by more than a 4 to 1 ratio, which would make it difficult to tune the PID controller for quick and stable control throughout the flow range.

This wasn’t really a valve problem but a system pressure characteristic problem. Looking at the Installed Pressure Level graph we see that the pressure difference between P1 and P2 is falling off very rapidly as the valve gets close to 78% open. The problem in this case was that the wrong pump had been specified.

The Notes and warnings from the Notes and warnings tab have been superimposed on the screen shot.

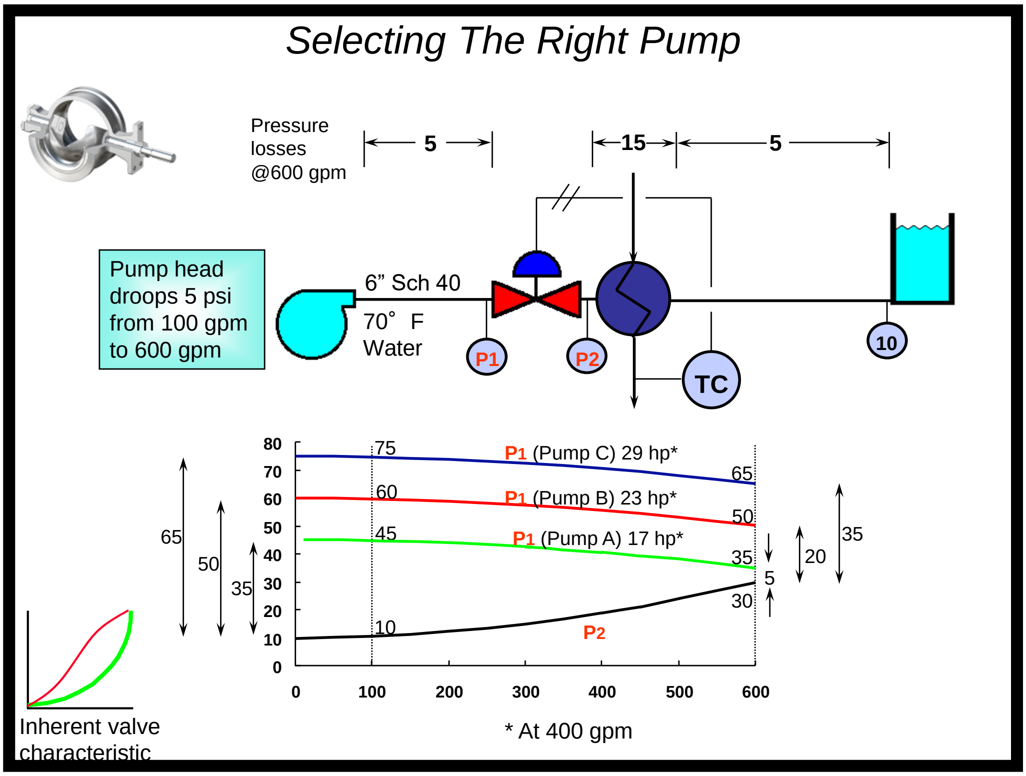


The pump had been specified, but not purchased. The customer found another pump that stared out with a higher head, and a curve that had less droop as the flow ran out to the maximum required.

After analyzing the system for the pressure drop available to the valve at maximum and minimum flow incorporating the new pump’s head curve, it turned out that a 4 inch valve instead of a 6 inch valve was required, there was much more safety factor at the high end (at the maximum specified flow, the flow is less than 80% of the fully open flow) and the installed gain was much more constant throughout the flow range.

Determining the Optimum Control Valve Pressure Drop

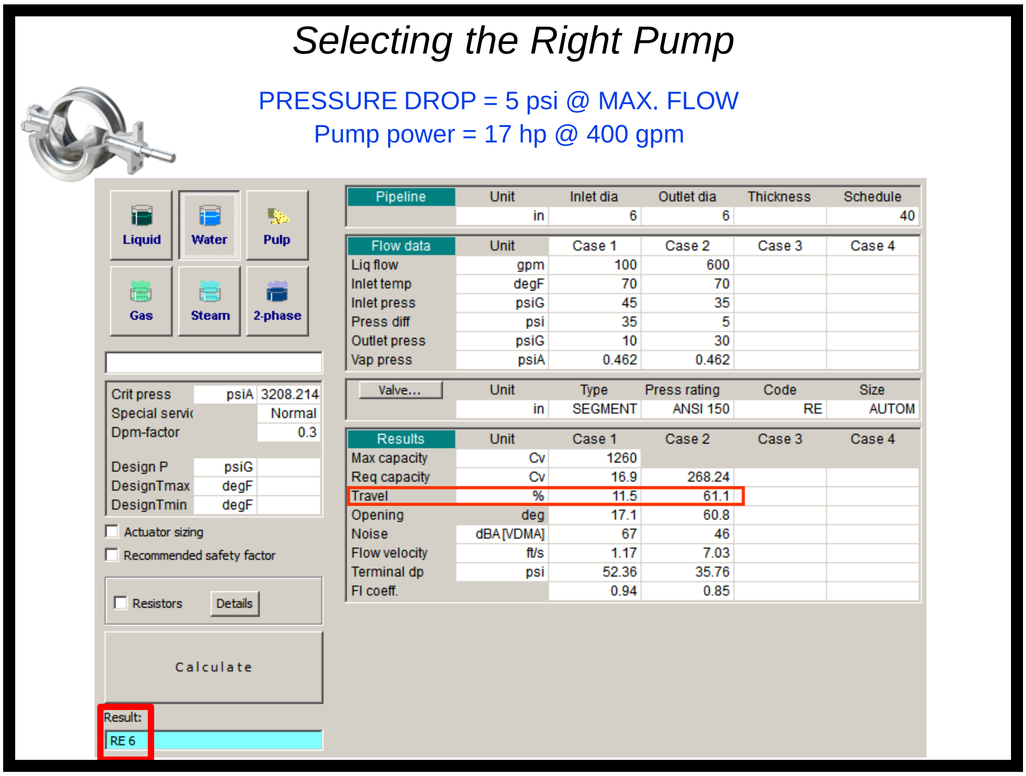
Using an installed gain calculation to select the *optimum control valve pressure drop* to design into a system to ensure adequate control while avoiding the use of excessive pumping power.



The figure shows an example system for which we are to choose the optimum control valve pressure drop by evaluating three possible scenarios. Piping and heat exchanger pressure losses are given for the maximum specified flow rate of 600 gpm. At the minimum specified flow rate of 100 gpm the losses will be approximately one thirty sixth of these values and for all practical purposes can be treated as zero for this type of analysis. The pressure at the end of the system is fixed by the constant head in the tank. The curve for P2, the pressure at the control valve outlet starts with the 10 psig static head of the tank at very low flows and increases in proportion to flow squared to 30 psig as the downstream piping and heat exchanger pressure losses increase to their 600 gpm values.

We will consider three possible pumps for the system and select the one that allows satisfactory controllability while minimizing energy consumption. Curves of P1, the pressure just upstream of the valve, are shown for each of the three pumps along with the power required by each at a normal flow rate of 400 gpm. These curves each slope downward in proportion to flow squared from the 100 gpm pump head (45, 60, and 75 psig respectively for Pumps A, B, and C) to a pressure 10 psi lower due to the combined effect of the 5 psi pressure loss in the upstream piping and the 5 psi droop in pump head from 100 gpm to 600 gpm stated in the figure. The control valve pressure drops (the difference between P1 and P2) are indicated in Figure 10 by the arrows at the left side of the figure for 100 gpm and at the right side of the figure for 600 gpm.

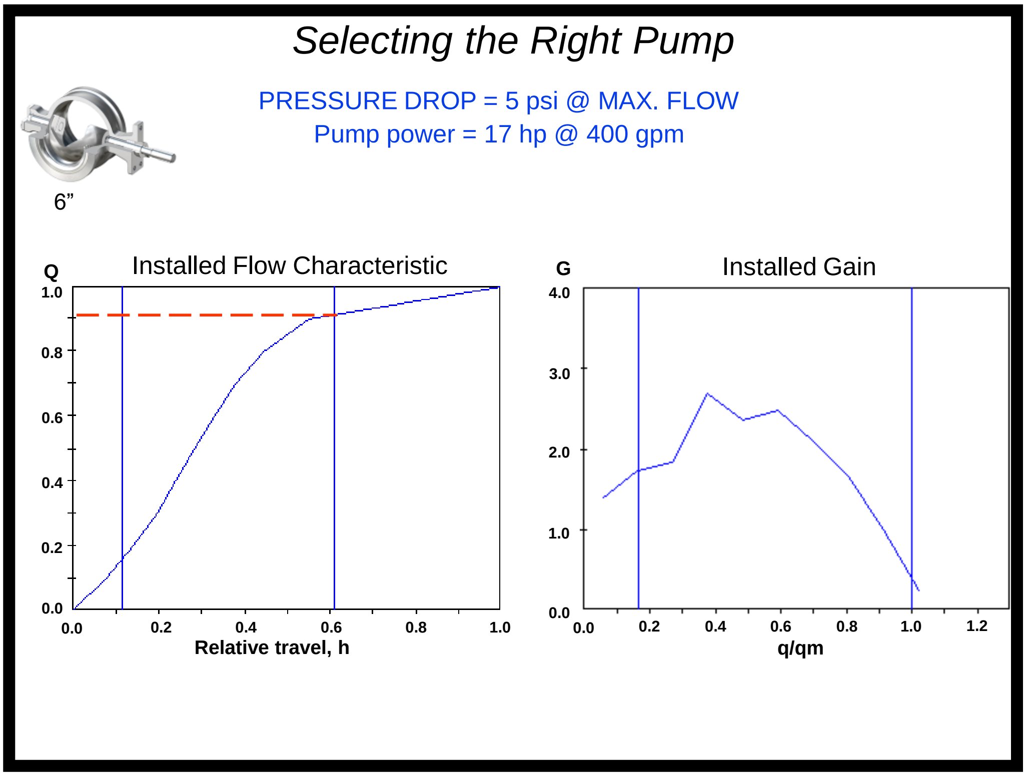
We will perform the analysis based on using a segment ball valve. Segment ball valves have an equal percentage inherent flow characteristic. Since it will be in a system with significant pipe losses, we will expect an installed characteristic that is nearly linear.



This is a screen shot of the valve sizing calculation for the case where a 17 horsepower pump gives a 5 psi pressure drop across the control valve at the maximum required flow of 600 gpm.

The program has automatically selected a 6 inch valve.

Since the valve opening ranges between 11 and 61 percent opening we might suspect that the program has selected an oversized valve, but when we see the installed characteristic we will see why the program has not selected a smaller valve.

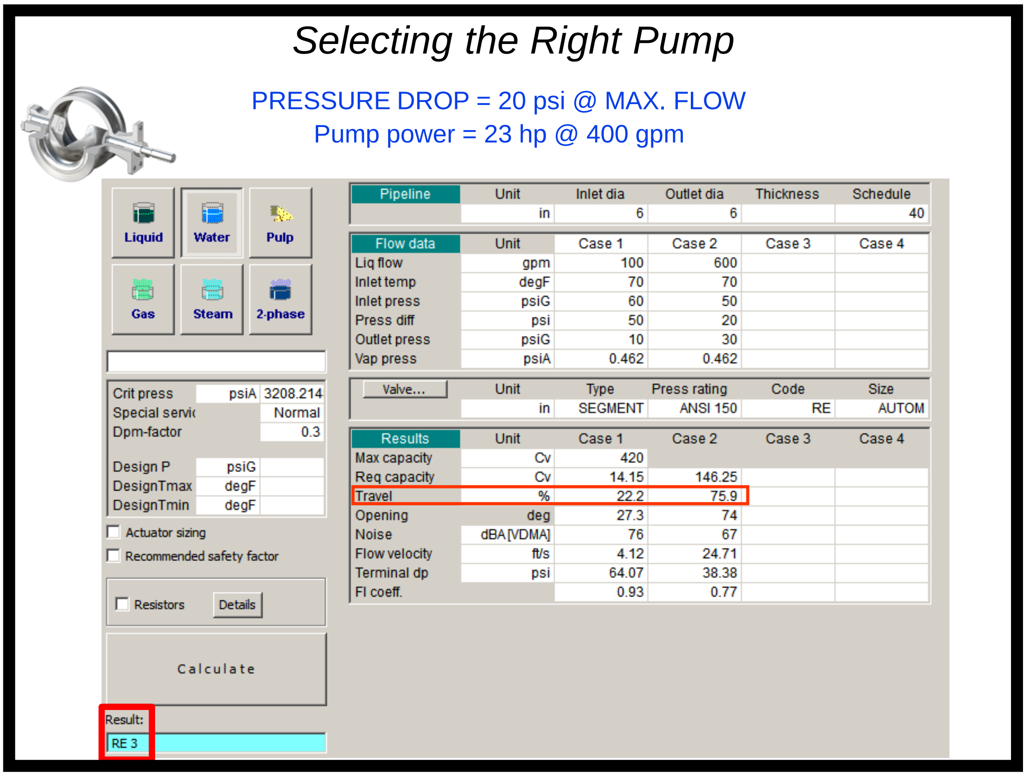


These are the installed characteristic and gain curves.

Although the valve is only about 60% open, it is already passing just over 90% of its fully open flow rate. A smaller valve would be even worse.

Looking at how the installed characteristic has flattened out near the maximum flow and how the installed gain has taken a nose dive, it appears that this valve would control poorly.

Later we will compare the installed gain with all three pumps, and we will see this one is by far the worst of them all.

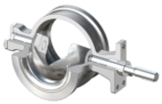


This is a screen shot of the valve sizing calculation for the case where a 23horsepower pump gives a 20 psi pressure drop across the control valve at the maximum required flow of 600 gpm.

The program has automatically selected a 3 inch valve.

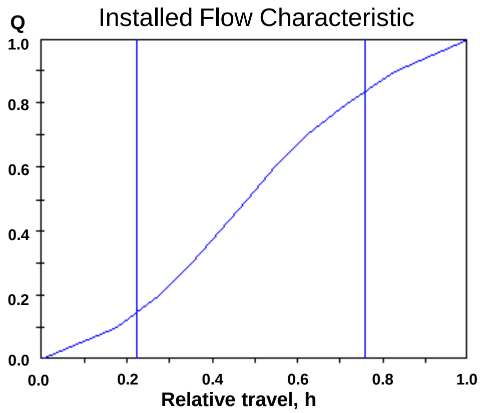
Since the valve opening ranges between about 22 and 76 percent opening we can assume that this valve might be a good choice.

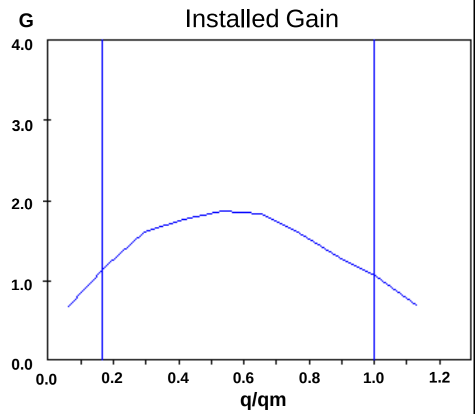
Selecting the Right Pump



3”

PRESSURE DROP = 20 psi @ MAX. FLOW
Pump power = 23 hp @ 400 gpm

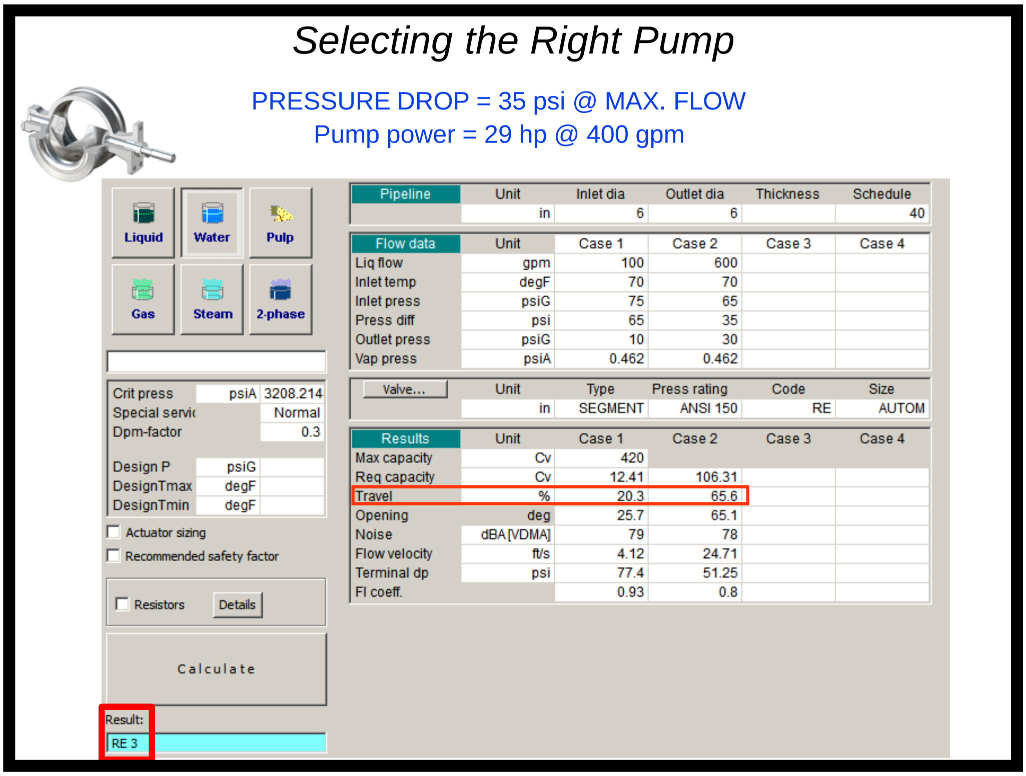




The graphs both look good.

The installed characteristic is quite linear between the specified minimum and maximum flow rates, and the installed gain meets the recommended criteria.

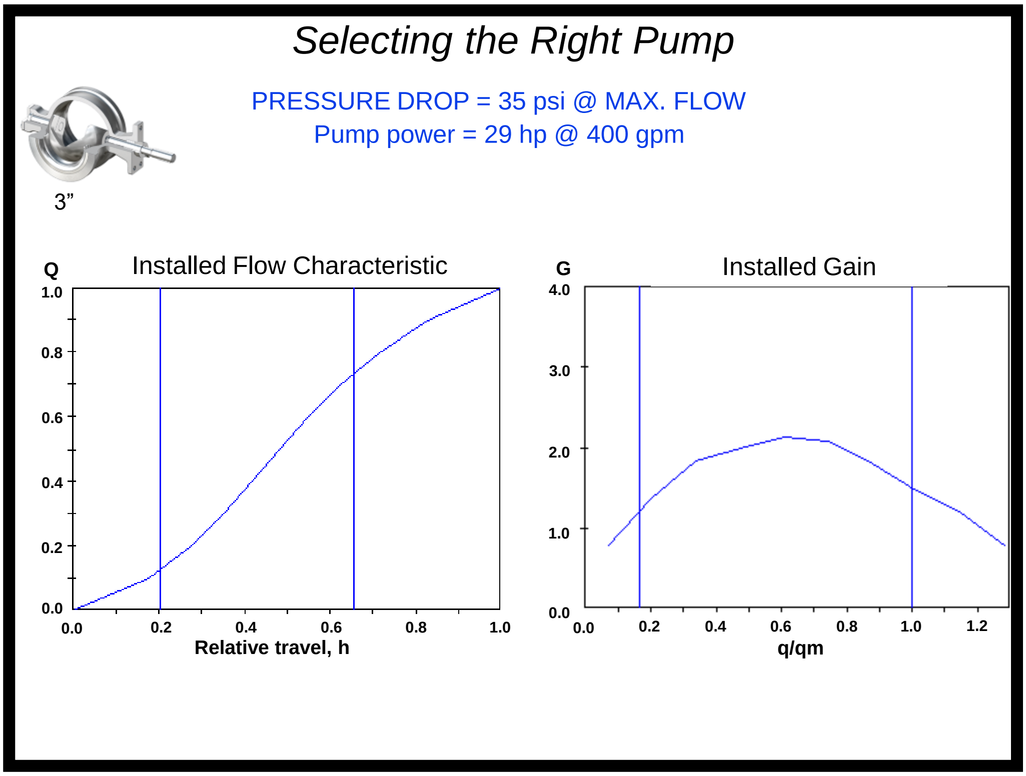
We will compare the installed gain curves to get an idea of which valve will give the best control.



This is a screen shot of the valve sizing calculation for the case where a 29 horsepower pump gives a 35 psi pressure drop across the control valve at the maximum required flow of 600 gpm.

The program has automatically selected the same 3 inch valve that it did in the previous calculation for the 23 horsepower pump.

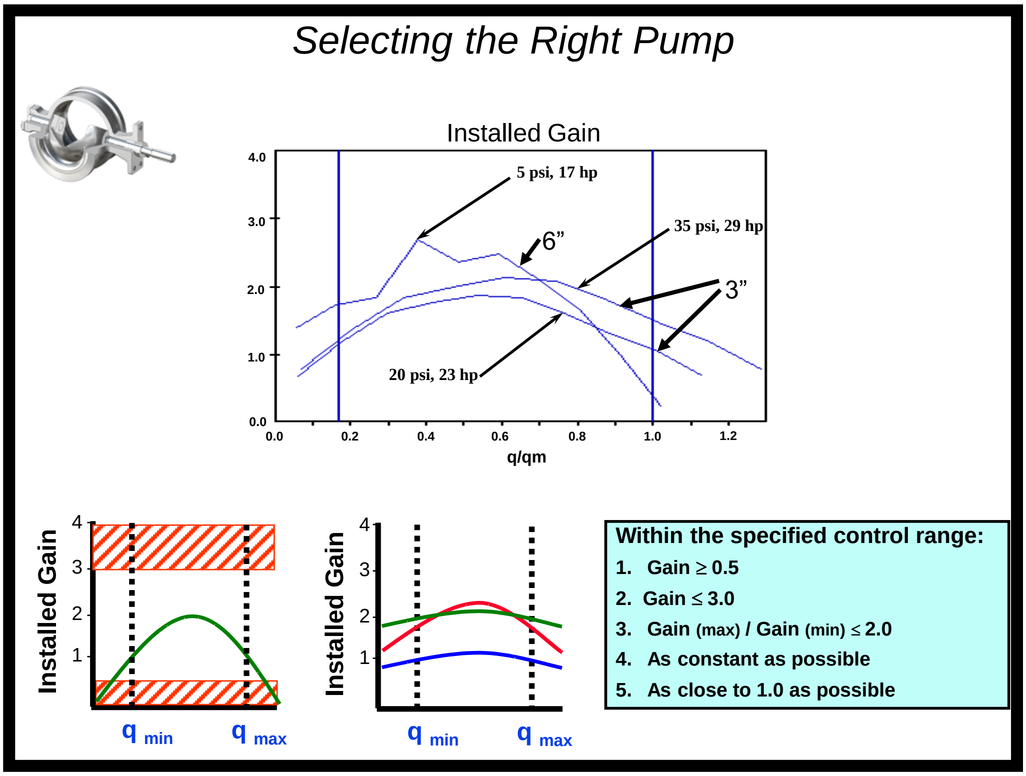
This valve operates between 20 and 66 percent open which would normally be considered quite satisfactory, though not quite as good as with the pump in the previous calculation.



The graphs both look good.

The installed characteristic is quite linear between the specified minimum and maximum flow rates, and the installed gain meets the recommended criteria.

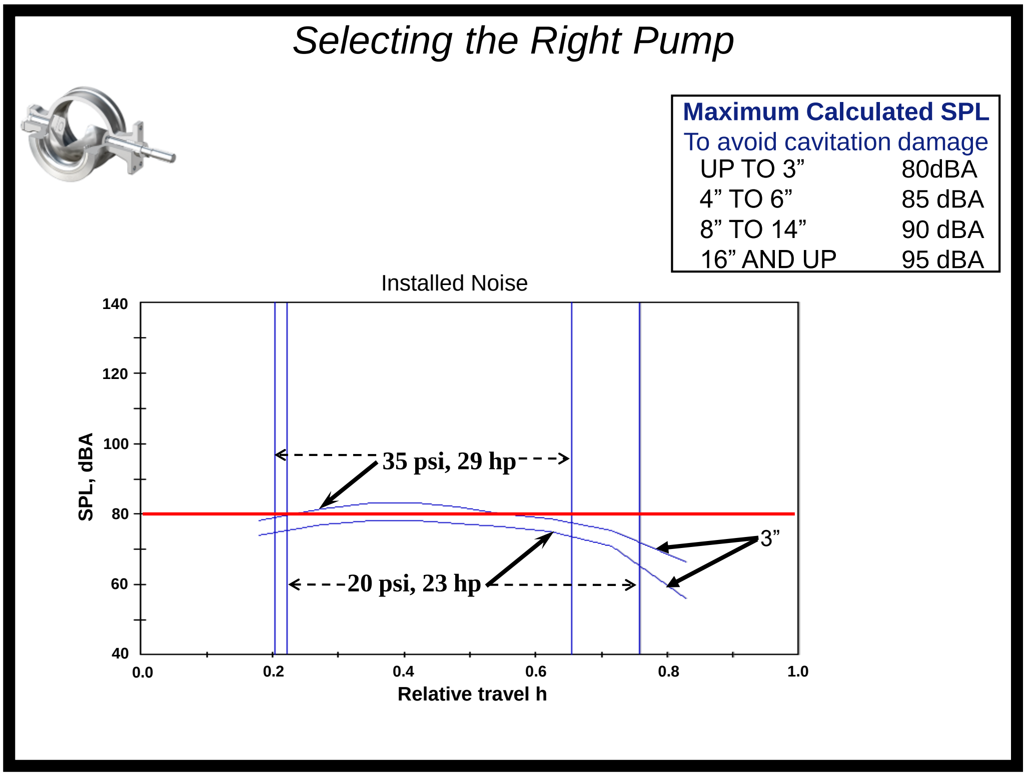
We will compare the installed gain curves to get an idea of which valve will give the best control.



Here we see the installed gain calculations for all three pumps compared.

With the 17 horsepower pump, besides requiring a more expensive valve, the gain graph looks terrible. The installed gain is the highest of the three (meaning larger flow error for the same valve position error), it drops to just under 0.5 at the maximum flow and the variation in gain over the flow range is large enough that it would be very difficult to come up with PID tuning parameters that would give both good and stable control over the entire required flow range.

The gain graphs of both the 23 and 29 horsepower pumps fall within the recommended gain criteria. But the 23 horsepower is the winner, since its gain is more constant and closer to 1.0, and it is also the most economical of the two to operate.



From our earlier lesson on liquid flow in control valves you will recall that there is a correlation between calculated hydrodynamic noise and cavitation damage.

Specifically, in a 3 inch valve, cavitation damage can be expected if the calculated noise (based on Schedule 40 pipe) exceeds 80 dBA.

With the 29 horsepower pump, the noise exceeds this value over a portion of the specified flow range.

Some of the extra 6 horsepower required by this pump is being used to generate noise and the energy to potentially cause damaging levels of cavitation.

Process Variability

What is Process Variability?

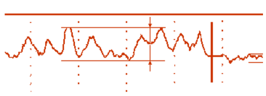
Unwanted Variation of the Controlled Variable

Flow, pressure, temperature, level or any other process variable..

Process variability is any unwanted variation in whatever it is you are controlling, flow, pressure or whatever. The reason we don’t want these controlled variables changing too much is because these changes show up as changes is the properties or quality of the end product.

Why is Process Variability Important?

- **Product exceeds specifications**
- **Poor quality product**
- **Wasted energy**
- **Lowered capacity**
  - **May have to reduce average throughput to avoid exceeding process constraints**
- **Extra stress on the valve.**
  - **Other valves, pumps, relief valves and vessels are also stressed by these unwanted variations**
- **Unscheduled down time**



VARIABILITY = $$$

If you are having a variability problem there are a couple of ways of addressing it.

You can make the product to exceed the specification, so that, for instance, the thinnest spots in a roll of paper are still thick enough. The problem is that then you are giving away free product and in today’s competitive market we usually can’t afford to do this.

If you end up making a poor quality product, you either have to sell it for less, recycle it (run a hydrocarbon back through a column or reactor or re pulp a roll of paper) which uses energy and time and is therefore expensive, or just sell it and hope no one notices. If your customers discover they are getting a low quality product, they will start buying from someone else.

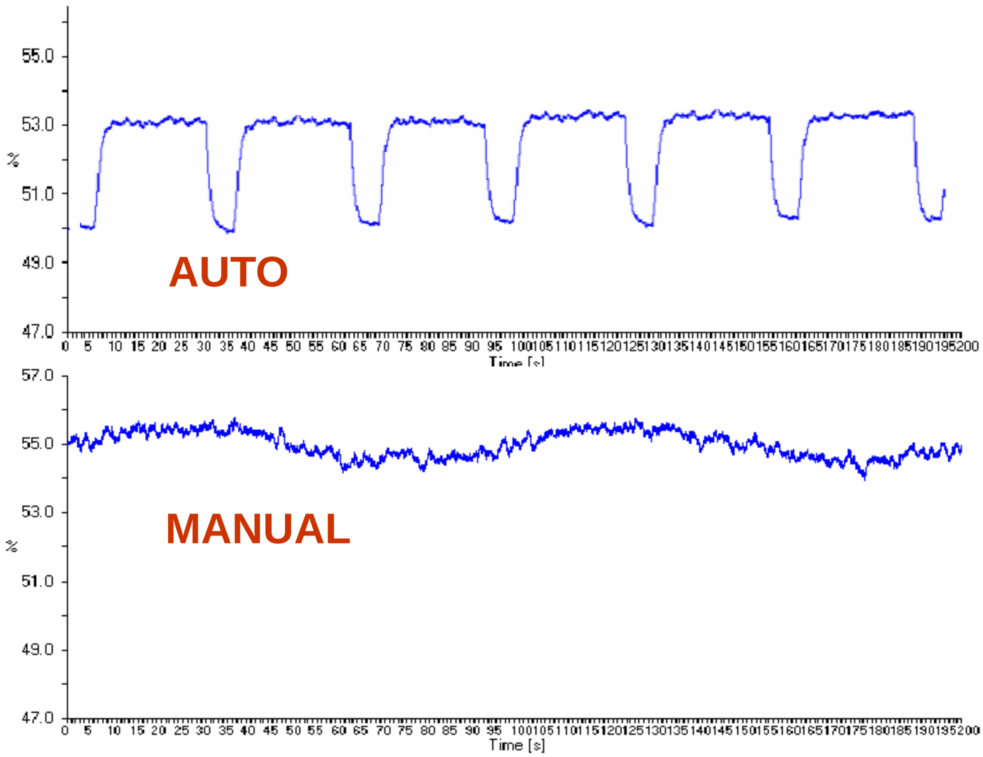
In any case, process variability costs you money.

Causes of High Variability

- **Composition of incoming raw materials**
- **Composition of incoming additives**
- **Process equipment**
- **Process control design**
- **Inadequate mixing/agitation**
- **Ineffective tuning of control loops**
- **Cycling loops**

There are a number of things that can cause high process variability. The end users are concerned with all of them, but for this discussion we will only address the last one, cycling control loops. Probably half the time there is a variability problem, it is caused by a cycling loop.

Cycling Control Loop



It is not uncommon to find a control loop that behaves like this. In automatic it cycles, and in manual it is reasonably stable. As a result it is not uncommon to find that a large percentage of the loops in a process plant are left in manual.

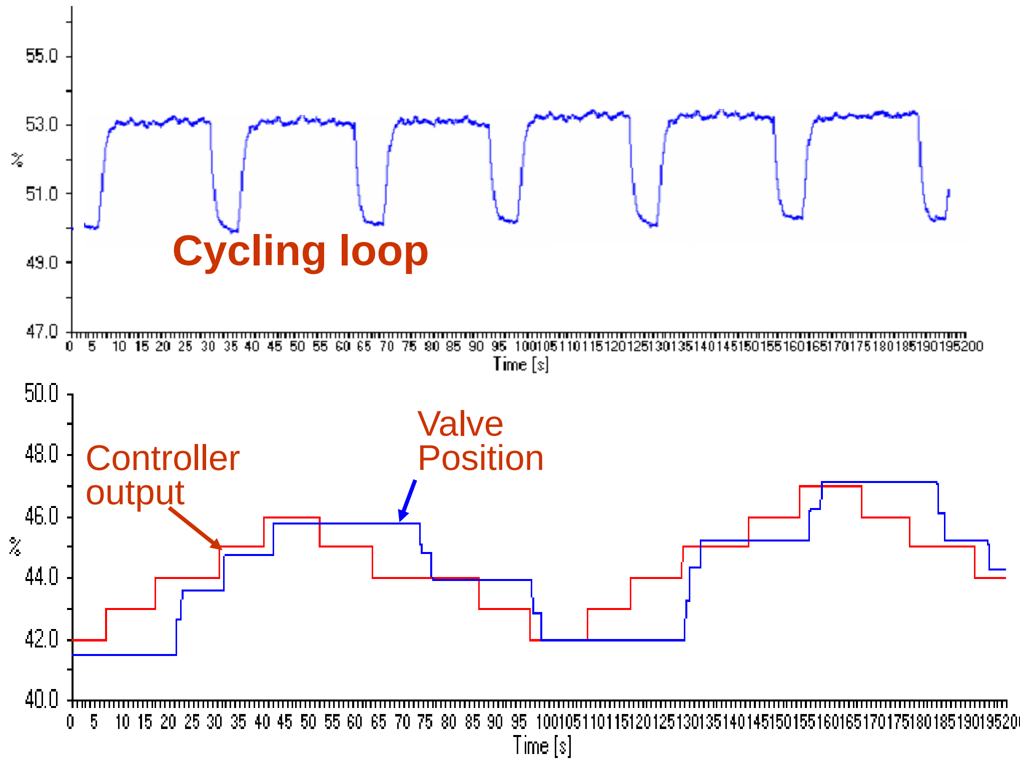
Causes of Cycling Loops

- **Tuning method**
  - **Aggressive tuning by “feel”**
- **Difficult process dynamics**
  - **Variable process gain**
  - **Variable dynamics**
  - **Dead time**
  - **Interacting loops**
- **Improperly selected or poor performing control valve**

There are a number of things that can cause a cycling loop. The end users are concerned with all of them, but for this discussion we will only address the last one.

About half the time, a cycling loop can be traced to the control valve.

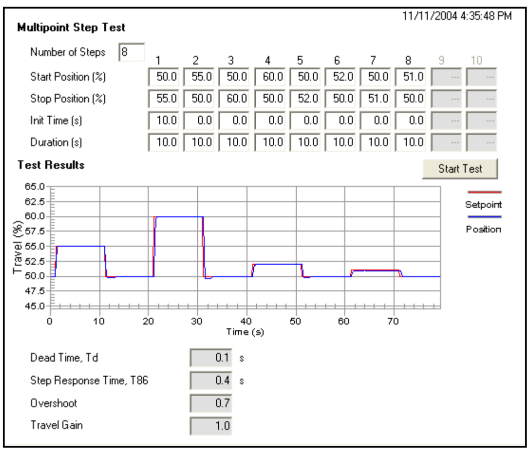
Valve Bump Test



In our example, the cycling was caused by a poorly performing control valve. The loop was put in manual and the control signal was stepped up and down in 1% steps and the valve's position response was observed. You can see that this valve is not capable of following the control signal. This type of inability to follow small changes in control signal results in cycling.

For good control you need a valve that:

- **Has its characteristic selected to match the process**
- **Is properly sized**
- **Has good dynamic performance.**

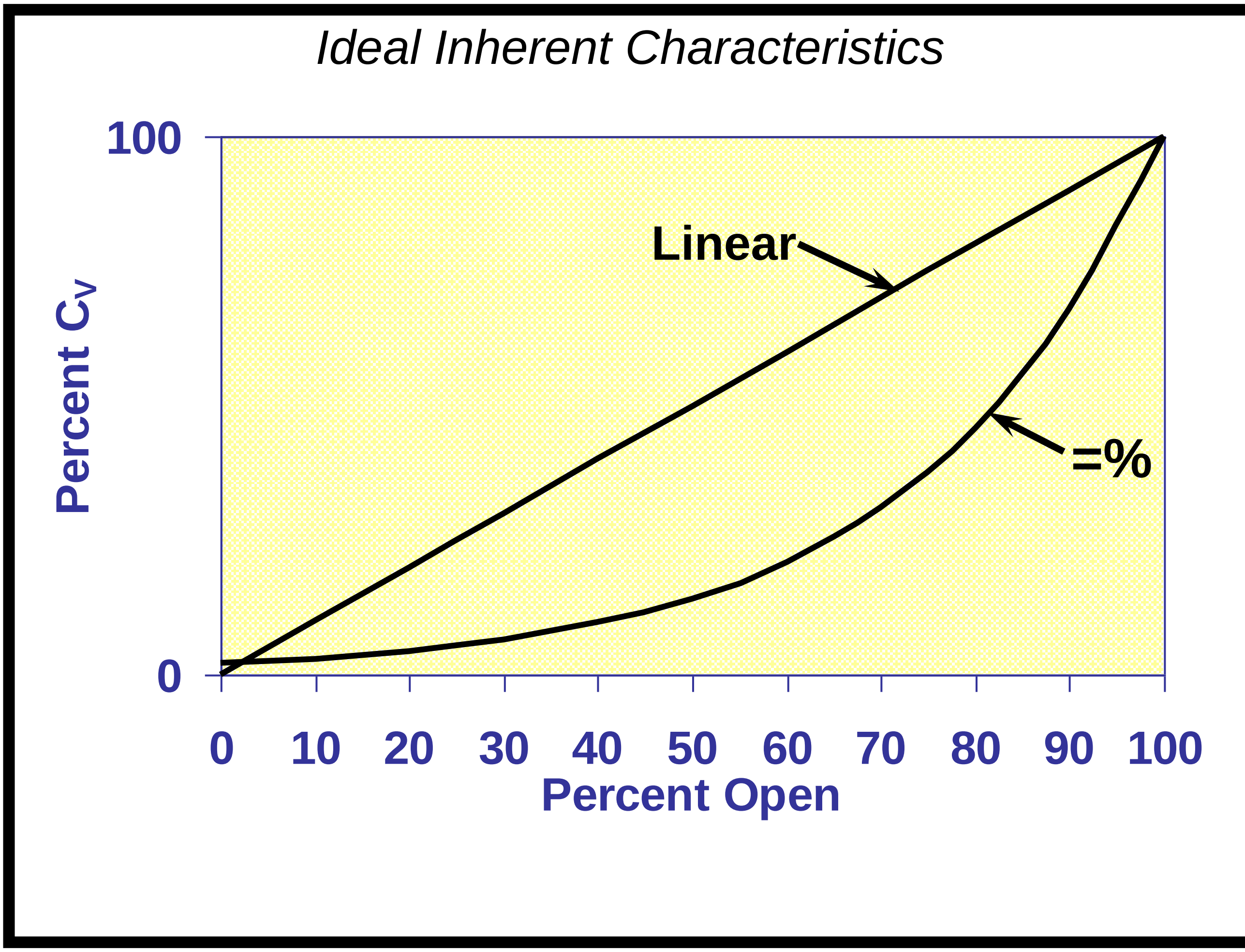


There are three areas that can affect a control valve's contribution to process variability.

Its flow characteristic needs to be selected to match the process.

The valve must be properly sized.

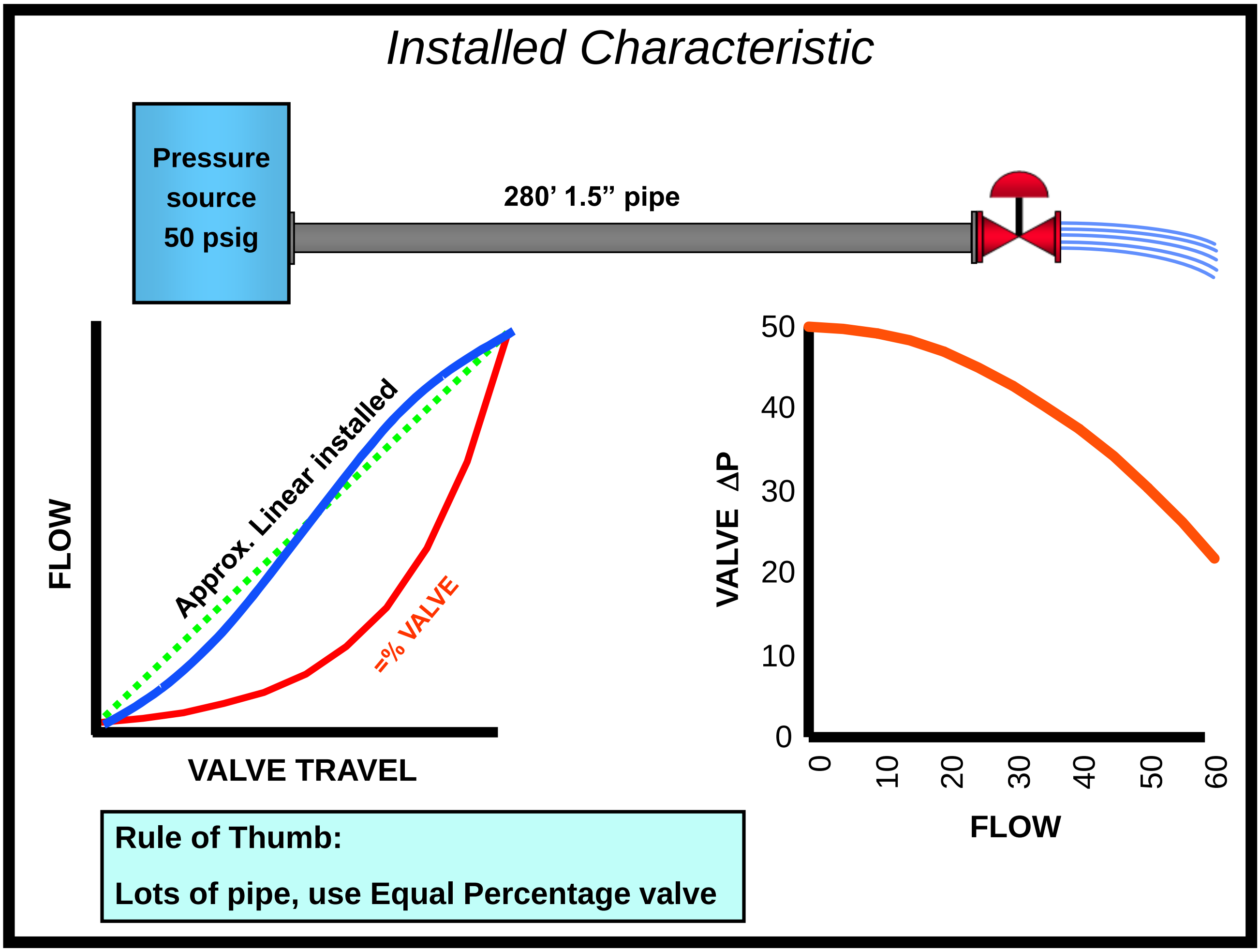
The valve needs to have good dynamic performance.



The two most common “inherent” characteristics are the Linear and the Equal Percentage. It turns out that the equal percentage characteristic is used 80% to 90% of the time and the linear characteristic is only used 10% to 20% of the time.

This may seem strange, since a study of control theory tells us that systems that behave in a linear manner are easier to control. If this is so, why is something as non-linear as the equal percentage valve used so often?

The answer is “because of the **installed characteristic**.”



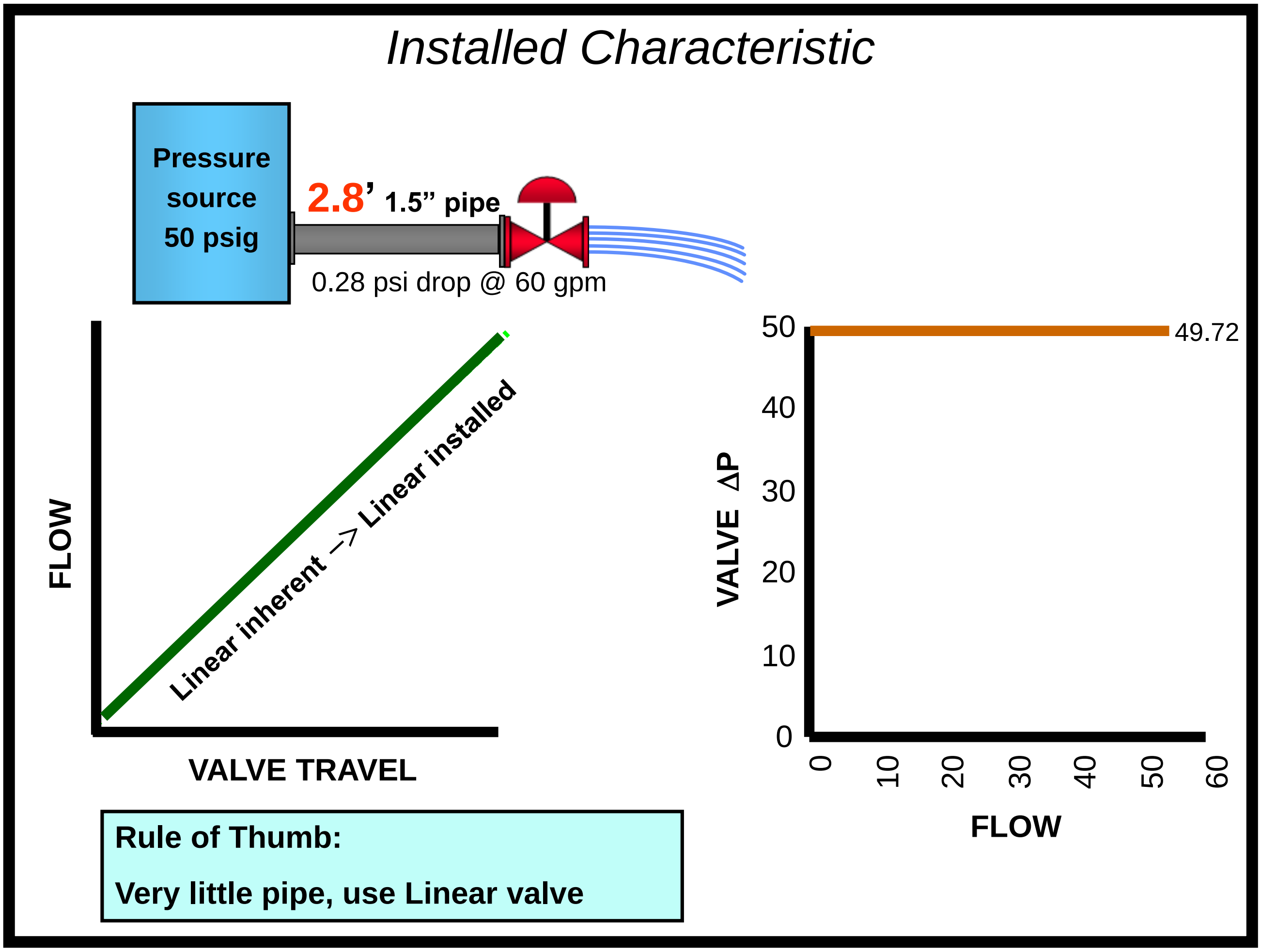
Let's see what happens when we install an equal percentage valve in this typical system with a lot of pipe. Starting with the valve 100% open, the flow would be at its maximum value. Then start closing the valve. As the flow decreases, the pressure drop across the valve increases (as shown in the Right-hand graph).

The increasing pressure drop partially resists the decreasing flow

The result is an INSTALLED characteristic that is very nearly linear. The reason that so many equal percentage valves are used is that most systems include a lot of pipe or other process equipment (centrifugal pumps have the same effect). Using equal percentage valves in these systems gives a nearly linear installed characteristic, which makes the system easier to control. We will see later why a linear installed characteristic makes the system easier to control.

The system characteristic graph and the equal percentage graph are not exact mirror images of each other, so the installed characteristic tends to be somewhat "S" shaped as shown here.

As a general rule of thumb, an equal percentage valve is the best choice for systems with a lot of pipe.



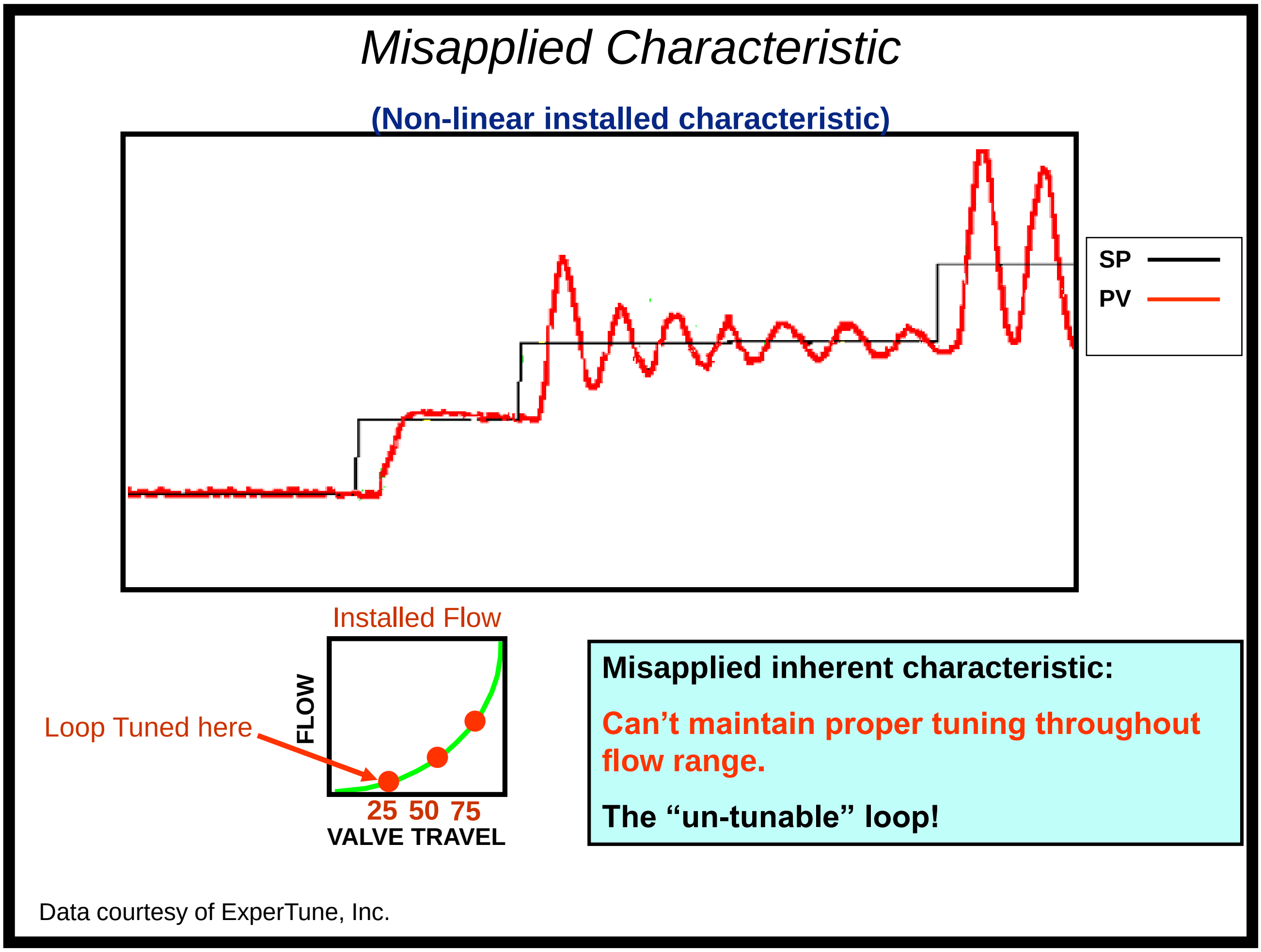
If we had only 2.8 feet of pipe instead of 280 feet, (1 one hundredth as much pipe) the pressure loss in the piping would be 1 one hundredth of the loss in the 280 feet of pipe. At 60 gpm the pressure loss would be about 0.28 psi instead of 28.

As the valve goes from zero to 100% open the pressure drop across the valve goes from 50psi to 49.72 and the graph of pressure drop across the control valve is essentially a flat line.

If we installed a valve with a linear inherent characteristic in this system, as the valve starts to close from 100% open the flow goes down, but the pressure drop remains constant, so the installed characteristic is the same as the inherent characteristic of the valve which is linear.

Remember that the way the inherent characteristic is determined by the manufacturer is to measure flow throughout the valves opening in a system where the pressure drop remains constant, so it is not surprising that the installed characteristic is the same as the inherent characteristic in a system where the pressure drop remains constant.

As a general rule of thumb, a linear valve is the best choice for systems with very little pipe.

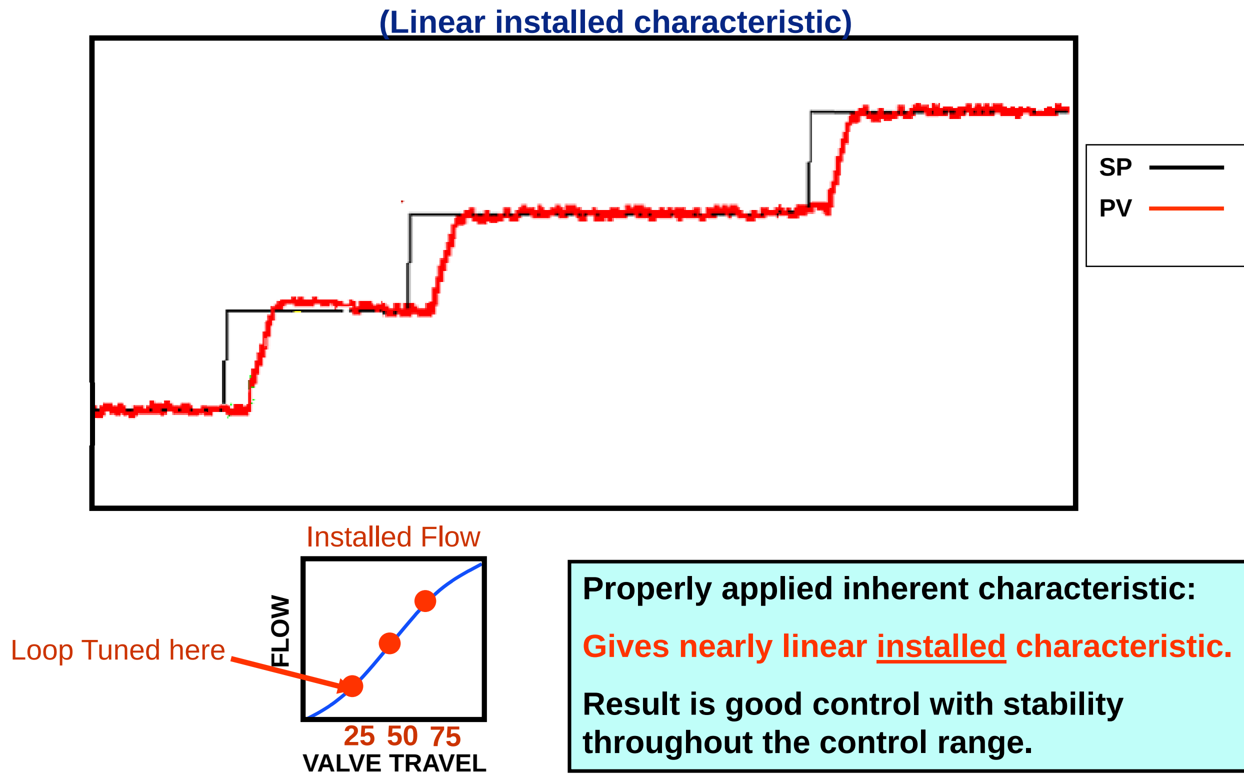


When a valve with the wrong inherent characteristic is put in a system, its installed characteristic will be non-linear as shown in the small graph. If the loop was tuned when the system was running at a control valve opening of 25%, and where the valve is fairly insensitive to changes in controller output, a high value of proportional gain would be required to get good response. When system demand increases to where the control valve is 50% open, the valve is more sensitive to changes in controller output and as a result the tuning parameters selected at a valve opening of 25% are too aggressive and a step change in set point results in an unstable response. When the system demand results in a control valve opening of 75%, the valve is very sensitive to changes in travel and the situation is even worse and a set point change results in extremely oscillatory response.

If instead, the loop was tuned when the control valve was 75% open, a lower value of proportional gain would have been used, and we would get fast stable response to a step change in set point, but if we then operated at lower loads, the response would be very sluggish.

If we had misapplied a linear valve in a system with a lot of pipe the situation would be the opposite of what is shown here. Just as the system with a lot of pipe pushes the equal percentage inherent characteristic upward into a linear installed characteristic, it would push a linear inherent characteristic upward into a quick opening characteristic. Now we would have a system that would be very sensitive at low valve openings and very insensitive at large openings. The system would still be difficult or impossible to tune so as to get fast stable response throughout the flow range.

Properly Applied Characteristic



Data courtesy of ExperTune, Inc.

Here we have a linear installed characteristic. Since the sensitivity of this system to changes in valve position remains constant, the same set of controller tuning parameters will give fast response with stability throughout the control range.

Valve Sizing

$$Cv = Q\sqrt{\frac{G}{\Delta P}}$$

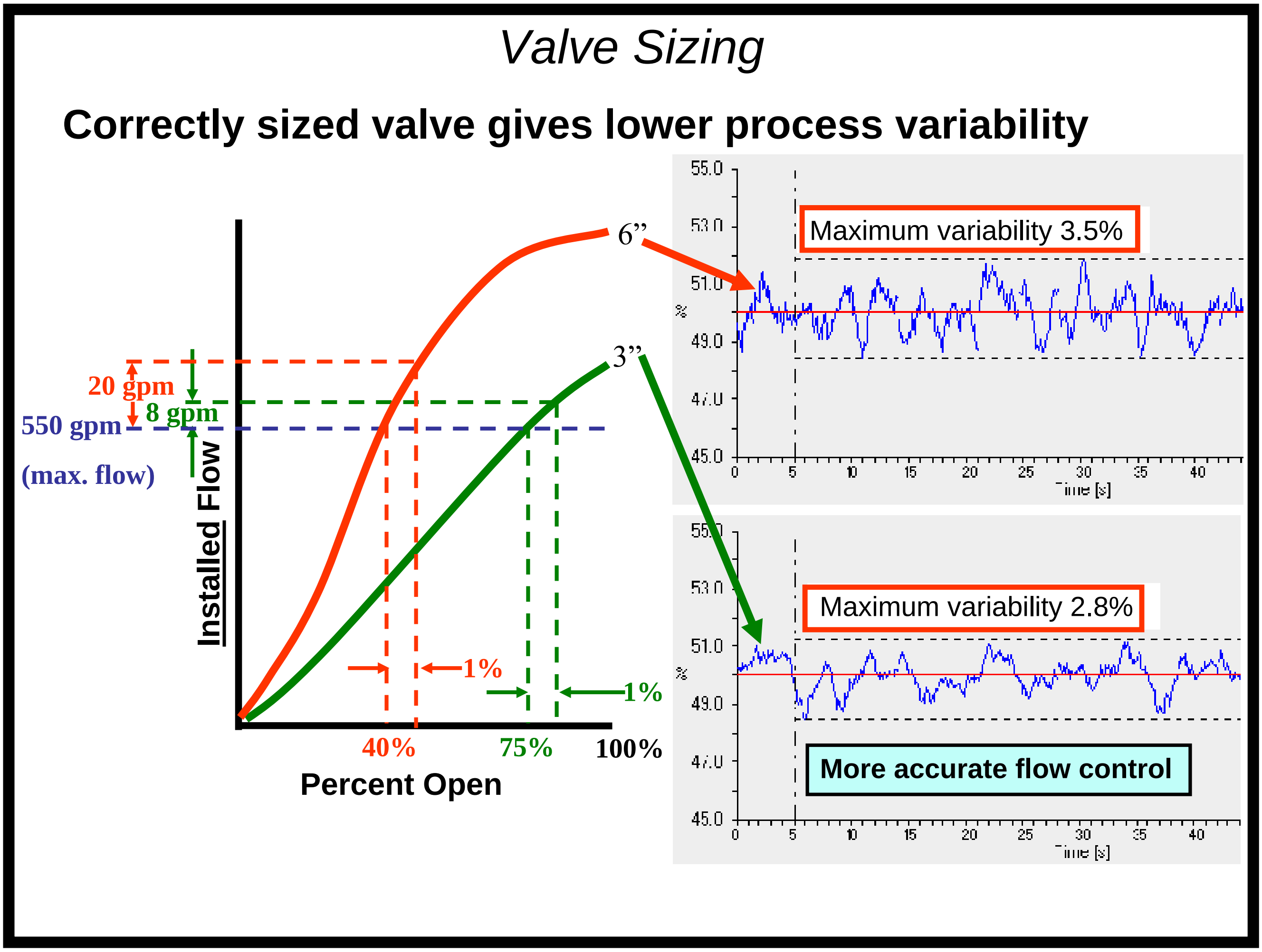
- Control Valve Sizing: The process of selecting a control valve that will do the best job of controlling the process.
- Most often by using a computer program
- Too small
  - Won't pass the required flow
- Too large
  - More expensive than necessary
  - Too sensitive
  - Difficult or impossible to adjust exactly to the required flow…

Next we will discuss the importance of proper control valve sizing.

This slide is pretty much self explanatory. It is meant as an explanation of what we mean by control valve sizing, and why both undersized valves and oversized valves are undesirable.

Over sizing is much more common than under sizing.

The biggest problem with an over sized valve is that it will be too sensitive, making good control difficult or impossible to achieve. This is demonstrated on the next page.



Several years ago we did a test in a flow lab comparing two valves, one oversized and one properly sized. It wasn’t these exact two valves, but it was a very similar application..

The results of the test are shown in the figure. The process variability was lower with the properly sized valve. .

Assuming both valves have the same amount of stickiness, the properly sized valve will be able to adjust the flow in smaller increments and therefore be able to adjust it more accurately to the required flow.

Control Valve Performance

Sensitivity (resolution)

Dead band

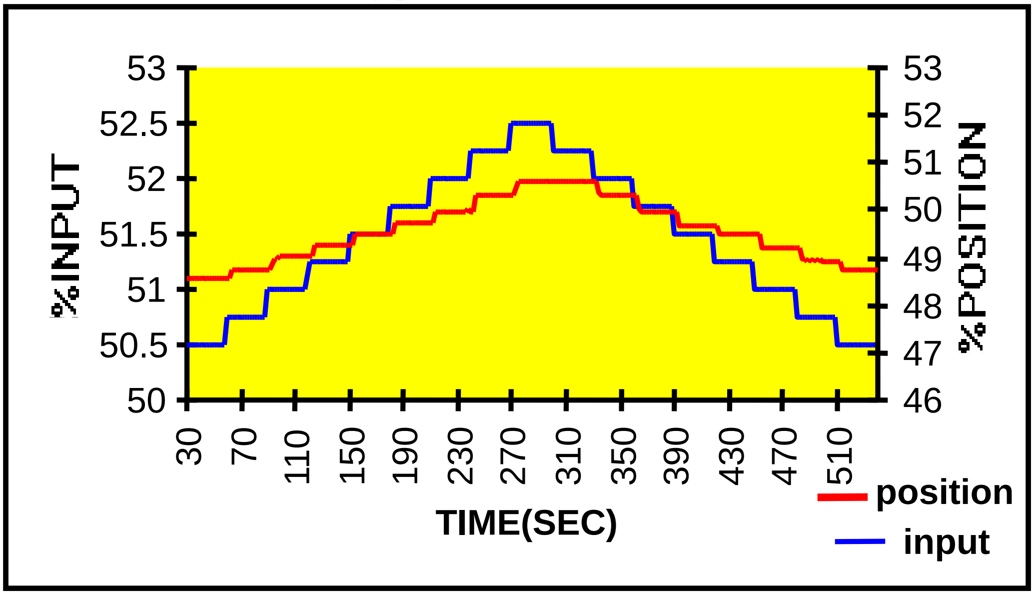
Speed of response

Control valve performance also has a major impact on the valve’s contribution to process variability.

The most important measures of performance are sensitivity (also called resolution) dead band, and speed of response.

Static Dead Band, Resolution (Stiction)

Typical Results



This is the test that is specified most often, the static dead band test, which also shows resolution. The sticky behavior of valves is often referred to as “Stiction.” It is the result of the interaction between static friction and dynamic friction. Static friction is usually much higher than dynamic friction. As a result. a valve tends to stick in place until enough pressure builds up in the actuator to break the static friction, then the valve moves quickly to the new position. Resolution is a measure of the smallest movement in the same direction that a valve is capable of. This is called a “static” test, because we always wait long enough after each step for any possible movement to take place. We don’t make any measurements while the valve is moving , but only concern ourselves with the valve’s (static) position after it has come to rest.

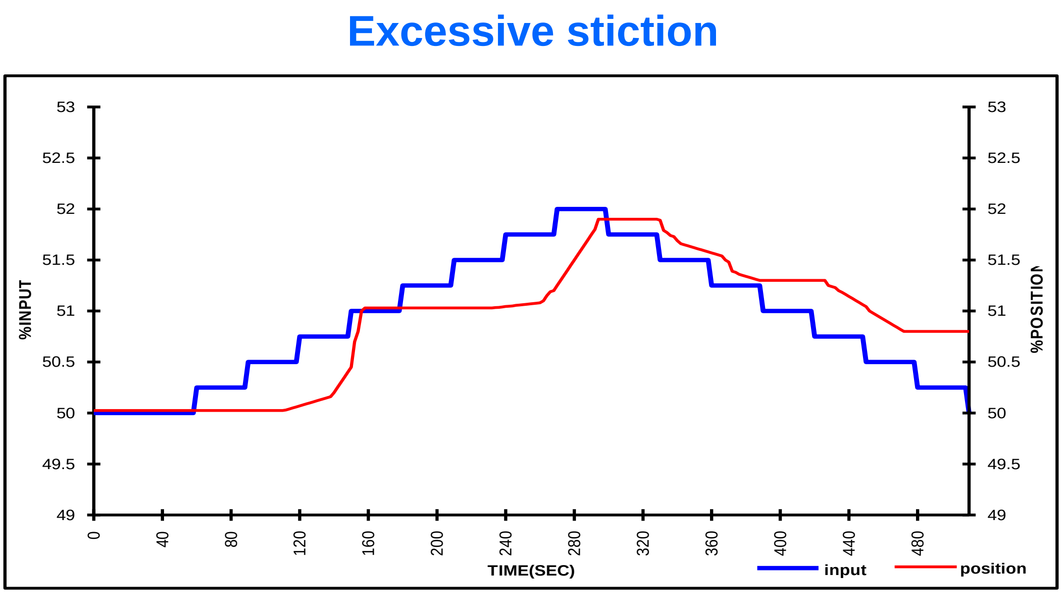
The control signal is stepped in one direction in very small steps. After each step there is a waiting period to make sure that there is time for the valve to make any move it is going to make before the next step is initiated. Observing the number of control signal steps that are required to make the move tells us how sensitive the valve is, and the term used to describe this is “resolution.”

After a number of steps in the same direction, the direction of the steps is reversed. Observing the number of steps required to initiate reversal of valve motion tells us what the “dead band” is.

In this example the step size is ¼%. In the same direction, this valve responds to each ¼% step, so it has a sensitivity or “resolution” of at least ¼%. Upon reversal, it took two of the ¼% steps before the valve started moving in the reverse direction, so this valve has a dead band of no more than ½%.

Note that the scales for the input and position are different so that the two graphs will be easier to differentiate from each other.

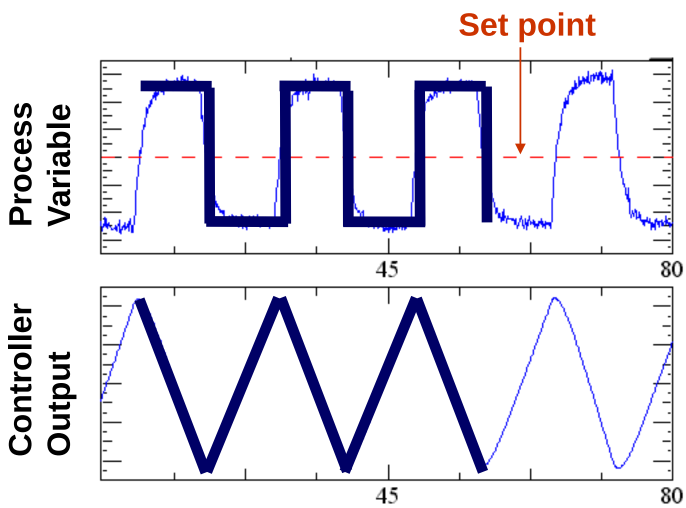
Static Dead Band, Resolution (Stiction)



This is an example of an excessively sticky valve.

Stiction

Excessive stiction
Limit cycle in closed loop control

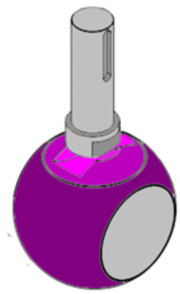


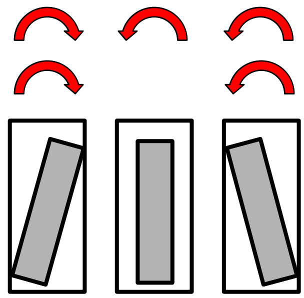
The result of excessive stiction in a closed loop control system is a limit cycle and process variability. Look at the process variable trace and the horizontal line that has been drawn over the PV trace at the upper left side of the graph. The valve is stuck and the process variable is above set point. The integral (or reset) action of the PID controller ramps the controller output in an attempt to correct the error until enough pressure builds up in the actuator to break the static friction. Because dynamic friction is lower than static friction, the valve moves quickly to a new position. Before the static friction broke, enough pressure had built up in the actuator, that the new process variable value is now below set point. As a result the PID controller reset action now starts ramping the controller output in the opposite direction in an attempt to correct the new error. The result is a “Limit Cycle.”

The characteristic of a limit cycle is that the process variable oscillates in a “square” wave fashion and the controller output oscillates in a “Saw Tooth” wave fashion.

Backlash and Dead Band

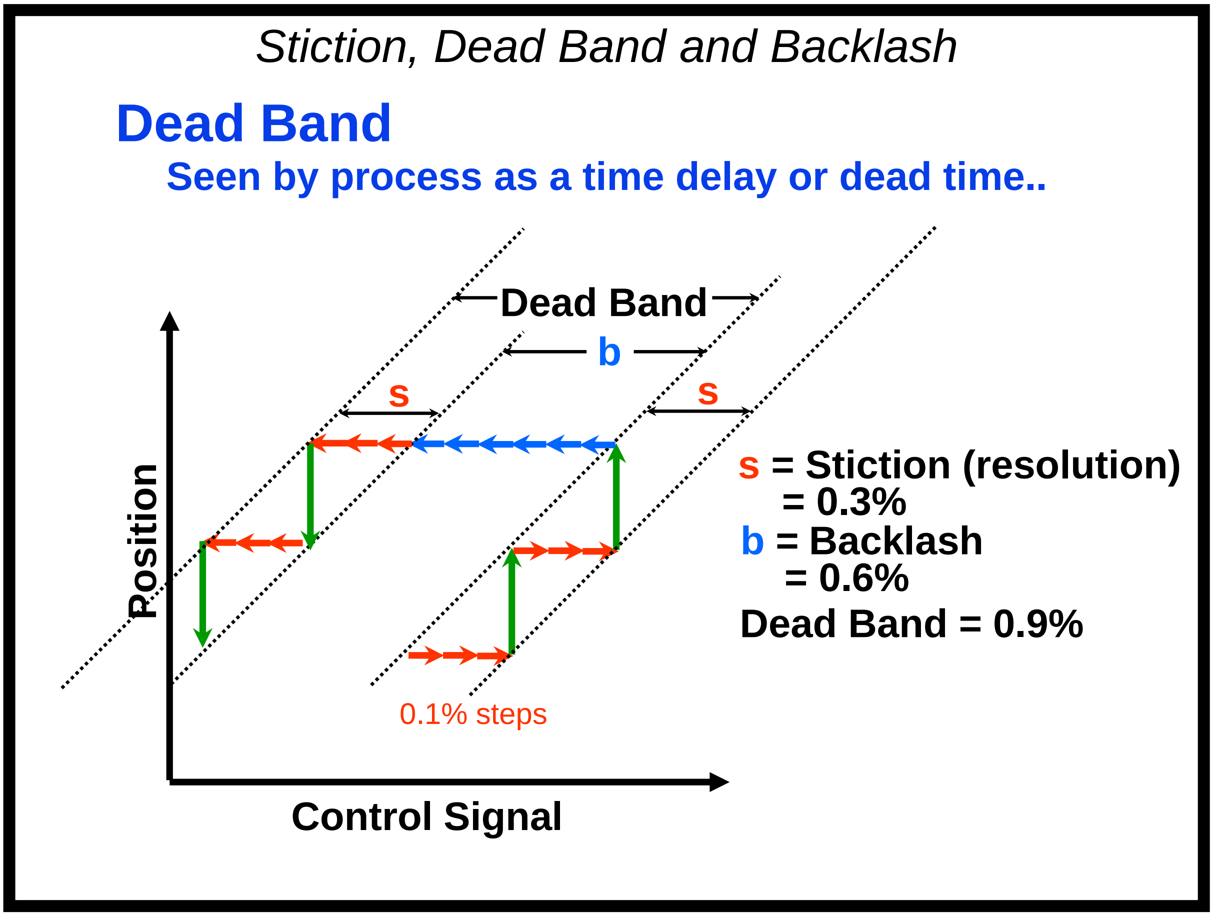
Backlash (Slop)





In addition to stiction, dead band can be the result of backlash, a technical term for “slop.” Backlash can result when there is looseness in the connection between two parts. The example here is the connection between the stem and the ball in a ball valve. Many ball valve designs require that the ball be able to float downstream into the seat when the valve is closed in order to create a tight seal between the seat and ball. A common design that allows the stem to turn the ball, but also permit the ball to float slightly when the valve is closed is pictured in the figure. The ball has a slot into which a “blade” on the end of the stem fits. In order to allow manufacturing tolerances and to assure that the ball can freely float into the seat there will be a small gap between the blade and the slot in the ball. In the figure, this gap has been greatly exaggerated.

In the left-hand drawing of the slot and blade, the stem is turning clockwise and driving the ball in the clockwise direction. If the stem stops turning, then continues in the clockwise direction the ball will immediately follow the stem without delay. If the stem stops, then reverses direction, turning in the counterclockwise direction it will take a finite amount of time for the blade to take up the space between the stem and ball before the ball starts turning in the counterclockwise direction. The result is backlash which contributes to dead band.



This page discusses in more detail the details and definitions relating to stiction, resolution and dead band.

Stiction and Backlash are two important factors that determine how well a control valve can control the process.

For the purposes of this discussion, position is graphed as a function of control signal, and time is left out. Remember, the resolution and dead band test is a static test and we don't concern ourselves with time.

Stiction is the result of the interaction between static friction (which is high) and dynamic friction which is much lower. If you push lightly on a valve stem, nothing happens. If you push a little harder, still nothing happens. Finally when you push hard enough to overcome the static friction the valve moves fairly fast, because the dynamic friction is lower.

Backlash is lost motion due to looseness or "slop." Imagine a loose actuator-to-valve stem coupling. Once the "slop" is taken up, as long as motion continues in the same direction, the valve will follow the actuator. When the actuator reverses direction, the valve does not move until the actuator has moved enough to take up the play.

The test for stiction and backlash sends a series of small steps of control signal to the valve, first in one direction, then in the reverse direction.

Look at the three right-hand facing arrows on the lower right side of the graph: In our example it takes three 0.1% steps of control signal before there is enough actuator force to break the static friction. Under the lower dynamic friction the valve then moves 0.3% and catches up to the control signal.

If we send some more 0.1% control signal steps in the same direction, it takes another three of them before the static friction breaks and the valve again moves 0.3% and catches up to the control signal.

In the example the stiction is 0.3%, that is the control signal must change by 0.3% before the valve will move. This represents the resolution or precision with which it is possible to position this valve.

If there is some "slop", play or lost motion in the valve's linkages or actuator linear-to-rotary conversion, it will take a number of the small control signal steps in the opposite direction just to take up this play before the actuator can even begin exerting force to try and break the static friction. You can see that it takes 6 of the 0.1% steps, so in this example the backlash is 0.6%

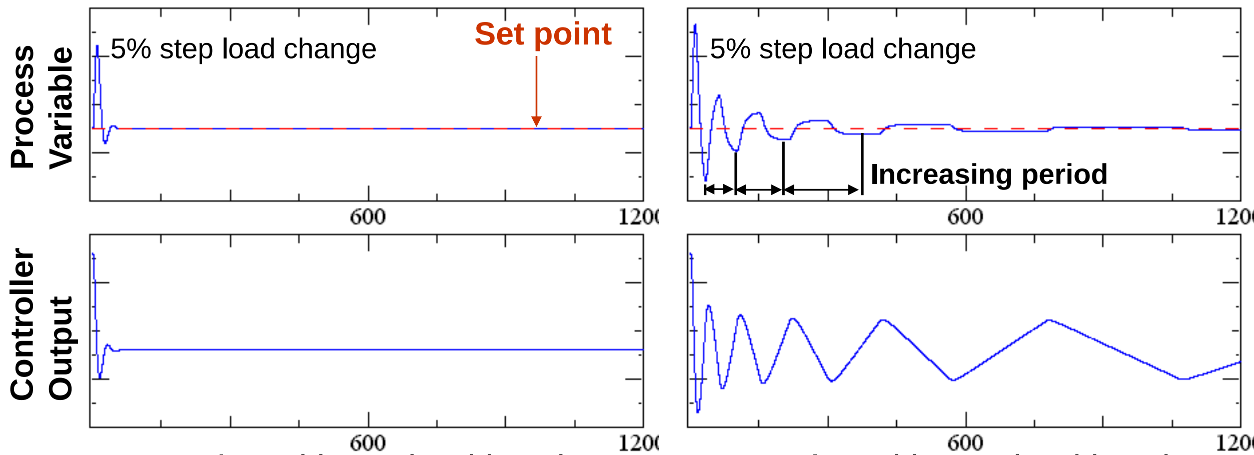
Now it will take a number of additional control signal steps for the actuator to exert enough force or torque to break the static friction (stiction) and start the valve moving in the reverse direction and a similar process is repeated. The dead band, which is defined as the amount the control signal must be changed upon reversing signal direction before motion begins, equals the backlash plus the stiction. Here the dead band is 0.9%

NOTE that the dead band does not degrade the resolution. This valve can be positioned within 0.3% of where you want it positioned even upon reversal of control signal. The dead band does require a certain amount of time for the reset action of the controller to drive the valve through it, and this will show up as process dead time. Dead time is one of the worst things you can add to a process from the viewpoint of making the process difficult to control.

Sometimes performance specifications specify the "Total Hysteresis" which is total width of the figure, in this case it is 1.2%.

Dead Band

Seen by process as a time delay or dead time..



Valve without dead band

Valve with 3% dead band

As error decreases, controller reset action is slower and it takes longer for the controller output to drive through the dead band. This has the effect of an increasing time delay.

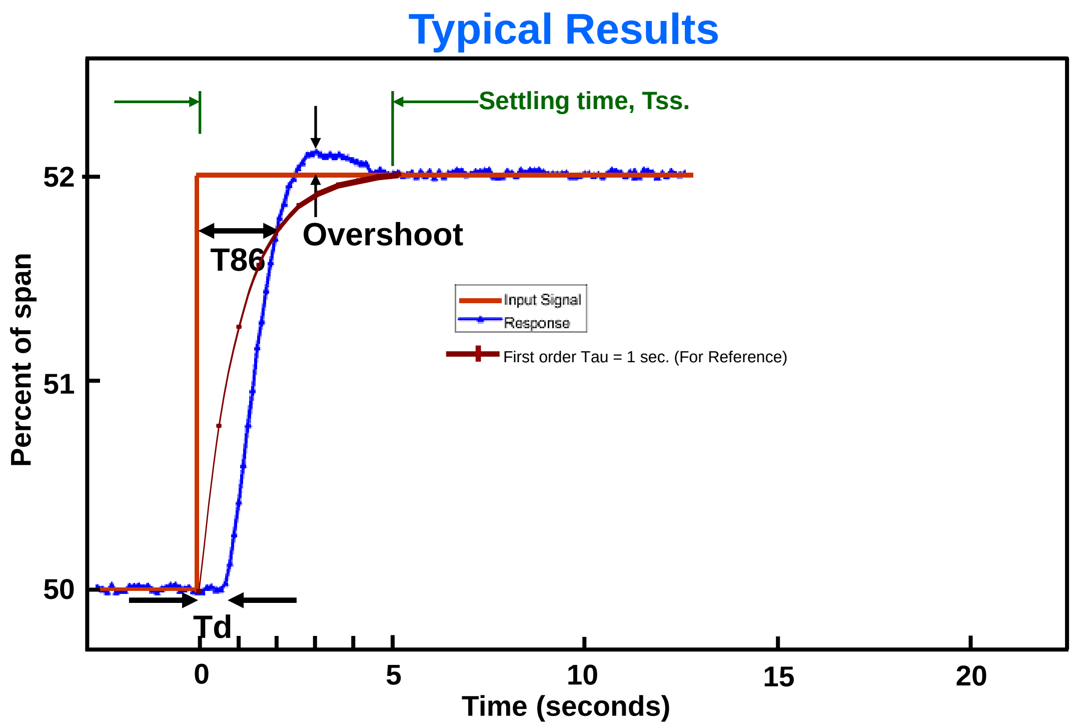
This page demonstrates the effect of dead band on process control.

Both pairs of graphs represent the response to a 5% step change in load. The pair of graphs on the left is a process where the valve has zero dead band and the pair of graphs on the right represent a process where the valve has a dead band of 3 percent. In both cases the controller is tuned for fast but robust response to load changes.

The increased instability is caused by the fact that when the controller sees the upset to the process that requires the valve to move in the opposite direction from its last movement, the reset action starts driving the valve right away but takes some time for the dead band to be overcome and for the valve to start moving. During this time no correction is being applied to the process upset. Each time a correction is made, the error is less. The smaller the error, the slower the controller reset action is, and the longer it takes for the controller to overcome the valve's dead band. This appears as an increasing time delay or dead time.

(In case you are wondering, the full scale on the Process Variable graph is 4% of span and the time scale is in seconds.)

Step Response



The next most important measure of control valve performance after resolution and dead band is the the speed of response to step changes in control signal. This is a “dynamic” test because we are concerned with what the valve is doing while it is moving, and we record the entire movement.

This is a typical response. There will be some dead time before anything happens. There may or may not be overshoot.

In the past, two parameters were commonly used to measure speed of response, T63, the time required for the valve to respond through 63% of its total response, and T98, the time required for the valve to reach 98% of its final position.

T63 was chosen as being the equivalent of the time constant of a first order system. The word “time constant” was not used because control valve response is rarely first order.

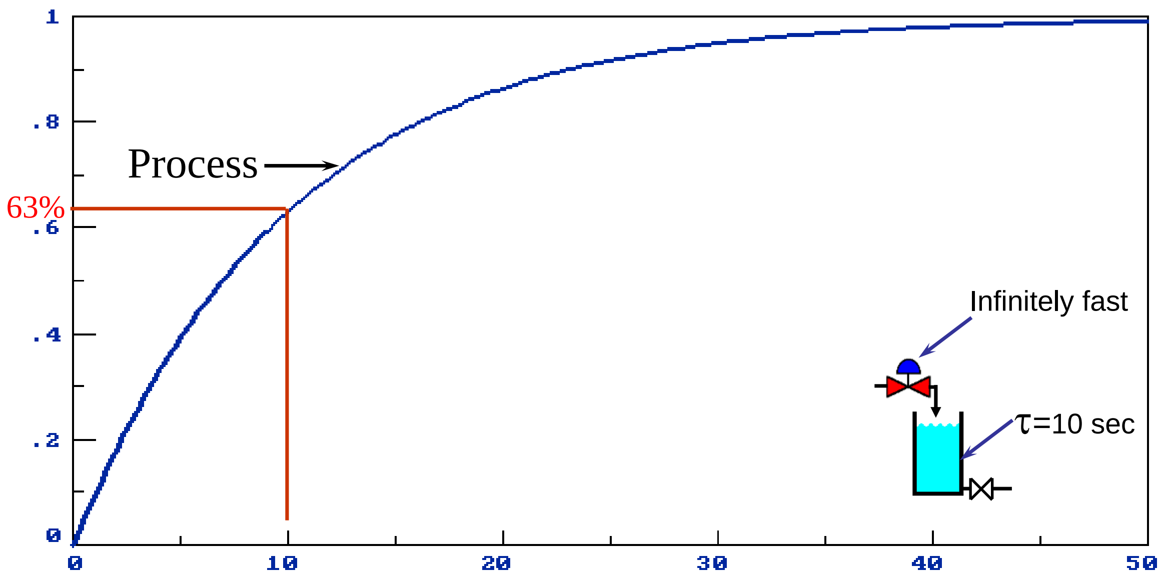
For reference, a first order response with at T86 (two time constants) and settling time, similar to the T86 and settling time of the valve’s response, has been drawn in to emphasize that the valve response in not first order.

ISA S75.25.01 and Entech 3.0 now use a single parameter, T86 (which corresponds to two time constants of a first order system).

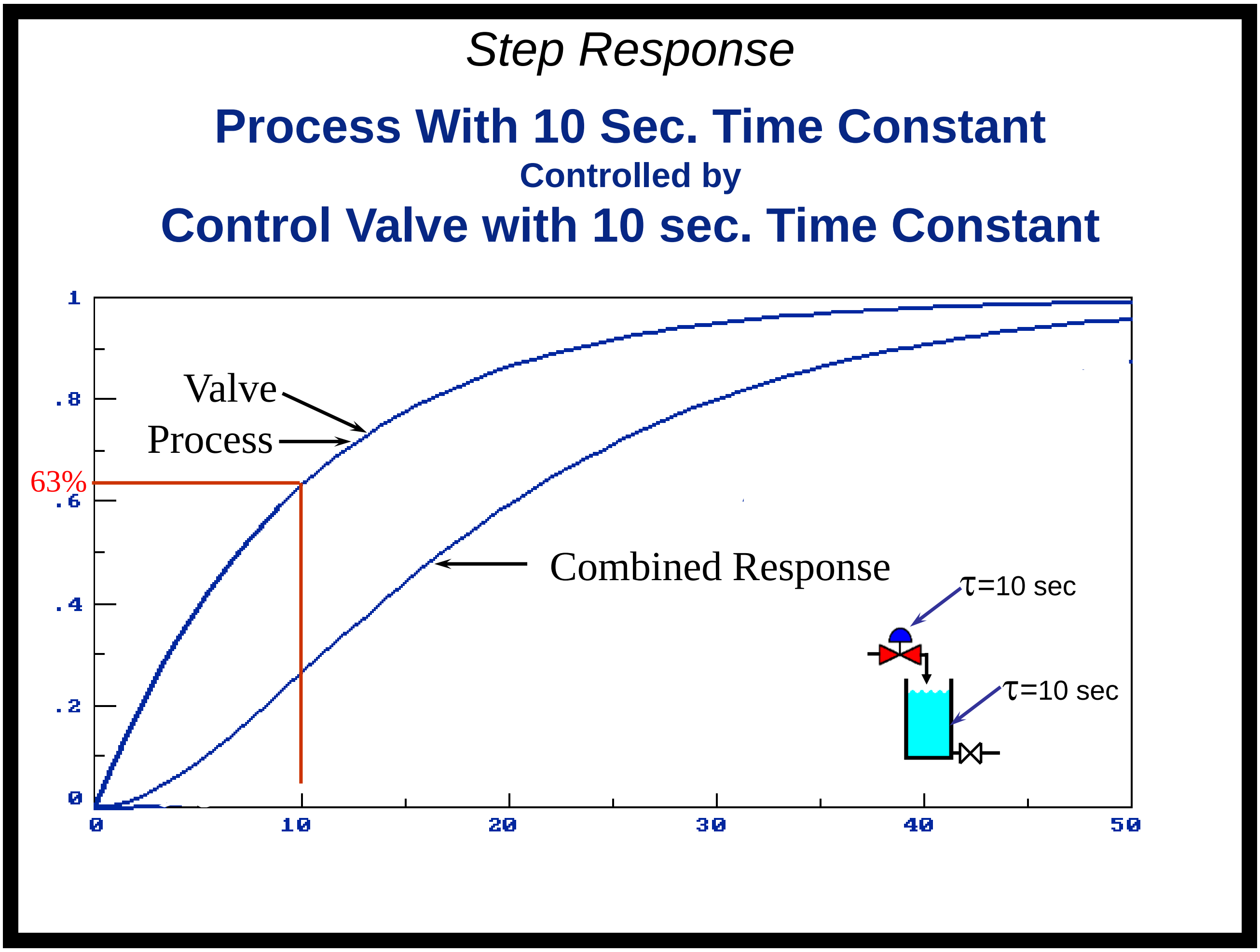
Note that T86 is measured from the time of the step change in control signal.

Step Response

Process With 10 Sec. Time Constant



Let's see why we are concerned with the speed of response of a control valve. This graph shows the response of a first order system that has a time constant of ten seconds, that is a process that responds through 63% of its full response in ten seconds.



Although control valve response is usually more complex than first order, it is allowable, for the sake of comparing the effect of valves with various speeds, to treat them as first order systems.

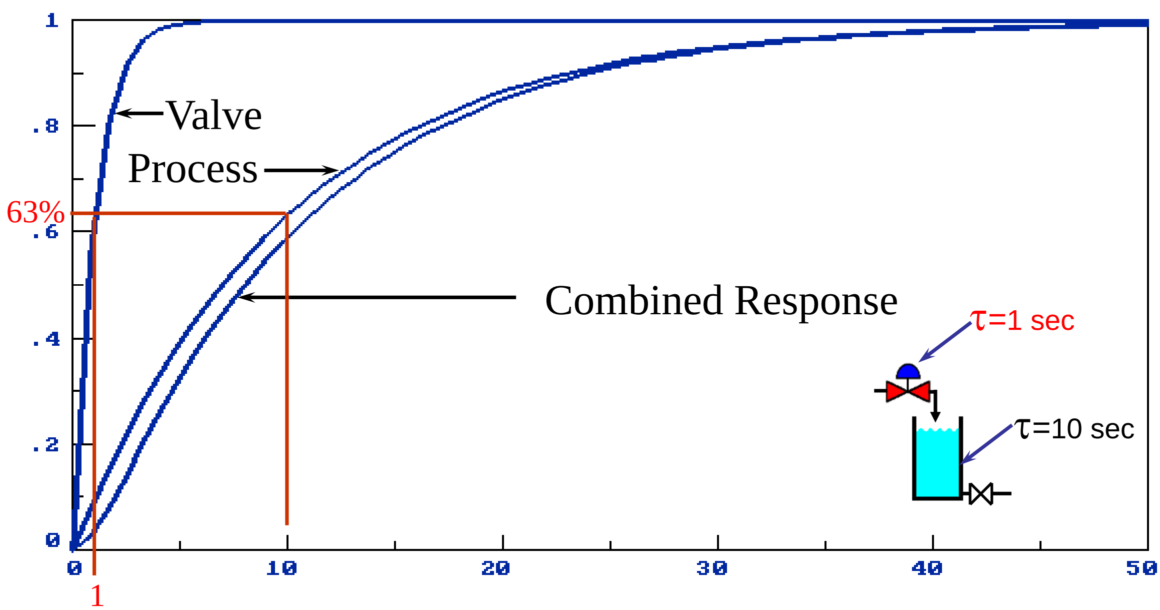
If our 10 second system were controlled by a valve with a 10 second time constant, the overall response would be as shown here. The combined response is much slower than what the process is capable of by itself.

Step Response

Process With 10 Sec. Time Constant

Controlled by

Control Valve with 1 sec. Time Constant



Here the same ten second process is being controlled by a valve with a 1 second time constant. Because the valve is much faster than the process by itself, the combined response is nearly as fast as the speed with which the process could respond with an infinitely fast valve.

Typically, a valve that is five times faster than the process will have little effect in slowing the process from responding as quickly as it is capable of.

Control Valve Dynamic Performance Guidelines

for critical processes

Resolution (stiction): ≤ 0.5%
Dead band: ≤ 0.5%
Speed of response:

- **Fast loops:**
  - **Valve Td ≤ 20% of desired closed loop process time constant**
  - **Valve T86 ≤ 40% of desired closed loop process time constant** **(This is equivalent to saying that the valve should be 5 times faster than the desired closed loop process response time)**
  - **Valve settling time ≤ desired closed loop process time constant**
- **Slow loops: Not important**

Step overshoot: 20% maximum..

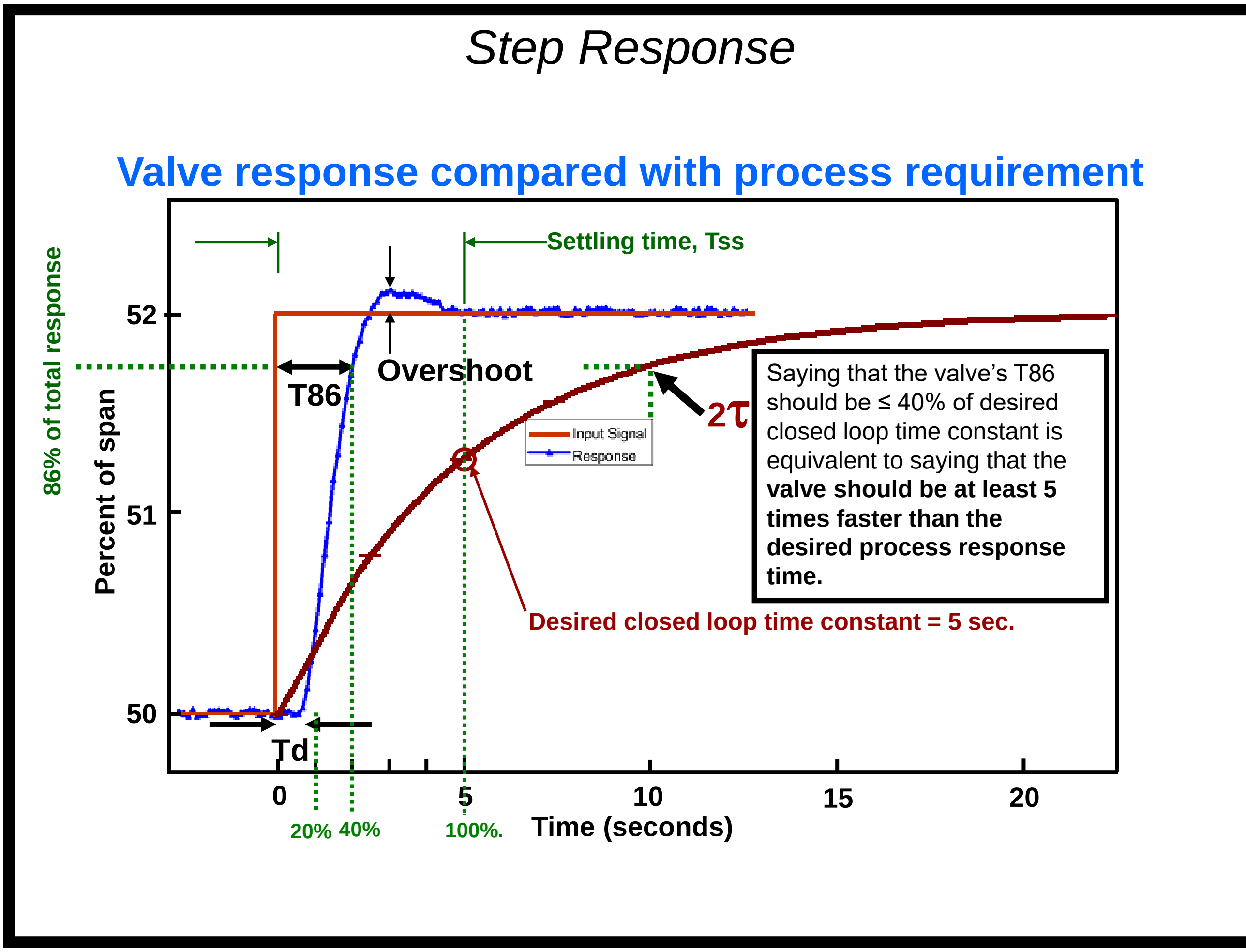
These are some guidelines suggested by a process control expert that we know, for valves in processes where very good control is required.

The values given for Resolution and Dead band agree with the “nominal” values given in the Entech “Control Valve Dynamic Specification” Version 3.0 as does the maximum overshoot.

Twenty percent overshoot means 20% of the step size, for example for a 10% step, the overshoot should not exceed an overshoot of 2% of scale.

The recommendation for speed agrees with suggestions in the ISA valve performance technical report.

The ISA Standard, ISA S75.25.01 does not specify acceptance criteria, only the definitions and the methodology for the tests.



This slide is included to demonstrate why the recommended speed of response criteria that I suggest make sense.

This is the same valve we looked at earlier, and it just meets the requirements for a process where the desired closed loop time constant is 5 seconds.

The dead time is just under the recommended 20% of the desired closed loop time constant. The dead time is over in time to have little impact on the overall process response.

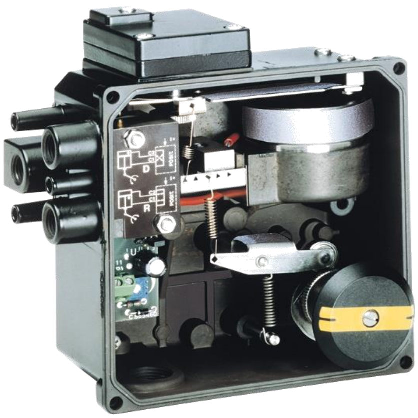
The valve reaches 86% of its full travel after only 40% of the desired closed loop time constant (T86). It can be seen that the valve is way ahead of when the process needs to reach 63% of its final value, and even farther ahead of when the process needs to reach its 2 time constant (86%) value. Because the valve reaches 86% of its total response in two seconds, and the desired process response should reach 86% of its total response in ten seconds, it is equivalent to saying that the valve is by 5 times faster than the desired process response time.

At the early stage of the full response, a small overshoot will contribute very little to an overshoot in the process.

The valve response has settled to its final value after one desired process time constant, long before the process is expected to reach its final value.

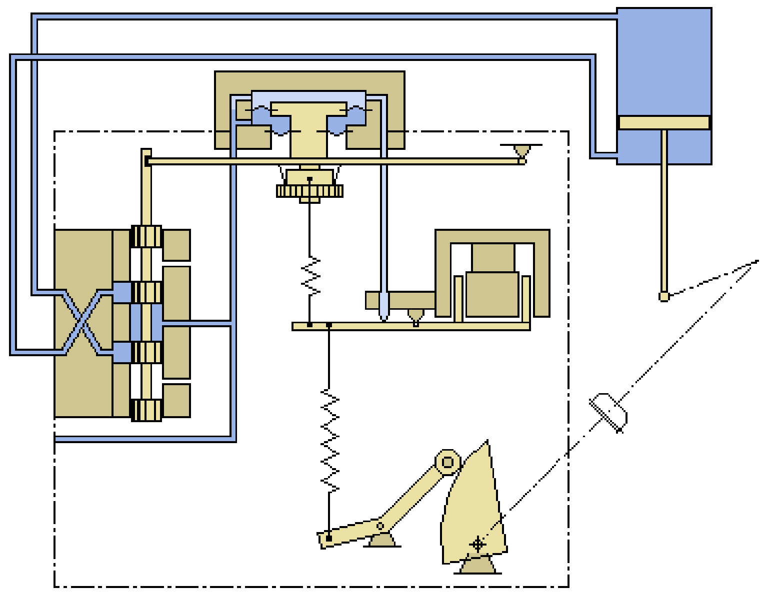
Smart Digital Positioners Can Reduce Process Variability

Standard Analog Positioner



Proportional only control:

Beams, levers, springs, cam.



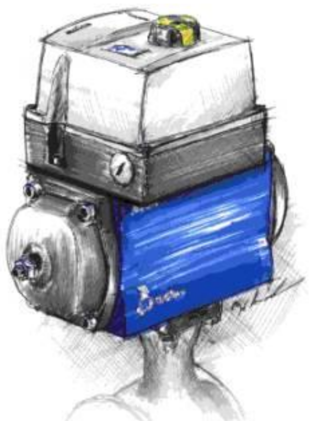
Shown as a reference point here is a standard mechanical analog electrical positioner.

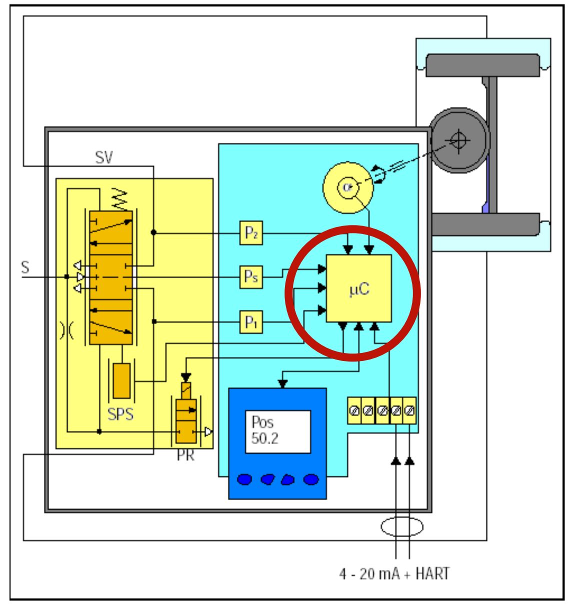
Standard analog positioners are proportional only controllers that use a combination of beams, levers, springs and a cam to compare control signal with actual position and then meter the correct amount of air to the actuator to make the valve go to the required position.

Smart Digital Valve Positioner

$$\left[m_{red}\cdot\frac{dx}{d\psi}+J\cdot b(\psi,\mu)\right]\cdot\frac{d^2\psi}{dt^2}+m_{red}\cdot\frac{d^2x}{d\psi^2}\cdot\left(\frac{d\psi}{dt}\right)^2+\left[f_m\cdot\frac{dx}{d\psi}+f_v\cdot b(\psi,\mu)\right]\cdot\frac{d\psi}{dt}$$

$$+\,\operatorname{sign}\left(\frac{d\psi}{dt}\right)\cdot b(\psi,\mu)\cdot M_{v\mu}(\psi,\mu,\Delta p)+b(\psi,\mu)\cdot M_d(\psi,\Delta p)-F_m(P_AP_B)=0$$





Advanced control algorithm:

Based on a detailed mathematical study of control valve dynamics.

In a digital positioner the levers, links and springs are replaced by a microprocessor. With a microprocessor you can do a lot of things that could not be done with the links levers and springs.

The microprocessor can run an advanced control algorithm that is based on a rigorous mathematical study of control valve dynamics.

At the top of the screen you see the mathematical model of a control valve that was developed by Neles' research department.

For those who are interested, the things that need to be considered in constructing a mathematical model that describes control valve dynamics are(listed in order of the terms in the equation):

1. Rotational inertia of things that rotate like balls and butterfly valve disks.

2. Linear inertial of things like actuator pistons and diaphragm plates.

3.The dampening forces. (These are the frictional forces that are proportional to velocity and is what keeps springy things from continuing to bounce up and down.

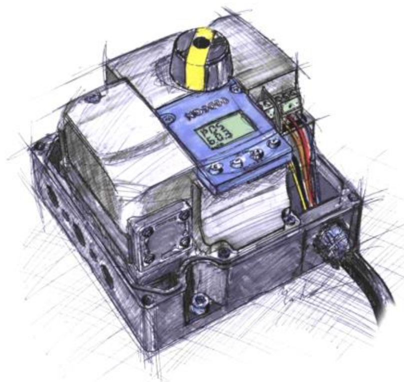
4. Frictional torque

5. Dynamic torque

6. Actuator forces.

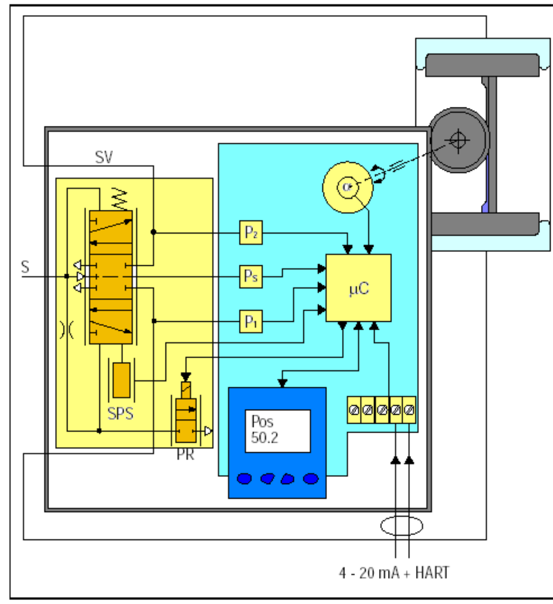
Actually, this is just the outline or skeleton of the model. The actual model takes many pages, and is proprietary. For example, in order to calculate the frictional and dynamic torque you need to know the pressure drop, delta p. This requires a non steady state equation of flow in the system. In order to know the pressures in each side of the piston actuator, you need the complete mathematical model of the positioner, and a thermodynamic model of the actuator.

Advanced Control Algorithm



Advanced control algorithm corrects for frictional effects:

- Algorithm learns static friction levels
- Pressure “boost” breaks static friction quickly
- Continually updated during operation

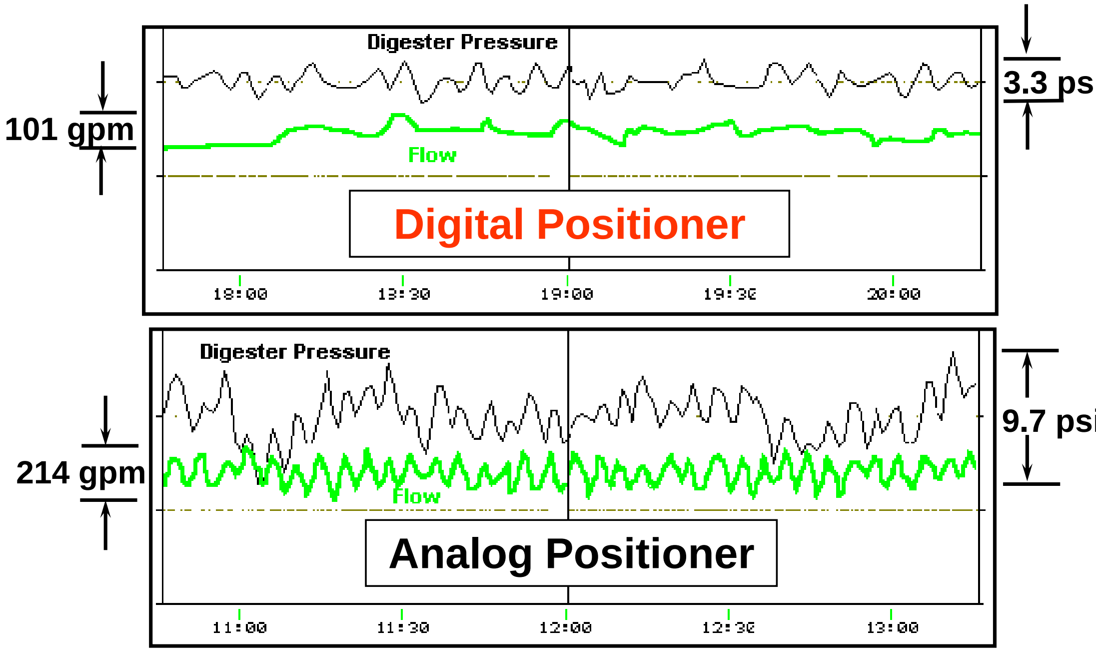


When the control algorithm detects a position error, it not only starts sending air to the actuator in proportion to the size of the error, but it also sends an additional shot of air based on what it learned during the self tuning process was necessary to quickly break the static friction.

As soon as the positioner sees that the valve is moving, it knows that the friction is lower (dynamic friction), so it cuts back on the amount of air going to the actuator. This gets the valve moving quickly, reducing dead time, while avoiding overshoot.

Each time the positioner moves the valve, it compares the amount of pressure boost it used to what was actually needed. If the actual friction has changed, the positioner updates its information on the pressure “boost” required by the valve’s current condition.

Reduced process variability



These screen shots from a customer's DCS with an analog electrical positioner and and a Neles digital positioner on the same valve show a significant reduction in the variability of both the flow of liquor into the digester and the digester pressure

In the process in the example it is important to reduce process variability because pulsations in liquor flow and digester pressure effect the quality of the product coming out of the digester and the cost of downstream processing.

Control Valve Material Considerations

The following pages are intended as a general overview of control valve materials.

They are not intended as engineering recommendations or advice.

It is the end user’s responsibility to make the final decision regarding the selection of materials based on their knowledge of their process.

Properties

- **Mechanical and Physical Properties**
  - **Yield Strength** – Stress required to cause 0.2% permanent deformation.
  - **Hardness** - Materials resistance to penetration or indentation. Often used to estimate sliding wear resistance and resistance to abrasion and erosion.
  - **Toughness** - Materials ability to absorb energy without fracture. Impact testing. A tough material is not brittle.
- **Wear Properties**
  - **Sliding Wear (galling)**
  - **Erosive Wear**
  - **Cavitation Wear**
- **Corrosion Properties..**

Yield Strength: Stress (force per unit area) that produces 0.2% permanent deformation.

Hardness: Resistance to penetration or indentation. Usually measured by loading an indenter into the material and measuring the depth of penetration or surface area of the indentation. The more the penetration, the less hard the material is

Toughness: The ability of material to absorb energy and deform plastically before fracturing. Usually measured using impact tests. Number of foot pounds required to fracture a stressed sample. A tough material is NOT BRITTLE.

Sliding wear “galling”. Occurs when the heat and pressure between irregularities on the surface is high enough to cause localized welding and transfer of material between the sliding parts. PREVENTION: Use materials with dissimilar composition (makes welding less likely). Use materials with different surface hardnesses.

Erosive wear: Caused by high velocity fluid impingement or erosive particles if flow medium.

Cavitation wear: This is discussed at length in the incompressible flow presentation.

Corrosion properties: All materials resistance to corrosion is of course extremely important.

Pressure Effects

- **Inlet and Differential Pressure**
  - **Cavitation**
  - **Flashing**
  - **Erosion**
    - High pressure drop → high velocity
    - Abrasive media, wet steam
  - **Erosion-Corrosion**
    - High velocity → erode passivated layer

The inlet pressure and differential pressure determine whether a liquid will cavitate or flash, both of which cause damage. The damage mechanisms are discussed in the incompressible flow presentation.

Pressure differential affects the velocity and thus the potential for erosion caused by entrained solids, or in the case of steam, moisture drops in wet steam.

Erosion-corrosion results when high velocity fluid streams wash away the “passive layer” that protects many stainless steels.

Temperature Effects

Elevated Temperature

- **Yield Strength**
- **Creep**
- **Coefficient of Thermal Expansion**
  - CS, alloy steel, 410SS → low
  - 300 series SS → high
- **Graphitization of Carbon Steel**
  - > 800F carbides decompose into carbon and iron reducing strength and toughness
- **Sensitization of Stainless Steel**
  - At high temperatures, such as encountered when welding, there is a risk that the chrome will form chrome carbides with any carbon present in the steel. This reduces the chrome available to provide the passive film and leads the potential for corrosion on the portion of the stainless steel affected by the high temperature

Increasing temperature reduces yield strength.

At high temperatures a phenomenon called "CREEP" comes into play. Normally, metals are elastic (a certain stress produces a predictable strain. When the stress is removed, the material goes back to its original dimension. At high temperatures the behavior becomes inelastic. The strain will slowly increase with time. Hence the name "creep"

Coefficient of thermal expansion: When heated, metals expand in a predictable way. Carbon steels, alloy steels and 400 series stainless steels have low thermal expansion coefficients. 300 series stainless steels have high thermal expansion rates. This must be kept in mind when selecting valve components: Globe cage must match body thermal expansion. In ball valves, th ball must match body to avoid seizing.

Graphitization: Temperatures above 800 F cause the CARBIDES in carbon steel to decompose into iron and graphite, reducing strength and toughness. Addition of Chromium and Molybdenum make the carbide phase more stable in Chrome Molly steel and 300 series stainless steel.

Sensitization - At high temperatures, such as encountered when welding, there is a risk that the chrome will form chrome carbides with any carbon present in the steel. This reduces the chrome available to provide the passive film and leads the potential for corrosion on the portion of the stainless steel affected by the high temperature.

Common Valve Body Materials

- Carbon Steel
  - Low Material Cost
  - \- 20°F (-28°C) to about 800°F (425°C) Max.
- Chrome Moly (alloy steel)
  - Material cost = 1.5 x Carbon Steel
  - To 1200°F (648°C) *
  - Greater Erosion (Flashing) Resistance
- Stainless Steel
  - Material cost = 2.5 x Carbon Steel
  - Cryo. to 1500°F (815°C) *
  - Greatest Erosion Resistance
  - Corrosion Resistant..

* Class 150 flanged ratings terminate at 1000ºF (538ºC).

Carbon steel should not be used above 800 F. It can become brittle at temperatures below -20 deg F.

Raw casting costs are compared to that of carbon steel.

Chrome Moly sees use in power plants due to resistance to flashing damage and high temperature steam. It sees use in refineries because of the elevated temperatures encountered there. Also many high temperature refinery applications are not compatible with 316 ss.

316 ss is corrosion resistant in a great number of applications. It is used in a majority of refinery applications. It is also the most erosion resistant and has the highest temperature capability of the common body materials. 316 is also good for cryogenic temperatures (-150 to –460), where CS and Chrome Moly become brittle.

Stainless Steel

- Selected for corrosion resistance and high temperature properties
- 12% Chrome (minimum)
  - Forms protective layer of Chromium Oxide (Cr_2O_3)
- Three forms
  - Ferritic (430) Fe, Cr (Not normally used in valves)
  - Martensitic (400 series) Fe,Cr,C
  - Austenetic (300 series) Fe,Cr,C,Ni

Body Material Ratings ANSI B16.34

VALVES — FLANGED,
THREADED, AND WELDING END

ASME/ANSI B16.34-1988

TABLE 2-1.1
RATINGS FOR GROUP 1.1 MATERIALS

A 105(a)	A 515 70(a)	A 675 70	A 672 B70(a)
A 216 WCB(a)	A 516 70(a)	A 696 Gr. C	A 672 C70(a)
A 350 LF2(d)	A 537 Cl.1(d)		

NOTES:
(a) Permissible, but not recommended for prolonged usage above about 800°F.
(d) Not to be used over 650°F.

TABLE 2-1.1A STANDARD CLASS

Temperature, °F	Working Pressure by Classes, psig							
	150	300	400	600	900	1500	2500	4500
−20 to 100	285	740	990	1,480	2,220	3,705	6,170	11,110
200	260	675	900	1,350	2,025	3,375	5,625	10,120
300	230	655	875	1,315	1,970	3,280	5,470	9,845
400	200	635	845	1,270	1,900	3,170	5,280	9,505
500	170	600	800	1,200	1,795	2,995	4,990	8,980
600	140	550	730	1,095	1,640	2,735	4,560	8,210
650	125	535	715	1,075	1,610	2,685	4,475	8,055
700	110	535	710	1,065	1,600	2,665	4,440	7,990
750	95	505	670	1,010	1,510	2,520	4,200	7,560
800	80	410	550	825	1,235	2,060	3,430	6,170
850	65	270	355	535	805	1,340	2,230	4,010
900	50	170	230	345	515	860	1,430	2,570
950	35	105	140	205	310	515	860	1,545
1000	20	50	70	105	155	260	430	770

This is an example of a page from ANSI B16.34 which gives pressure/temperature ratings of a broad range of materials used for valve bodies.

Body Material Ratings

Carbon steel

TABLE 2-1.1A STANDARD CLASS

Temperature, °F	Working Pre			
	150	300	400	600
−20 to 100	285	740	990	1,480
200	260	675	900	1,350
300	230	655	875	1,315
400	200	635	845	1,270
500	170	600	800	1,200
600	140	550	730	1,095
650	125	535	715	1,075
700	110	535	710	1,065
750	95	505	670	1,010
800	80	410	550	825
850	65	270	355	535
900	50	170	230	345
950	35	105	140	205
1000	20	50	70	105

Chrome moly

TABLE 2-1.13A STANDARD CLASS

Temperature, °F	Working Press			
	150	300	400	600
−20 to 100	290	750	1,000	1,500
200	260	750	1,000	1,500
300	230	730	970	1,455
400	200	705	940	1,410
500	170	665	885	1,330
600	140	605	805	1,210
650	125	590	785	1,175
700	110	570	755	1,135
750	95	530	710	1,065
800	80	500	665	995
850	65	440	585	880
900	50	355	470	705
950	35	260	350	520
1000	20	190	255	385
1050	20(1)	140	190	280
1100	20(1)	105	140	205
1150	20(1)	70	90	140
1200	20(1)	45	60	90

316 SS

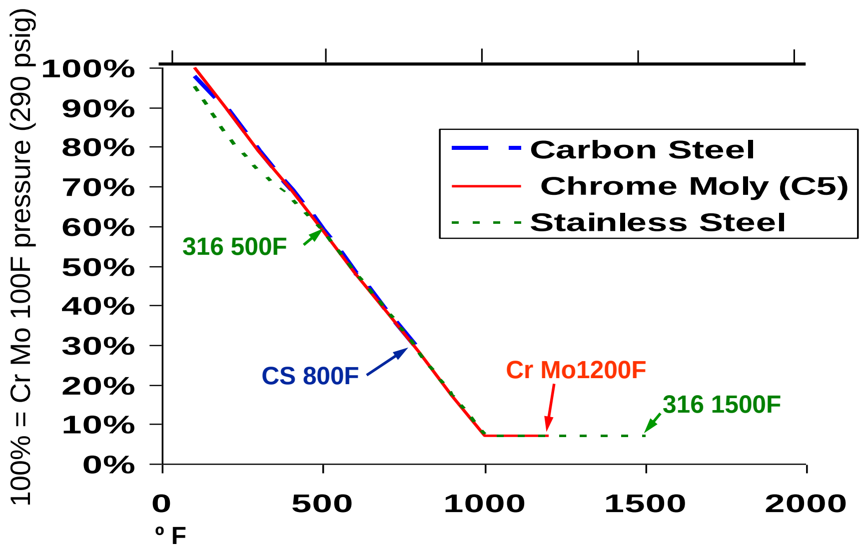
TABLE 2-2.2A STANDARD CLASS

Temperature, °F	Working Press			
	150	300	400	600
−20 to 100	275	720	960	1,440
200	240	620	825	1,240
300	215	560	745	1,120
400	195	515	685	1,030
500	170	480	635	955
600	140	450	600	905
650	125	445	590	890
700	110	430	575	865
750	95	425	565	845
800	80	415	555	830
850	65	405	540	810
900	50	395	525	790
950	35	385	515	775
1000	20	365	485	725
1050	20(1)	360	480	720
1100	20(1)	325	430	645
1150	20(1)	275	365	550
1200	20(1)	205	275	410
1250	20(1)	180	245	365
1300	20(1)	140	185	275
1350	20(1)	105	140	205
1400	20(1)	75	100	150
1450	20(1)	60	80	115
1500	15(1)	40	55	85

It is difficult to compare the tabular data of the common valve body materials, but examining them graphically shows some interesting things. (See next two pages.)

Body Material Ratings

ANSI 150



Cr Mo & 316 flange ratings for class 150 valves terminate at 1000F

In Class 150 the pressure/temperature curves of the three common materials are quite similar.

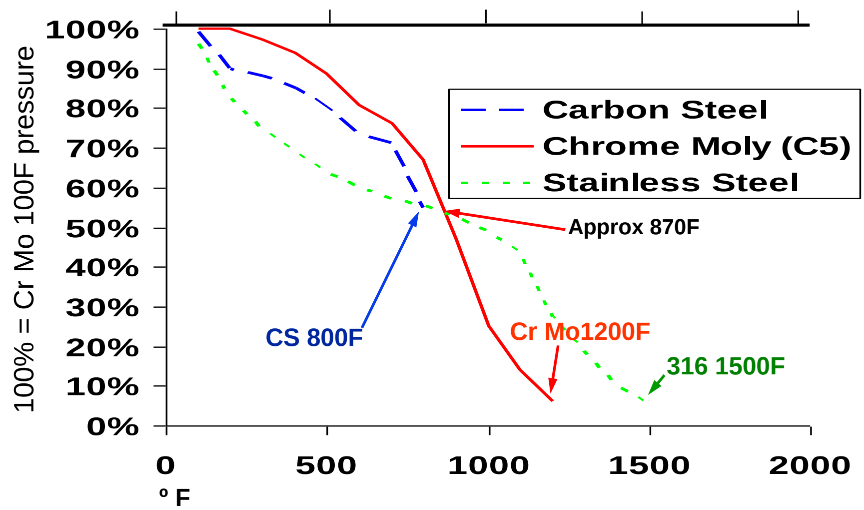
At 100 F, CrMo is 5 degrees higher than carbon steel, then CrMo and CS track each other to 800 F where CS ends.

316 is only slightly lower at lower temperatures, then starting at 500F it tracks the other two as far at they go.

100% on the graph is the 100 F rating for CrMo.

Body Material Ratings

ANSI 300 and up



In Class 300 and above the three materials have quite different pressure/temperature curves.

In Class 300 and above, the shapes of the curves for each pressure class are identical, only the scale is different.

100% on the graph is the 100 F rating for CrMo.

Body Material Temperature Limits

	F		C	
Cast Iron	-20	410	-28	210
Ductile Iron	-20	650	-28	340
Carbon Steel (WCB)	-20	800	-28	425
Carbon Steel (LCB)	-50	650	-45	340
CrMo (WC6)	-20	1000	-28	537
CrMo (WC9)	-20	1050	-28	565
CrMo (C5, C12)	-20	1200	-28	648
SS 304, 316	-425	1500	-253	815
Alloy 20	-50	300	-45	148
Aluminum	-325	400	-198	204
Bronze	-325	550	-198	287
Inconel	-325	1200	-198	648
Monel	-325	900	-198	480
Hastelloy C	-325	1000	-198	537
Titanium		600		315

This is a list of the temperature limits for a wide range of body materials.

Erosion Resistance

Bronze
Alloy 20
Monel
Hastelloy B & C
316 SS
K Monel
17-4 ph
416 SS
Inconel
Alloy 6 hard facing
Chrome & Tungsten Carbide
Ceramic

Note:Top to bottom: worst to best.

This is a list from the ISA Control Valve Handbook (1976), listing relative resistance to erosion.

Relative Resistance to Cavitation Damage

Material	Hours Tested	Index*
Stellite 6 over 316	120	20
17-4 PH 45 Rc	12	2
316 SS	6	1
Carbon Steel	2.25	0.38
Brass	0.5	0.08
Aluminum	(2 Min.)	0.006

*** 316 SS is the reference. The others were tested until they showed approximately the same amount of damage as did the 316 SS sample after 6 hours of testing.**

From ISA Control Valve Handbook, second edition, 1976.

316ss was tested for 6 hours under cavitating conditions. Then the others were tested until they showed approximately the same degree of damage and the time as shown was recorded.

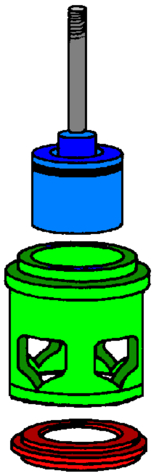
The index was determined by dividing the times required to produce damage in the various materials by the time required to produce equivalent damage in 316.

Alloys

Alloying elements are added to basic steels to enhance corrosion resistance, hardness and toughness.

- **Carbon** - carbon is the principal hardener in steel. The more carbon that is added (up to 1.2%), the harder it gets.
- **Molybdenum** -. Molybdenum adds toughness and increases corrosion resistance to chlorides.
- **Chromium** - Protects against corrosion and adds heat resistance.
- **Sulfur** - sometimes added in controlled amounts for easier machining ands welding.
- **Nickel** - improves corrosion resistance and toughness and helps austenitic stainless steels to maintain their austenitic structure.
- **Silicon** - principal deoxidizer used in steel making. It also increases strength and hardness.
- **Vanadium** - Adds toughness and fatigue resistance.
- **Manganese** - contributes to strength and hardness

Valve Trim Material - Internal parts of the valve



Trim Materials vs. Body Materials

- Stronger
- Special characteristics
  - Resist Galling
  - High wear resistance
  - Erosion resistant
  - Corrosion Resistant
- Treated or Coated
  - Chrome Plated, Stellite overlay
- Expensive relative to body material..

Typical Trim Materials (Ball Valves)

Typically selected based on strength & level of corrosion resistance

- Balls and seats
  - 316 SS
  - 410 SS (used in some high temp refinery processes, very expensive
  - Typically coated
  - Usually dissimilar to prevent galling..

Typical Trim Materials (Ball Valves)

Trim Coatings

- **Hard Chrome (HCr)** Balls
  - Coating process – Electroplated
  - Hardness – 64-69 HRC
  - Corrosion resistance similar to 316 SS
  - Not for Chlorides, pH<2, erosive or abrasive media
  - Max. temp – 840 F
- **Cobalt based hard facing (Stellite)** Balls, seats
  - Coating process – PTA (Plasma transferred arc welding)
  - Hardness – 36-43 HRC
  - Resistant to abrasive wear, erosion and better corrosion resistance than hard chrome
  - Max. temp – 1100 F

- **Nickel Boron (NiBo)** Balls
  - Coating process – thermal spray and fuse
  - Hardness – 55-60 HRC
  - Not suitable for corrosive liquids
  - Used in high temp and abrasive applications
  - Better than hard facing in erosive or abrasive media
  - Max. temp – 1100 F
- **Chrome Carbide (CrC)** Balls, seats
  - Coating process – High Velocity Oxy Fuel (HVOF)
  - Hardness – 60-65 HRC
  - Excellent wear and and corrosion resistance
  - Max. temp – 1470 F
- **Tungsten Carbide (WC-Co)** Balls, seats
  - Coating process – High Velocity Oxy Fuel (HVOF)
  - Hardness – 65-70 HRC
  - Excellent wear and and corrosion resistance, especially in high cycle applications
  - Max. temp – 840 F.

The most common coating for balls in metal seated ball valves is hard chrome. Its hardness makes 316SS balls much more wear resistant than bare 316. It does not hold up well with chlorides or very strong acids and is not good for very erosive or abrasive media.

The most common coating for metal seated ball valve seats is Stellite (or a similar material). These hard facing materials are very resistant to abrasive and erosion wear and provide sufficient differential hardness to resist galling when used in conjunction with hard chrome plated balls. Balls can also be hard faced, but the machining can be expensive.

Nickel Boron coating on balls (usually used in conjunction with hard faced seats) is used at high temperatures and for very erosive media (but not with corrosive liquids)

Chrome carbide has excellent wear and corrosion resistance and can be applied at the highest temperatures of the common coatings.

Tungsten Carbide gives good service in high cycle applications..

Typical Trim Materials (Ball Valves)

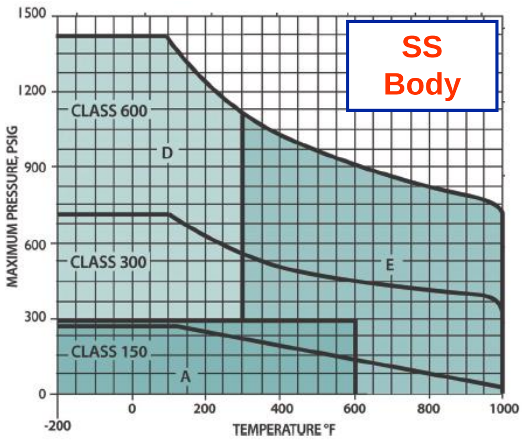
- **Shafts**
  - 316
  - 17-4PH
  - XM-19 (Nitronic 50, Carlson Alloy)..

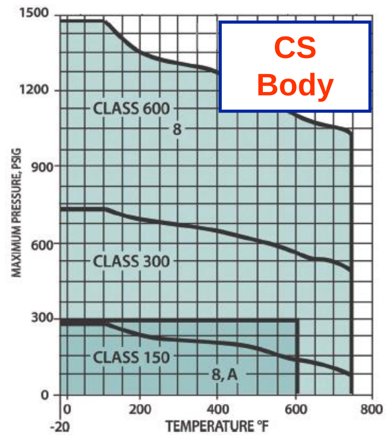
316 SS shafts have good corrosion resistance and moderate strength, and are suitable for many soft seated ball and butterfly applications

17-4PH is a precipitation hardening stainless steel that is very strong. Before the precipitation hardening heat treatment it is easily machineable. After it has been machined, a single low temperature (900F for 1 hour) heat treatment can be applied to increase the strength and hardness of the steel. This is known as “age-hardening.” 17-4 PH is much stronger than 316SS and is used in applications where torque requirements are too high for 316SS. It is not as corrosion resistant as 316SS. Not used above 800F

Nitronic 50 is a trade name of Carlson Alloy for XM-19 Stainless steel. Similar corrosion resistance to 316 and 317 with twice the yield strength. Not as strong as 17-4PH, but retains its strength well above 800F

Globe Valve Trim Considerations





Trim Code	Plug	Seat Ring	Cage	Stem
A	316 SS	316 SS	316 SS/CP	316 SS
D	316 SS/HFS	316 SS/HFS	316 SS/CP	316 SS
E	316 SS/HFS+G	316 SS/HFS	316 SS/CP	316 SS

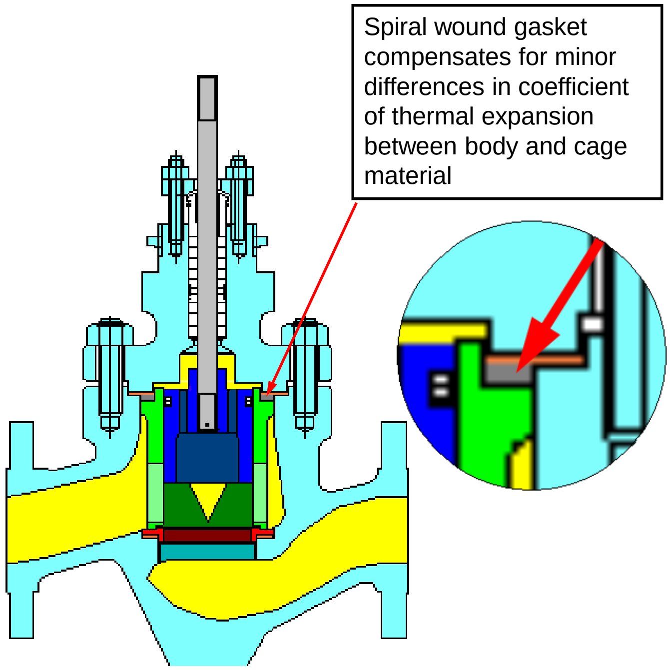
Trim Code	Plug	Seat Ring	Cage	Stem
8	416 SS	416 SS	17-4PH/CP	316 SS
A	316 SS	316 SS	316 SS/CP	316 SS

This is an example of how we use trim materials in globe valves that have compatible coefficients of thermal expansion with the various body materials.

When 316 stainless steel is used for a globe valve body and bonnet, the cage, which is trapped between the body and the bonnet (this is illustrated on the next page) must also be 316 stainless steel so that the body and cage will have the same coefficient of thermal expansion.

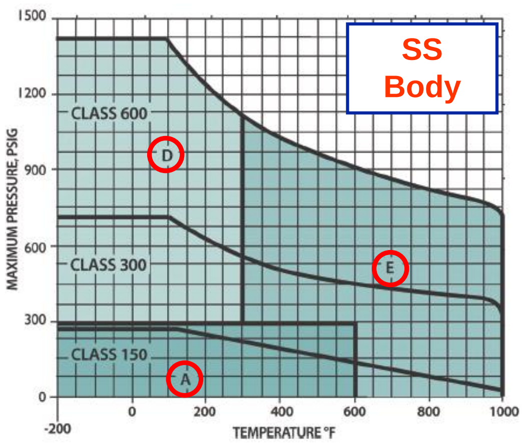
Globe Valve Trim Considerations

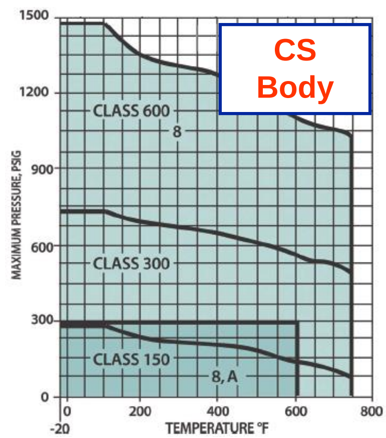
- Cage is trapped between the body and bonnet
- Cage and body must have similar thermal expansion coefficients
- Plug and cage must have similar thermal expansion coefficients



The figure illustrates how the cage in a cage guided valve is trapped between the body and bonnet.

Globe Valve Trim Considerations





Trim Code	Plug	Seat Ring	Cage	Stem
A	316 SS	316 SS	316 SS/CP	316 SS
D	316 SS/HFS	316 SS/HFS	316 SS/CP	316 SS
E	316 SS/HFS+G	316 SS/HFS	316 SS/CP	316 SS

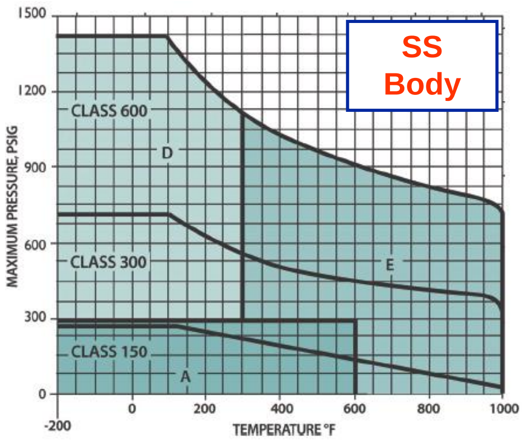
Trim Code	Plug	Seat Ring	Cage	Stem
8	416 SS	416 SS	17-4PH/CP	316 SS
A	316 SS	316 SS	316 SS/CP	316 SS

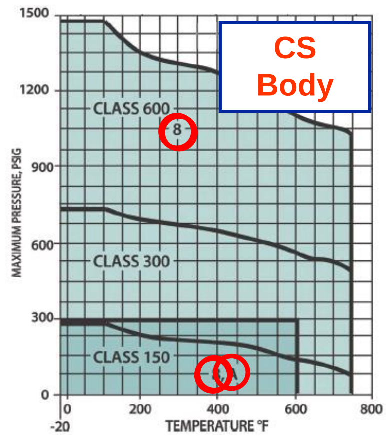
This manufacturer offers three standard trim configurations for their 316 stainless steel valve, all of which have a 316 SS cage. The plug and seat ring must also have thermal expansion properties that are similar to the cage to maintain proper clearances. Because the plug and cage are both 316 stainless steel, the inside of the cage is chrome plated to avoid galling.

For higher pressures the seating surfaces of the plug and seat have a Stellite overlay to give a very hard seating surface (HFS means “Hard Faced Seat.”)

For high temperatures, the guiding surface of the plug is also hard faced because 316 ss becomes softer at high temperatures. (HGS+G means “Hard Faced Seat and Guide.”)

Globe Valve Trim Considerations





Trim Code	Plug	Seat Ring	Cage	Stem
A	316 SS	316 SS	316 SS/CP	316 SS
D	316 SS/HFS	316 SS/HFS	316 SS/CP	316 SS
E	316 SS/HFS+G	316 SS/HFS	316 SS/CP	316 SS

Trim Code	Plug	Seat Ring	Cage	Stem
8	416 SS	416 SS	17-4PH/CP	316 SS
A	316 SS	316 SS	316 SS/CP	316 SS

The standard trim offering for this manufacturer's carbon steel body globe valve uses a cage made of 17-4 PH stainless steel which has thermal expansion characteristics similar to carbon steel. The plug and seat are 416 stainless steel, which is very hard and has similar thermal expansion characteristics to carbon steel.

At lower temperatures (below 600F) differential expansion between the body and cage are not so important, so as an option, this manufacturer offers a 316 SS cage, plug and seat for improved corrosion resistance.

If corrosion is not a problem, the 17-4 / 416 combination is preferred, since there is less difference between the coefficients of thermal expansion and the seating and guiding surfaces are much harder.

Corrosion & Chemical attack

- Corrosion
  - a destructive chemical process; most often applied to the conversion of a metal to one of its compounds, for example, the corrosion of iron by oxygen and water to produce iron oxides (rust)
  - All metals corrode; in our normal atmosphere of 21% oxygen, all metals except gold, platinum, and palladium corrode spontaneously
- Chemical attack
  - The chemical or electrochemical reaction between a material, usually a metal, and its environment that produces a deterioration of the material and its properties.

The next two pages are a corrosion guide that was originally published by a globe valve manufacturer a number of years ago. It is this author's observation that the valve manufacturer that published them is knowledgeable of control valve subjects.

However, this author has not personally verified the correctness of the corrosion guide. It is presented here with no warranty as to its correctness or applicability.

It is not intended as engineering recommendations or advice.

It is the end user's responsibility to make the final decision regarding the selection of materials based on their knowledge of their process.

	BODY MATERIAL			TRIM MATERIAL										
	CAST IRON	CAST STEEL	316 ss	17-4PH	420 or 440 ss	416 ss	316 ss	ALLOY 20	HASTELLOY C	MONEL	NICKEL	STELLITE	HASTELLOY B	TITANIUM
Acetic Acid 100% 24°C(75°F)	X	X	A	A	X	X	A	A	A	B	B	A	A	A
<50% Boil	X	X	A	A	X	X	A	A	A	X	X	A	A	A
>50% Boil	X	X	B	X	X	X	C	A	A	X	X	A	B	A
Acetic Anhydride Boil	X	X	B	B	X	X	B	A	A	B	B	A	A	A
Acetone	B	B	A	A	C	C	A	A	A	A	A		A	A
Acetylene	A	A	A	A	A	A	A	A	A	A	A		A	A
Alcohols	B	B	A	A	A	A	A	A	A	A	A		A	A
Alum Solution (K)	C	C	A	C	C	B	A	B	B	B	B	A	B	A
Alumnium Acetate	X	X	B	B	B	B	B	B	B	B	B		B	A
Aluminum Chloride	X	X	B	C	X	X	B	A	A	B	C		A	
Aluminum Hydroxide	B	B	A	B	B	B	B	B	B	B	B		B	A
Aluminum Sulphate < 10% Boil	X	X	A	A	X	X	A	A	A	C	C		A	A
> 10% Boil	X	X	A	B	X	X	A	A	A	C	C		A	A
Amines	A	A	A	A			A			B	B			
Ammonia (Anhydrous)	A	B	A	A			A	A	A	A	A		A	
Ammonium Bicarbonate	C	C	A				B			X	X			
Ammonium Carbonate	B	B	B		B	B	B	B	B			A	B	
Ammonium Chloride < 10% Boil	X	X		C				A	A	B	B			A
> 10% Boil	X	X							A	B	B			A
Ammonium Hydroxide		B	A	A	A	A	A	A						A
Ammonium Nitrate	X	A	A	A	B	B	A	A	B	X	X			
Ammonium Oxalate	X	X	B		C	C	B		B	X	X			
Ammonium Persulphate 5%	X	X	B	B	C	C	B	B	B	B	B		B	
Ammonium Phosphate 5%	X	X	B				B		B	B	B			
Ammonium Sulphate < 10%	X	X	B	B	C	C	B	B	B	B	B			
Ammonium Sulphite Boil	X	X	B		X	X	B	B	B	X	X			
Amyl Acetate	B	B	A	A	A	A	A	A	A	A	A			A
Amyl Chloride	C	B	A				A	A		B	B			A
Aniline	A	A	A				A	A	B	B	B			A
Aniline Hydrochloride							B	B		B	X		B	A
Animal Fat & Oil			A				A	A	A		A			
Antimony Trichloride	X	X	X		X	X	X	X	B	B	B		B	X
Arsenic Acid	X	X	A		X	X	B	B	B	X	X			
Barium Carbonate	C	C	B				C		B	B	B		B	
Barium Chloride < 5%	C	C	B	B	X	X		B	A	B	B		A	A
> 5%	X	X	B	B	X	X		B	B	B	B		A	A
Barium Hydroxide	C	C	A	A	C	C	A	A	B	B	A		B	
Barium Nitrate	C	C	A				A	A	B	C	C		B	
Barium Sulphate	C	C	B				B	B	B	B	B			A
Barium Sulphide	X	C					B	B	X	X	B		X	A
Beer	X	X	A	A	C	C	A	A	A	A	A			
Benzoic Acid	B	B	A	A	A	A	A	A		A	A			
Benzene (Benzol)	B	B	A	B	X	X	B	A	B	A	A		B	A
Black Liquor (Sulfate)	C	C	A	A	A	A	A	A	A	A	A			
Blood			A	A			A	A		A	A			
Boric Acid	X	X	A	A	B	B	A	B	A	B	B		A	A
Brines (Calcium)	C	C						C	A	A	A	A	A	
Brines (Sodium)	C	C						C	B	A	A	A	B	
Bromine Water			B				C	B					B	
Butyric Acid Dil.	X	X	B				B	B	A				A	A
Conc.	X	X	B				B	B	A				A	A
Butyl Acetate	X	X	B				B	B		B	B			
Calcium Bisulfate	C	C	B				B	B	B				B	A
Calcium Carbonate	B	A	A				A	B	B	B	B			
Calcium Chloride < 20%	B	B	A	B	C	C	B	B	A	A	A		A	A
Calcium Hydroxide Boil 10%	B	B	A				B		A	B	B			
20% Boil	B	B	B				B		A	C	C			
30% Boil	C	C	B				C		A	C	C			
Calcium Hypochlorite < 5%	X	X	A				B		A	X	X	A		A
Calcium Sulfate	C	C	B	B	X	X	B		B	B	B		B	A
Carbolic Acid Phenol 90%	C	C	A	B	X	X	B	A	A	A	A		A	A
Carbonated Beverage			A					A	A		A			
Carbonic Acid > 90%	C	B		B	X	X	B		A				A	
Carbon Bisulfide	B	B	A		X	X	B	B		B	B			A
Carbon Dioxide	A	A	A	A	A	A	A	A	A	A	A		A	A
Carbon Disulfide	A	A	A				A	A						
Carbon Monoxide	A	A	A	A	A	A	A	A	A	A	A		A	A
Carbon Tetrachloride Dry			A	A	A		A			A	A			A

	BODY MATERIAL			TRIM MATERIAL										
	CAST IRON	CAST STEEL	316 ss	17-4PH	420 or 440 ss	416 ss	316 ss	ALLOY 20	HASTELLOY C	MONEL	NICKEL	STELLITE	HASTELLOY B	TITANIUM
Carbon Tetrachloride Wet			B				C			B	B			B
Cellulose Acetate	C	C	B		X	X	B	B	B	A	A		B	A
Chloroacetic Acid	X	X	X		X	X	X	X	B	C				A
Chlorinated Water	X	X	X		X	X	X	B	A					A
Chlorine Gas Dry	X	B	B	C	X	X	C	B	A	B	X			
Wet	X	X	X	X	X	X	X	X	C	X	X			
Liquid														
Chlorobenzene	B	B	B				B	B	B	B	B		B	A
Chloroform Dry 100% 24°C(75°F)	A	A		B	X	X	B	B		A	A		B	A
Chromic Acid < 10% Boil	X	X	C	C	X	X	C	B	B					A
Chromic Acid > 10% Boil	X	X	X	X	X	X	X	B	B					A
Citric Acid Dil. 52°C(125°F)	X	X	A	B	X	X	A	A	A	B	B			A
Citric Acid < 50% 52°C(125°F)				X	X	X	A	A	A	B	B		A	A
Copper Acetate < 20%			B		X	X	B	A	B	A	A			
Copper Chloride														A
Copper Nitrate Conc. 93°C(200°F)	X	X	B		X	X	B	B	X	X	X		X	
Copper Sulfate < 40% 93°C(200°F)	X	X	B	B	X	X	B	A	A	X	X		A	A
Creosote 100°C(212°F)	B	A	A	B	X	X	B	B		B	B			
Cresylic Acid		A					A	B	B				B	
Cupric Chloride < 5% 24°C(75°F)									B				B	B
Cyanide Solution (Plating)	B	B	A				B	B	B	B	B		B	B
Cyanogen Gas 100% 24° (75°F)			B				B	B						
Dichloroethane 100% 100° (212°F)	B	B	A				B	B	B	B	B		B	A
Ether	B	B	A	B	B	B	B	A	B	B	B		B	A
Ethyl Acetate	C	C	A	A	X	X	A	A	B	B	C		B	A
Ethyl Chloride Dry	C	A	A	A	B	B	A	A	B	B	X		B	A
Ethylene Glycol	A	B	A	A	B	B	A	A	B	B				A
Fatty Acids 100°C(212°F)	X		A		B	B	A	A	A	B	A		A	A
Ferric Chloride < 1% 24°C(75°F)	X	X	X		X	X	X	X	A	X	X		A	A
Ferric Chloride > 1%	X	X	X		X	X	X	X	A	X	X		A	A
Ferric Chloride Hot < 1%	X	X	X		X	X	X	X	X	X	X			A
Ferric Chloride Hot > 1%	X	X	X		X	X	X	X	X	X	X			A
Ferric Hydroxide			B				B	B						
Ferric Nitrate 5%	X	A	A	B			B	A	B					
Ferric Sulphate 5%	X	X	A	B			A	A	A		B			A
Ferrous Chloride 10%	X	X					X		B				B	A
Ferrous Sulphate 10%	X	X	A	B	C	C	B	A	A	X	X		A	A
Fluorine Gas Dry	X	B	A		B	B	A	A		A	A			A
Fluorine Gas Wet										B	B			
Formaldehyde 40%	X	B	A	A	B	B	A	A	B	A	A		B	
Formic Acid < 50%	X	X	X	B	X	X	X	A	B	A	A		B	
Formic Acid > 50%	X	X	X	B	X	X	X	A	B	A	B		B	
Formic Acid < 50% 93°C(200°F)	X	X	X	X	X	X	X	A	B	B	C		B	
Formic Acid > 50% 93°C(200°F)	X	X	X	X	X	X	X	B	C	C	C		C	
Foods			A					A	A					
Freon Wet	B	B	A	B	B	B	A	A	A	A	A		A	
Furfural	B	B	A	B	B	B	A	A		B				
Gallic Acid			A		B	B	B	B		C	C			
Glucose														
Glycerine														
Green Liquor (NaOH)	C	C	A	A	A	A	A	A		A	A			
Hydrochromic Acid														
Hydrocarbons Crude & Refined	A	A	A	A	A	A	A	A	A	A	A			
Hydrochloric Acid < 1%	X	X	X							B	B		A	B
Hydrochloric Acid 1-20%	X	X	X							C	C		B	B
Hydrochloric Acid > 20%	X	X	X										B	C
Hydrochloric Acid < 1/2% 80°C(175°F)	X	X	X										B	C
Hydrochloric Acid > 2% 80°C(175°F)	X	X	X										B	C
Hydrocyanic Acid < 40%	X	X	A				B	B						
Hydrocyanic Acid > 40%	X	X	A				B	B						
Hydrogen Chloride Gas Dry	X	X								B	B			
Hydrogen Chloride Gas Wet	X	X								X	X			
Hydrogen Fluoride Dry 100%		A	A				A	A	B	B	A			
Hydrogen Sulfide Dry			A	B	C	C	A	A		B	B			A
Hydrogen Sulfide Wet			A				B	B		C	C			A

KEY TO RATING SYMBOLS: A RECOMMENDED B FAIR C PROBABLY UNSUITABLE X UNSATISFACTORY

	BODY MATERIAL			TRIM MATERIAL										
	CAST IRON	CAST STEEL	316 ss	17-4PH	420 or 440 ss	416 ss	316 ss	ALLOY 20	HASTELLOY C	MONEL	NICKEL	STELLITE	HASTELLOY B	TITANIUM
Iodine Dry 100%			A	B			B		A	A	A			
Lactic Acid 5%	X	X	A	B	X	X	A	A	B					A
10%	X	X	A	X	X	X	A	A	B					A
5% Boil	X	X	B	X	X	X	B	B	C					A
10% Boil	X	X	B	X	X	X	B	B	C					A
Lead Acetate			A				B	B	C	C	C			A
Magnesium Chloride 5% Hot		B		C			B	A	A	B	A		A	A
Magnesium Hydroxide	B	B		A			A	A	B	B	A		B	
Magnesium Sulfate	C	B	A	A	A	A	A	A	A	A	A		A	A
Mercury	A	A	A		A	A	A	A	B	B	B		B	
Mercuric Chloride < 2%									C					A
> 2% Boil									X					A
Mercuric Cyanide	X	X	X		X	X	X		C	X	X			
Methyl Chloride Dry			A				A	A		B	B			
Mine Water (Acid)			A	A	A	A	A	A	A	A	A			
Molybdic Acid < 5%			A				B	B						
Natural Gas (Liquid)			A				A	A	A					
Nickel Chloride			B				B						A	
Nickel Sulfate			A				B	A						
Nitric Acid 20%			A	A			A	A					A	A
20% Boil			B	X			B	A					A	A
65%			C	X			C	B					B	A
Conc. Boil			X	X			X	X					X	A
Fuming			X	X			X	X					X	A
Nitrobenzene 100%		A	B				B	B	A	B	B		A	A
Nitrous Acid	X	X	B				C	C						
Oleum See Sulfuric Fuming														
Oxalic Acid 10%	X	X	A	B	C	C	B	A	B	B			B	
10% Boil	X	X	B	X	C	C	C	A	B	C			B	
50% Boil	X	X		X	X	X	X	B	C	X			C	
Oxygen	A	A	A	A	A	A	A	A	A	A	A		A	A
Perchloroethylene	B	B	B				B	B		A	A			A
Phenol	B	A	A				A	A	A	A	A		A	
Phosphoric Acid 10%	X	X	A	B			A	A					A	
10-50%	X	X	A	B			A	A					A	
> 50%	X	X	B	C			B	B					A	
> 20% 175°F	X	X	B	X			B	B					A	
< 10% Boil	X	X	B	X			C	C					A	
85% Boil	X	X	C	X			X	X					B	
Phosphorous Trichloride Dry 100%	B	A	A				A				A		A	A
Phthalic Anhydride 100%	A	A	A		B	B	A	A	A	A	A		A	
Picric Acid < 38°C(100°F)	X	X	A	C			B	B						
Potassium Bromide < 40%	X	X	A	C			B	B	A	B	B		A	A
Potassium Carbonate	B	B	A	A	B	B	A	B	B	B	B		B	A
Potassium Chlorate < 40% < 93°C(200°F)	X	X	A	B	B	B	A	B	B	X	X			A
Potassium Chloride < 40% < 93°C(200°F)	X	X	A	B	C	C	A	A	B	B	B		B	A
Potassium Cyanide	X	X	B				B	B			B			
Potassium Ferricyanide < 20%	X	X	B	B	C	C	B	B	B	B	B		B	
Potassium Hydroxide 50%	B	B	A	B	C	C	B	B	B	A	A			
30% 80°C(175°F)	B	C	A	B	C	C	B	B	B	A	A			
50% 80°C(175°F)	X	X	B	C	X	X	B	B	B	A	A			
30% Boil	X	X	X	X	X	X	X	X		A	A			
50% Boil	X	X	X	X	X	X	X	X		B	A			
Potassium Hypochlorate Dil.	X	X	C						B				B	A
Potassium Iodide	X	X	A	B			B	B	B	B	B		B	A
Potassium Nitrate	A	A	A	B	B	B	B	B	B	B	B		B	
Potassium Permanganate Dil.	X	X	A	B	B	B	B	B	B	B	B			
Potassium Sulfate Dil.	X	X	A	A	B	B	A	A	A	A	A		A	A
Potassium Sulfate Dil. Boil	X	X	A				B	A	B	B	B		B	
Potassium Sulfide Sat.	B								B					A
Propane (Liq. Gas) 100%	A	A	A	A	A	A	A	A	A	A	A		A	A
Propionic Acid	X	X	A	X	X	X	B		A	B				

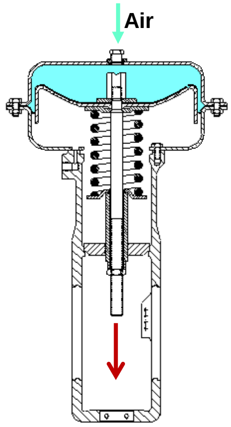
	BODY MATERIAL			TRIM MATERIAL										
	CAST IRON	CAST STEEL	316 ss	17-4PH	420 or 440 ss	416 ss	316 ss	ALLOY 20	HASTELLOY C	MONEL	NICKEL	STELLITE	HASTELLOY B	TITANIUM
Pyrogallic Acid < 100%	X	X	A	B			B	A	A				A	
Pyroligneous Acid < 20%	X	X	A				A	A		B				
Quinine Bisulfate			B				B	B	B				B	
Quinine Sulfate	X	X	A	B	B	B	B	B	B	B	B		B	
Rosin (Molten)	X	X	A	B	C	C	B			A	A			
Salicylic Acid	X	X	A				B	B						
Sewage	A	A	A	A	A	A	A	A	A					
Silver Bromide < 20%				C				B	A				A	
Silver Chloride								C	C				C	
Sodium Cyanide	B	B	A	B			B	B						
Sodium Fluoride 5%			B	B			B	B						
Sodium Hydroxide 50%	B	B	A	A	B	B	A	A	A	A	A		A	A
< 40% 80°C(175°F)	B	B	A	A	B	B	A	A	A	A	A		A	A
40-75% 80°C(175°F)	C	C	A	B	B	B	A	B	B	A	A		B	B
< 30% Boil			B	B	C	C	B	C	C	B	A		C	
> 30% Boil			B	C	X	X	C	X	X	B	A		X	
Molten										C	B			
Sodium Hypochlorite < 10% < 24°C(75°F)	X	X	C	X			C		A				A	
Sodium Hyposulfite	X	X	B				C		B				B	
Sodium Nitrate	B	B	A	B	C	C	A	A	B	B	B		B	
Sodium Perborate < 20%	B	B	B	B			B	B	B	B	B		B	
Sodium Peroxide	B	B	B	B			B	B	B	B	B		B	
Sodium Phosphate	A	A	A	B	B	B	B	B	B	B	B		B	A
Sodium Salicylate < 20% 24°C(75°F)			A				B	B						
Sodium Stannate < 50% 93°C(200°F)	B	B	A	B	C	C	B	B	B	B	B		B	A
Sodium Sulfate < 40% 100°C(212°F)	C	C	A	B			A	A	B				B	
Sodium Sulfide < 93°C(200°F)			A	A			A	A	B				B	
Sodium Sulfite < 20% < 80°C(175°F)	X	C	A	A			A	A	B	B	B			
Sodium Thiosulfate < 20% < 80°C(175°F)			A	B			B	B						
Stannic Chloride < 5%	X	X	B				C	B						
> 5%	X	X	X				X	C						
Stannous Chloride < 20% < 80°C(175°F)	X	X	A				A	A					B	
Steam 100°C(212°F)	A	A	A	A	A	A	A	A				A		
320°C(600°F)		A	A				A	A				A		
Sulfate Pulp Liquor	C	C	A	B			B	A						
Sulfite Pulp Liquor	C	A				B	A							
Sulfur (Molten)			A				A	A	A	B	B		A	
Sulfur Chloride	X	X	C				X	B	B		B			
Sulfur Dioxide Dry	X	X	A	A			A	A	A					
Sulfur Dioxide Wet	X	X	A	C			B	B						
Sulfuric Acid < 2%	X	X	A	B				A	A	A			A	
2-40%	X	X						A	A	A			A	
> 95% Conc.	A	A	A	B			B	A	A	A			A	
10-60% Boil	X	X						B		C			A	
Fuming	X	X												
Sulfurous Acid	X	X	A				B	A	B				B	
Tannic Acid (Sat.)	X	X	A	B			B	B						A
Tar (Hot)	A	A	A	A	A	A	A	A						
Tartaric Acid	X	X	A	A	C	C	A	A	B	B	B		B	A
Tin Chlorides < 10%	X	X						B					B	A
Titanium Sulfate < 10%			B				B	B					B	
Trichloroethylene 24°C(75°F)	A	A	A	A	A	A	A	A	A	A	A			
Trichloroacetic Acid									B				B	A
Turpentine 24°C(75°F)	B	B	A	A	B	B	A	A		B	B			
Water Fresh	B		A	A	B	B	A	A	A	A	A		A	A
Sea			B	C			C	A	A	A	A		B	
Mine	X	X	A	B			B	A						
Distilled	A	B	A	A			A	A		A	A			
White Liquor	C	C	A	A	A	A	A	A		A	A			
Zinc Chloride 5% 24°C(75°F)			A	C			A	A		B	B		B	A
5% Boil				C				A		B	B		B	A
Zinc Sulfate			A				A	A	B	B	B		B	

KEY TO RATING SYMBOLS: A RECOMMENDED B FAIR C PROBABLY UNSUITABLE X UNSATISFACTORY

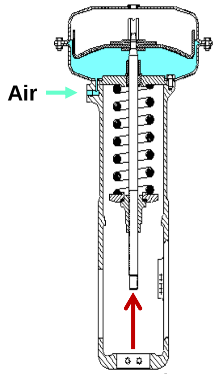
Valve Accessories

In this section on valve accessories we will discuss valve actuators, control valve positioners, limit switches, position transmitters, volume boosters, solenoid valves, quick exhaust valves and lock-up valves

Globe Valve Actuators



- Direct acting
  - Stem extends with increasing air pressure
- Air-to-close
- Fail open



- Reverse acting
  - Stem retracts with increasing air pressure
- Air-to-open
- Fail closed

Nearly all globe control valve actuators are the spring and diaphragm type shown above.

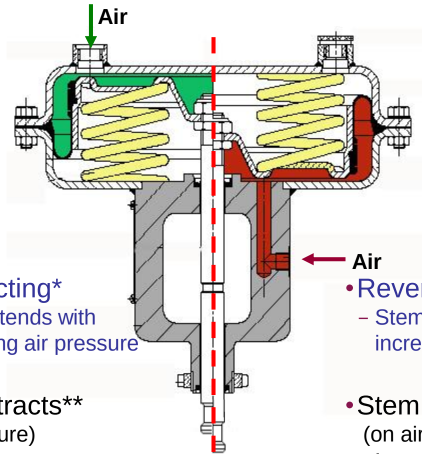
These come in two configurations, **Direct acting** and **Reverse acting.**

Direct acting means that putting air into the actuator causes the actuator stem to extend out of the actuator.

Reverse acting means that putting air into the actuator causes the stem to retract into the actuator.

Since all modern globe valves close when the valve stem is pushed down, that means that the direct acting actuator has an air to close (thus also fail open) action. The reverse acting actuator has an air to open (and thus fail closed) action.

Compact Globe Valve Actuators



- Direct acting*
  - Stem extends with increasing air pressure
- Stem retracts**
  (on air failure)
- Air-to-close
- Fail open

- Reverse acting*
  - Stem retracts with increasing air pressure
- Stem extends**
  (on air failure)
- Air-to-open
- Fail closed

* Conventional terminology
** Samson Terminology

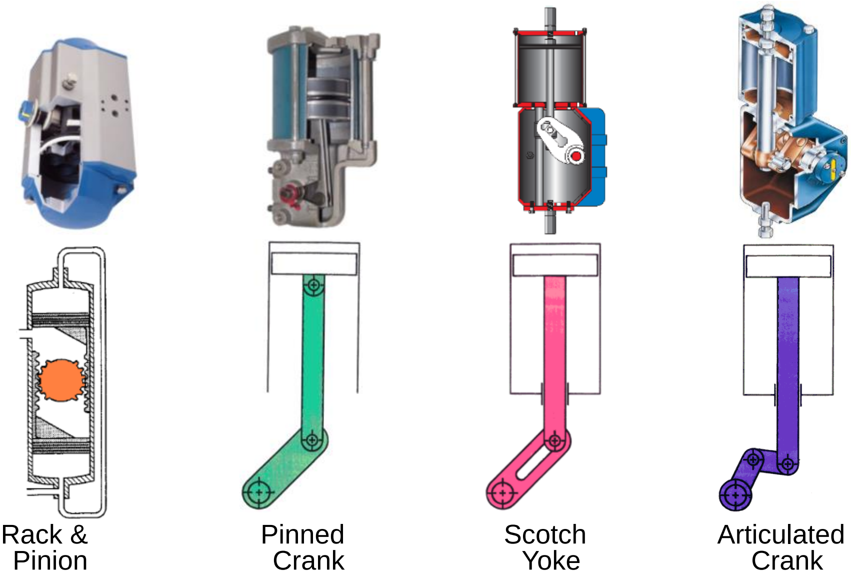
Samson (and some other manufacturers of newer design valves) use actuators like this one, where the springs are inside the diaphragm case, making a much more compact design, and also a design that can be easily changed between “fail open” and “fail closed.”

Something you need to keep in mind when specifying Samson actuators, is that Samson does not use the “direct” and “reverse” acting terminology. Instead, Samson names their actuator action by what the stem does on **LOSS** of air.

What would have been called “direct acting” (stem extends on application of air), Samson calls “**stem retracts**” because that is what happens on loss of air.

Likewise, what would have been called “reverse acting” (stem retracts on application of air) Samson calls “**stem extends**” because that is what happens on loss of air.

Double Acting Piston Rotary Actuators

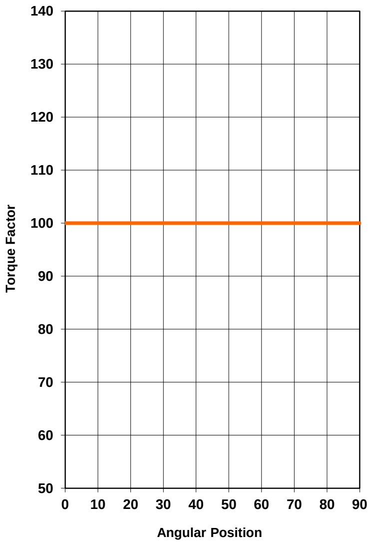


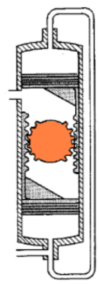
The simplest and least expensive of the rotary actuators is the double acting piston actuator.

There are four common types of mechanisms for converting linear to rotary motion, rack and pinion, pinned crank, scotch yoke and articulated crank.

Starting with the next slide, we will look at the torque output characteristics of each type.

Double Acting Piston Actuator Torque Curves





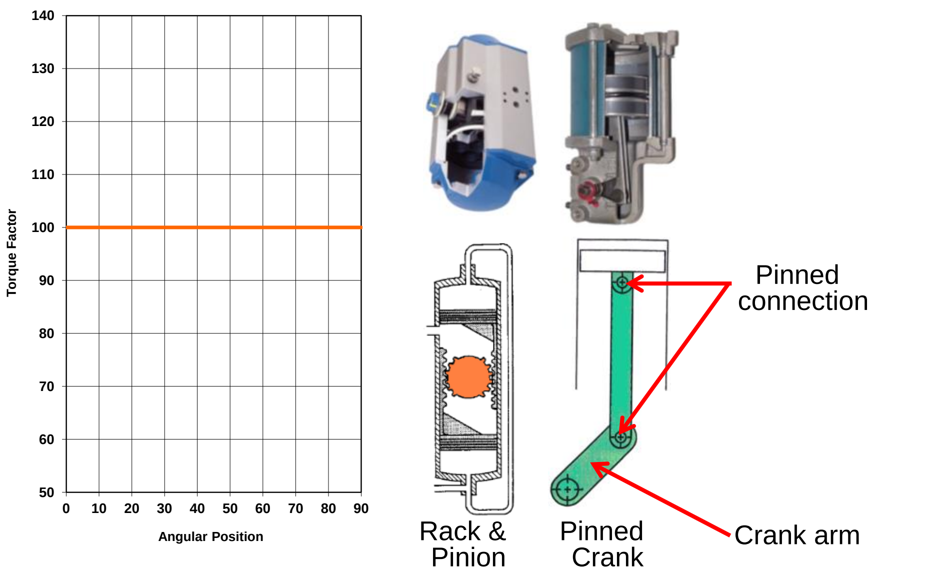
Rack &
Pinion

The torque graphs for all of these actuators is torque factor vs. the actuator’s angular position. A torque factor of 100 is the rated, or nominal, torque of the actuator.

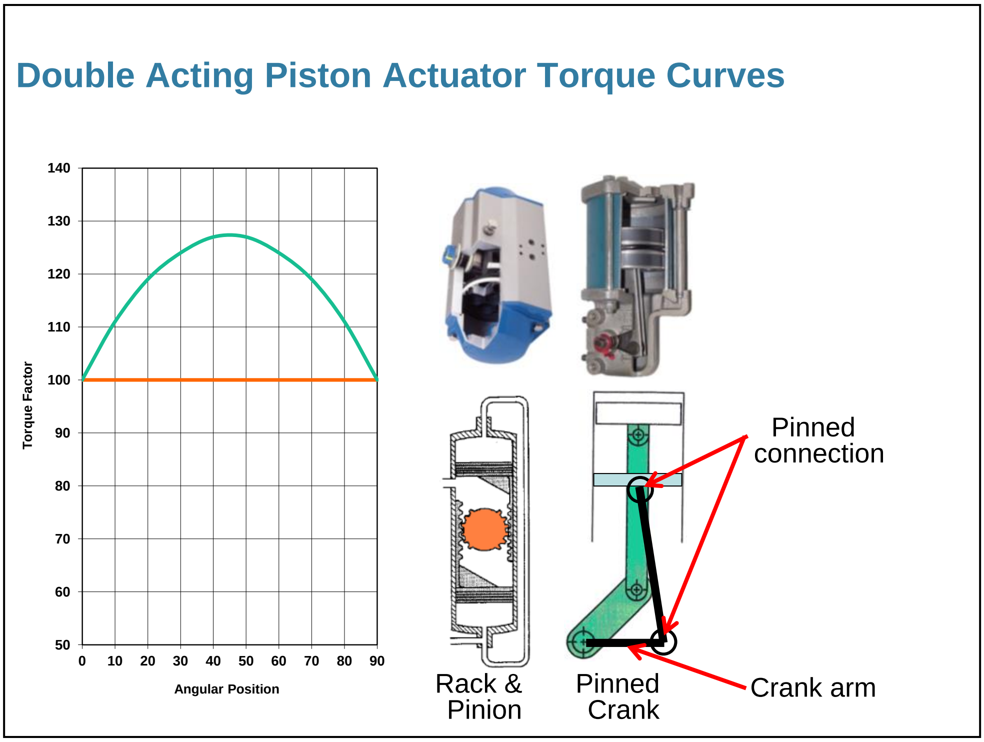
The motion conversion mechanism of the rack and pinion is gear teeth attached to the pistons, turning a gear (pinion). The moment arm distance between the rack and the center of the pinion remains constant at all times, so the torque output is constant at the actuator’s rated torque at all degrees of opening as shown in the orange graph.

The actuator pictured here is the Jamesbury “RP” (which stands for rack and pinion).

Double Acting Piston Actuator Torque Curves



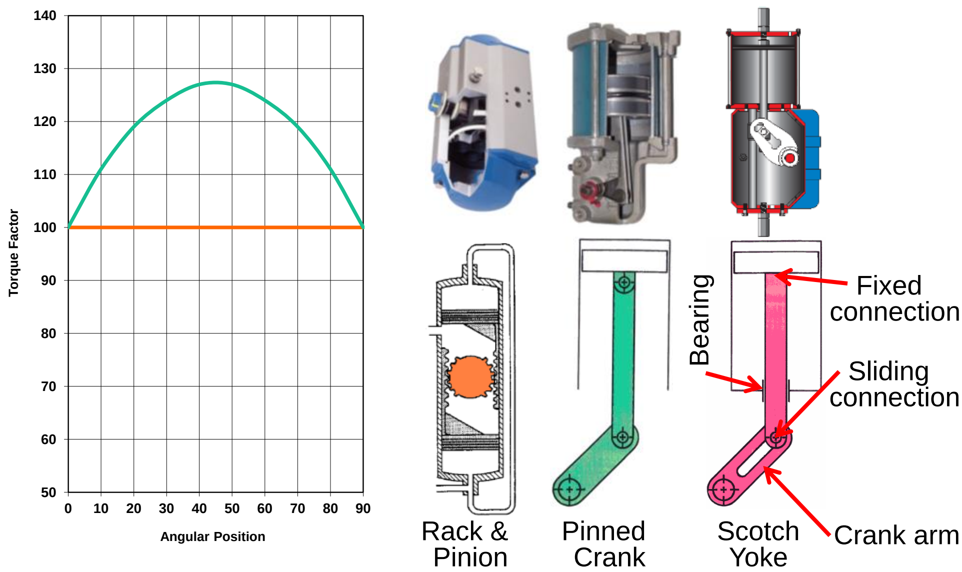
The pinned crank actuator gets its name from the fact that the rod that connects the piston and the crank arm is pinned at both ends and both connections are free to rotate around the pin.



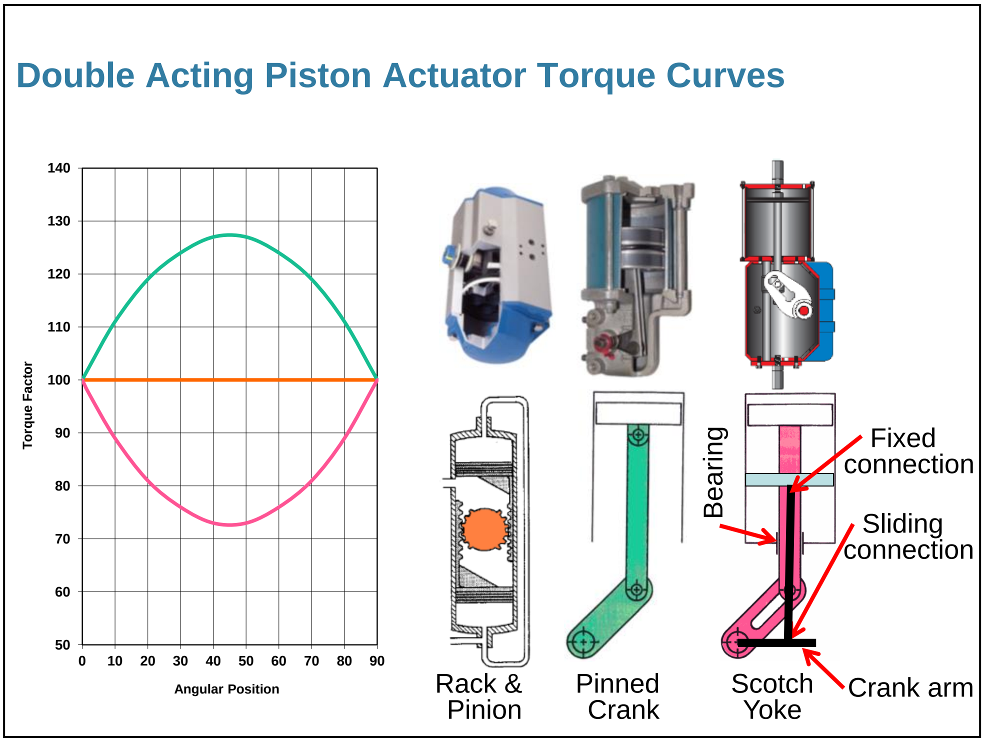
As air pressure moves the piston downward, the rod connecting the piston to the crank arm rotates around the pins at each end. When the piston is at mid position, as shown with the black lines, the moment of the crank arm is longer than it was at the beginning of the stroke, so the torque curve peaks at mid stroke as shown in the cyan curve.

The pinned crank actuator pictured here is the Jamesbury ST actuator which is no longer made. We are discussing the pinned crank design here, because it will help us understand the Jamesbury QuadraPowr spring and diaphragm actuator which is a pinned crank design.

Double Acting Piston Actuator Torque Curves



In the scotch yoke mechanism, the rod that is attached to the piston has a solid fixed connection that cannot rotate. Also there is a bearing at the bottom of the cylinder. As a result the rod can only move up and down in a straight line.



As air pressure moves the piston downward, the rod connecting the piston to the crank arm slides in the slot toward the rotating shaft. At mid position the moment arm is shorter than it was at the beginning of the stroke, so the torque curve starts and ends high and reaches a minimum at mid stroke as shown in the pink curve.

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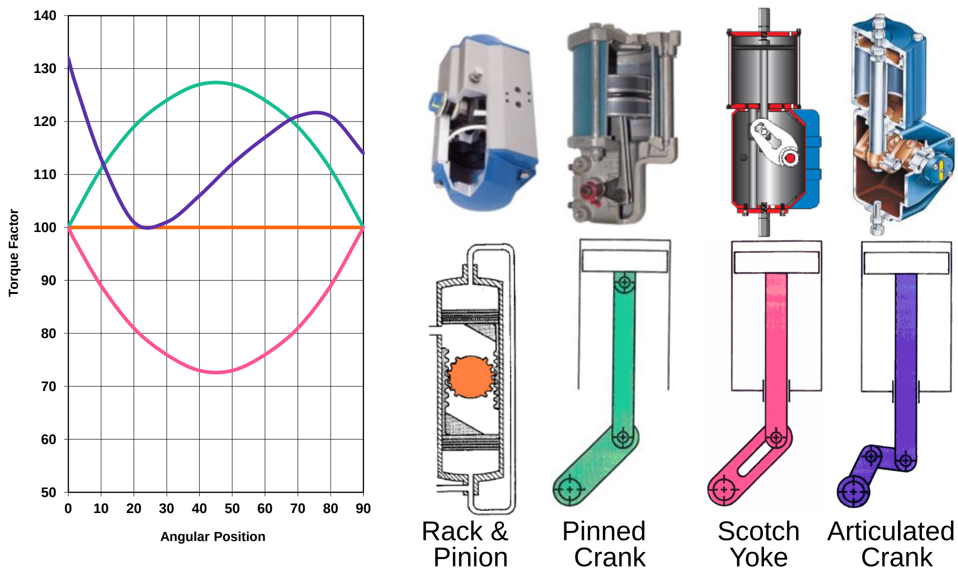
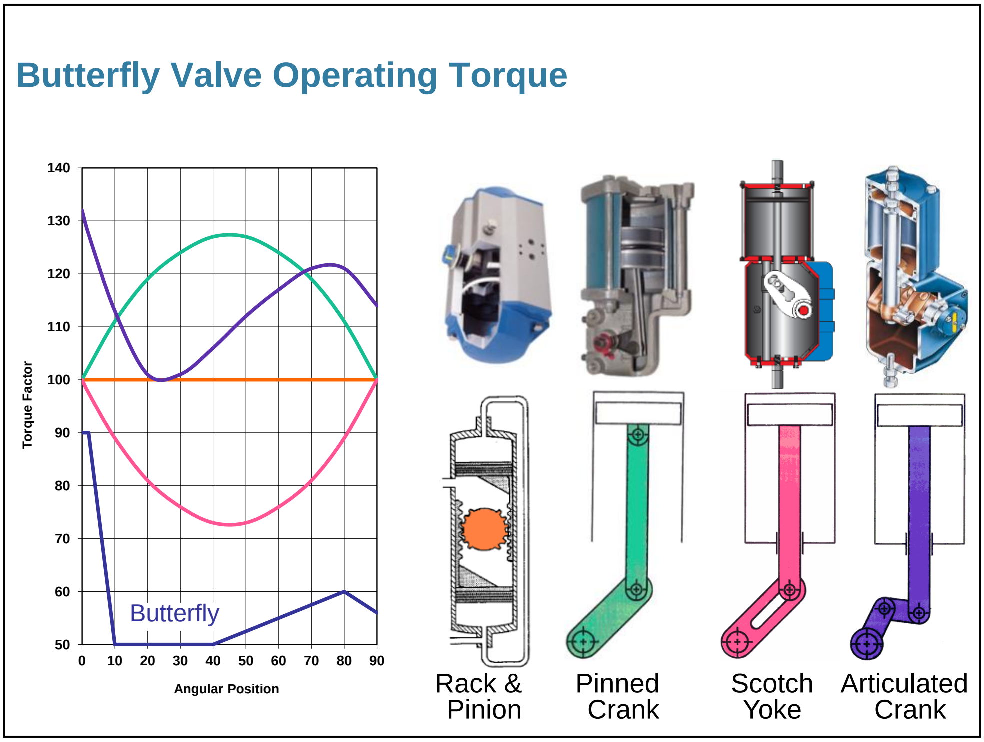
Double Acting Piston Actuator Torque Curves



The geometry of the articulated crank is fairly complex and produces a complex torque output curve as shown in the purple curve.

The picture and diagram are of the Metso BC double acting piston actuator.



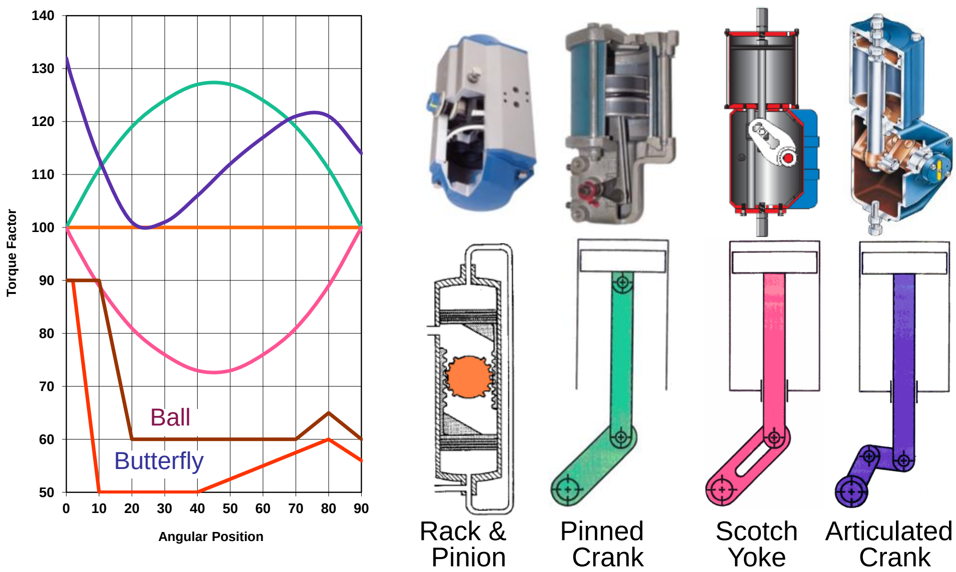
Here we have added a typical torque requirement curve for a butterfly valve. The torque requirement is greatest when the disk is coming out of or going into the seat at zero degrees of rotation. The torque requirement drops significantly when the disk has cleared the seat. Dynamic torque caused by flow interacting with the disk peaks somewhere around 80 degrees.

The maximum torque requirement was arbitrarily drawn at 90% of the rated actuator torque because actuators for on-off service are typically selected to have at least a 10% safety factor. Rated actuator torques and valve torque requirements are usually quite conservative, so only a small safety factor is necessary.

In the throttling range for valves in control service, in order to get very smooth accurate control, it is necessary to have the actuator torque capability be much greater than the valves torque requirement.

To easily compare torque requirements with available actuator torque, we often use the term “Load Factor,” where the load factor is the percentage of the actuator’s torque capability that is required by the valve at any particular degree of rotation. At the point where the valve is going into or out of the seat, Neles recommends that the load factor be less than or equal to 90%, that is the seating torque needs to be less than 90% of the actuator’s capability at that point. In the throttling range, Neles recommends that the load factor in the throttling range be less than 60%, and the lower the better.

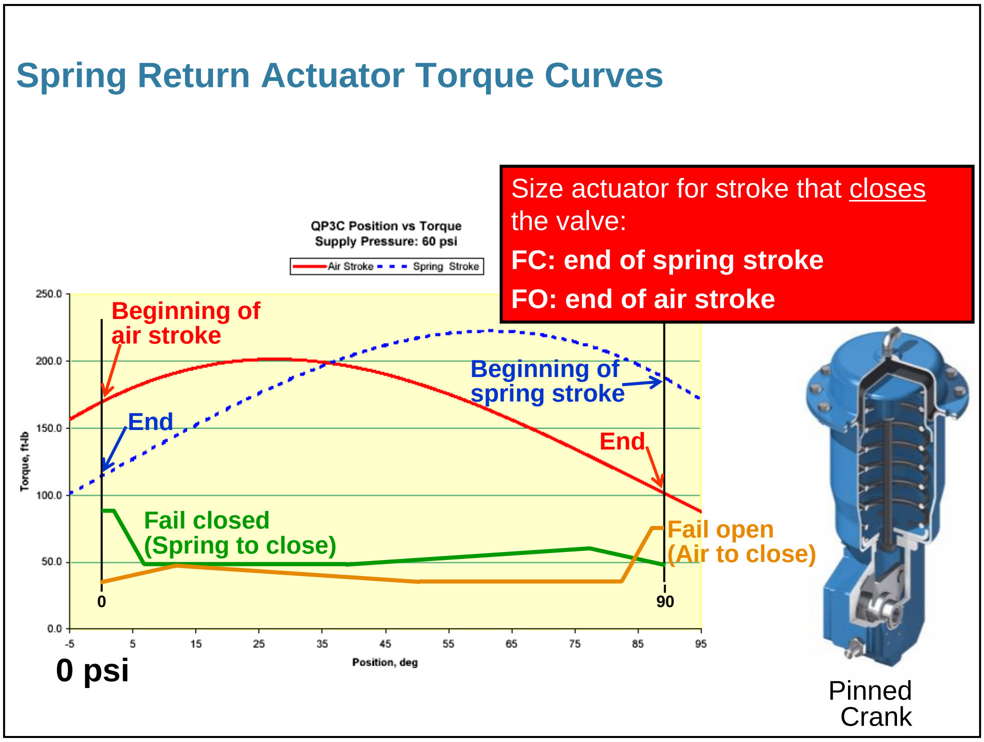
Butterfly and Ball Valve Operating Torque



The ball valve torque requirement is similar in shape to that of the butterfly valve. (As with the butterfly valve curve this one is typical, however actual values will depend on valve construction, shutoff pressure and the pressure drop across the valve when it is throttling.)

When seating and unseating, the ball valve has several degrees of “dead angle” where the ball is turning but the waterway in the ball is fully covered by the seat and there is no flow so the full shutoff pressure is pushing the ball into the seat. Also the ball valve torque does not drop as low as the butterfly does because the ball is always in contact with the seat.

The pinned crank and articulated crank are the preferred construction for control valves because of the extra spare torque in the throttling range.



The Jamesbury spring and diaphragm QuadraPowr rotary actuator uses the pinned crank mechanism, and therefore has a torque curve that peaks mid stroke.

The torque curves of spring return actuators are made more complex by the addition of the spring forces, and the curves are different for the air stroke and the spring stroke. During the air stroke (the red curve on the graph), air is causing the actuator to move away from the 0 degree position and generating torque, but is fighting against the spring whose opposing force increases as the actuator rotates toward the 90 degree position. The air stroke torque starts out high when the spring is lightly compressed, and ends up low when the spring if fully compressed.

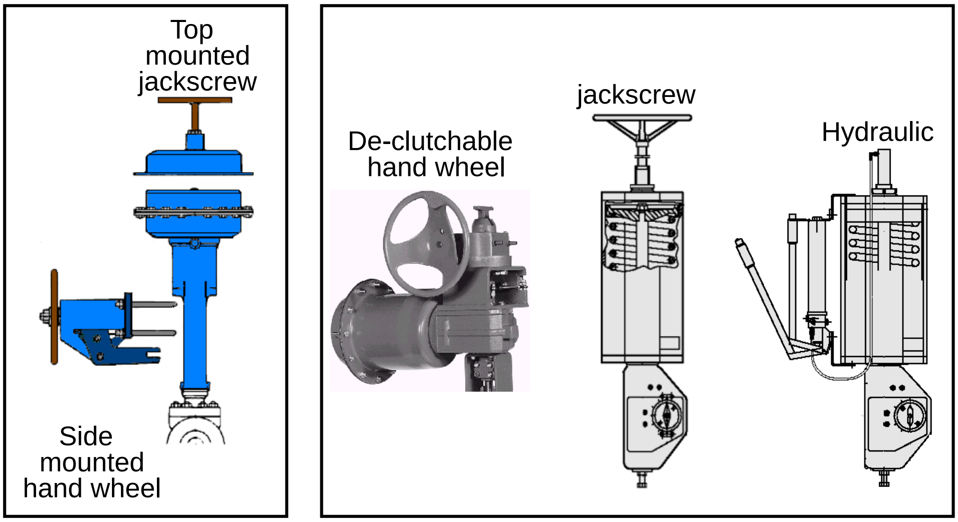
During the spring stroke (the blue curve on the graph), the spring force is causing the actuator to move away from the 90 degree position and is generating torque, but the spring force decreases as the actuator rotates toward the 0 degree position at the end of the spring stroke and the spring unwinds. The spring stroke starts out high and ends up low.

To be sure that an actuator has sufficient torque to get the valve into and out of the seat, size the actuator so that the torque at the end of the stroke that closes the valve is greater than the torque required to get the valve into and out of the seat.

For example, for a fail closed valve, that is, a valve that the spring stroke closes the valve, looking at the green graph you can see that the actuator torque at the end of the spring stroke that closes the valve into the seat is lower than the actuator torque at the beginning of the air stroke that opens the valve out of the seat, so it is the actuator torque at the end of the spring stroke that is critical.

For a fail open valve, that is, a valve that the air stroke closes the valve, looking at the orange graph you can see that the actuator torque at the end of the air stroke that closes the valve into the seat is lower than the actuator torque at the beginning of the spring stroke that opens the valve out of the seat, so it is the actuator torque at the end of the air stroke that is critical.

Manual Overrides



Globe valves

Rotary valves

Manual overrides are sometimes specified for both on/off and modulating actuators.

The jackscrews are the least expensive but require more force and are recommended for infrequent use. They can also be used to limit valve position in the spring direction.

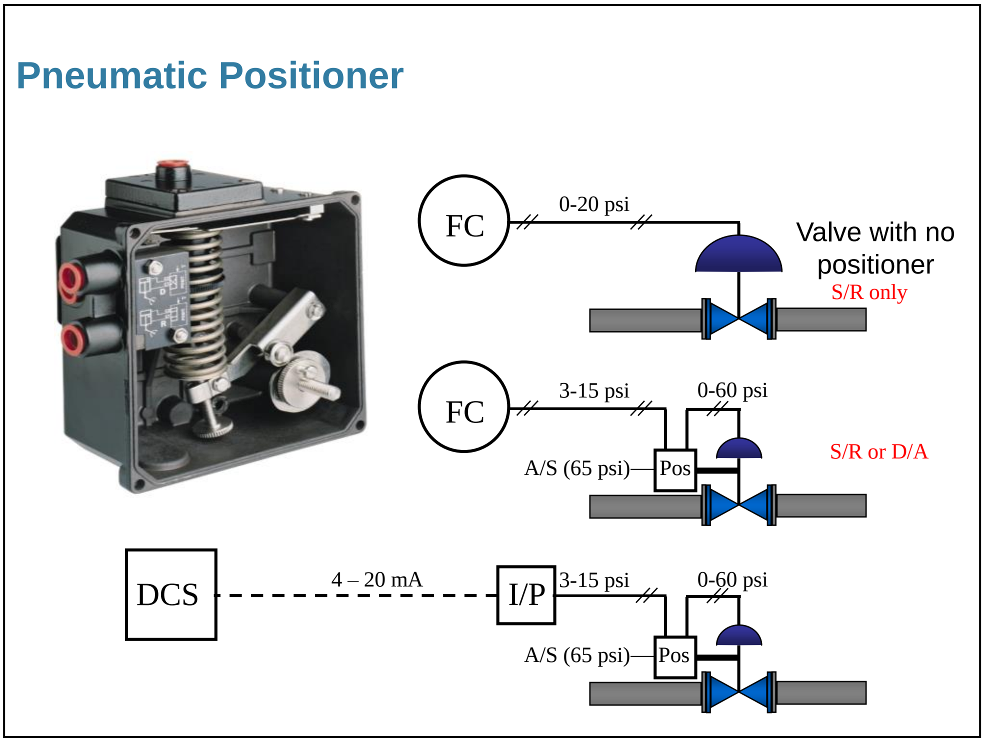
Though not shown, there is a top mounted jackscrew, similar to the one shown for globe valves, available for the spring and diaphragm rotary actuator.

The side mounted globe valve hand wheel, geared de-clutchable and hydraulic types are more expensive and easier to operate and are recommended when frequent manual operation will be required. Note that the de-clutchable hand wheel mounts in the same place where a positioner goes, so it is not applicable to control valves.

Positioners, Limit Switches and Position Transmitters

Next, we will talk about valve positioners, followed by a brief discussion of limit switches and position transmitters.

A control valve positioner is a device that receives a control signal from a process control system, measures the valve's current position, then sends air to the actuator until the valve positions itself at the required position.



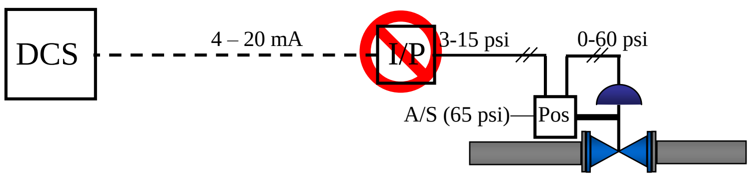
When pneumatic controllers and pneumatic control valves first came into use the controller output was connected directly to the valve's spring diaphragm actuator. (The circle with "FC" inside it is the symbol for a Flow Controller) The line connecting the flow controller to the valve's actuator with little hash marks drawn across it is the symbol for pneumatic tubing carrying a pneumatic control signal of 0 to 20 psi. Since a maximum of only 20 psi air pressure was available, the actuator had to be larger than it would have needed to be if more pressure was available. Because there was only a single output from the controller, the actuator had to be a spring return type.

The addition of a pneumatic positioner made it possible to use higher supply pressures making it possible to significantly reduce the size of the actuator and increase valve response time because the positioner could provide greater air volume. If process dynamic forces try to move the valve, the positioner immediately resists any undesired valve movement, having the effect of making the actuator much stiffer than it would be without the positioner. Without a positioner there is only one source of air pressure, so only spring return actuators can be used. Positioners can be designed so that there are two air outputs, one that tends to open the valve and one that tends to close the valve, making it possible us use double acting actuators which are lighter in weight, smaller and less expensive than spring return actuators. When a pneumatic control signal is used, it is almost always a signal ranging between 3 psi and 15 psi, where 3 psi represents 0% and 15 psi represents 100%.

With the introduction of electronic control systems, the addition of a current-to-pneumatic converter mounted on or near the valve took care of the signal conversion from an electrical control signal to a pneumatic control signal. Even with the introduction of electro-pneumatic positioners the use of pneumatic positioners and I to P converters has persisted in many places. One reason for this is that some electro-pneumatic positioners, especially older ones, are sensitive to vibration. The I/P converter is often mounted on a wall away from the vibration.

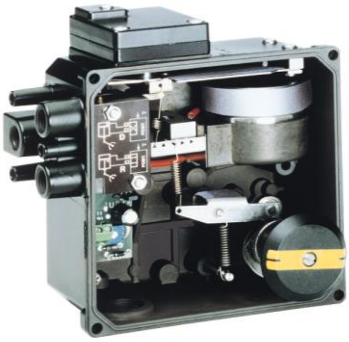
(Generally there is about a 5 psi pressure loss in the positioner. This is why the figures show a 65 psi air supply and 0-60 psi going to the actuator.)

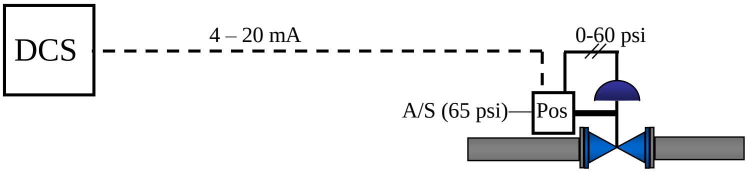
Pneumatic Positioner



The next step in positioner evolution was to eliminate the I/P converter …..(continued on next page)

Electro-Pneumatic Positioner

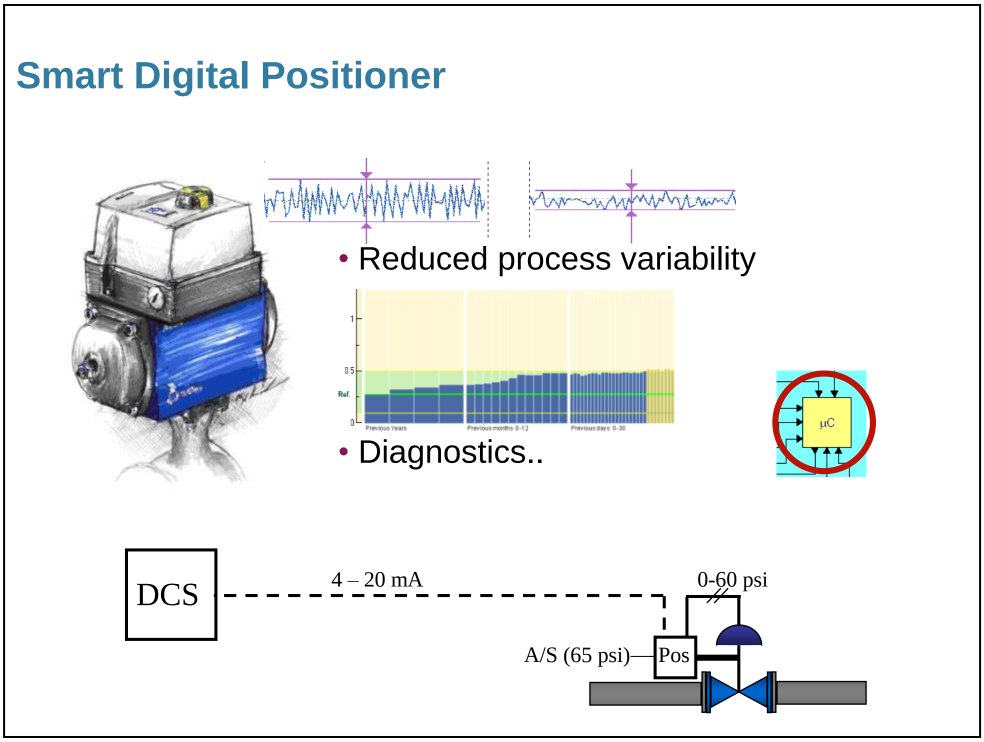




(continued from previous page)

…and replace the pneumatic positioner with an electropneumatic positioner.

The 4-20 mA signal from an electronic controller is connected directly to the electro-pneumatic positioner eliminating the need for an I to P converter. It also eliminates the dynamics introduced by the I to P converter and eliminates the need for calibrating the I to P converter.



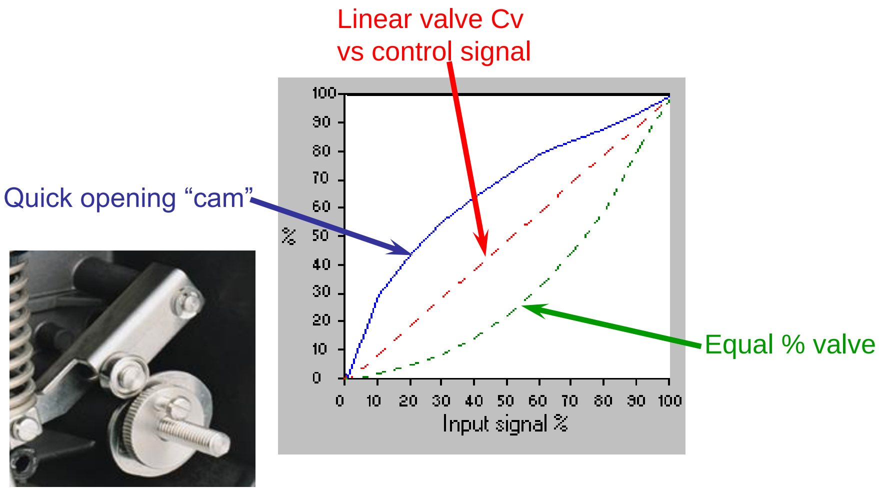
The next, and most recent step in the evolution of control valve positioners is the smart digital positioner. The levers and springs in the electro-pneumatic positioner are replaced by a micro processor. The microprocessor can improve performance and provide additional features.

It improves control performance because the microprocessor can run an advanced control algorithm.

The microprocessor can also generate diagnostic information that tells us if the valve's control performance is starting to degrade, so that we can plan maintenance at a convenient time before it impacts on product quality.

The connection diagram is the same as the one we saw previously for the electro-pneumatic positioner. The point is that there is nothing special that needs to be done to install a digital positioner. You simply connect the control signal and the air supply.

Positioners Permit Changing Flow Characteristic



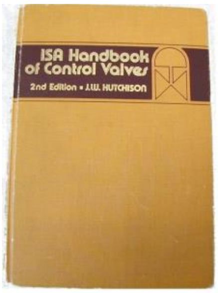
Positioners can change the flow characteristic of the valve. The mechanical analog positioners do this by installing specially shaped feedback cams. The digital positioners accomplish the same thing by modifying the positioners output through software.

Positioners, What They Do

- Reduce required actuator size
- Increase valve response time
- Increase actuator stiffness
- Allow use of double acting actuators
- Change flow characteristic..

This is a summary of the things that positioners do.

Emphasis in ISA Books



1976	1998
Pneumatic positioners: 6 pages	**Pneumatic positioners: 0 pages**
Electro-pneumatic positioners: 1 Paragraph	**Electro-pneumatic positioners: 0 pages**
	Digital positioners: 9 pages

Between the ISA control valve book published in 1976 and the one published in 1998 there has been a significant change in what type of positioners are emphasized.

Limit Switches

Electro-mechanical
- Most common
- Inexpensive
- High power applications
- 400,000 cycles.

Proximity
- Low power
  - Computer inputs
- Medium power (3 A max)
- 5,000,000 cycles…

Sometimes, both for on-off automated valves and control valves, there is the need for knowing when the valve is fully open or fully closed.

The actual switches are usually mechanical snap acting switches. These tend to be less expensive and have higher voltage and current ratings.

The inductive proximity switches usually have a much longer life and can be certified for use in hazardous atmospheres.

Position Transmitter

Position transmitter
4-20 mA output

For control valves there is sometimes the requirement for a position transmitter that will send a 4-20 mA signal that is proportional to actual valve position back to the control system.

Mounting Options

Rotary valve

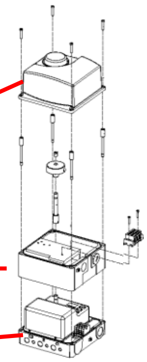

Linear valve

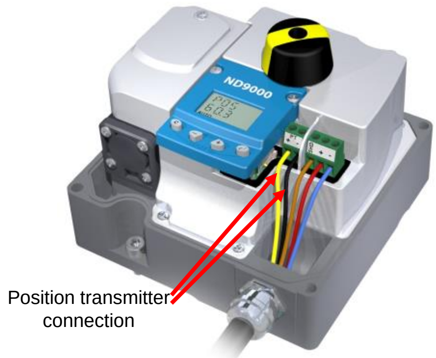

Positioner

Limit switches can be mounted directly onto actuators of on/off actuated valves. For control valves, many analog positioners can have the same types of limit switches and position transmitters mounted on the positioner so that the valve can have both a positioner and a limit switch and/or a position transmitter.

Limit Switch and Position Transmitter Options

For Smart Digital Positioner





Limit switch pack fits between positioner body and cover

Optional position transmitter is included on positioner circuit board

On the Metso digital positioner, the limit switch option consists of metal frame that installs between the positioner body and the positioner cover.

The position transmitter option has to be ordered with the positioner and consists of additional components on the circuit board and an extra terminal block inside the cover.

Solenoid Valves

Mainly used with in-off valves

3 – Way
For spring return actuators

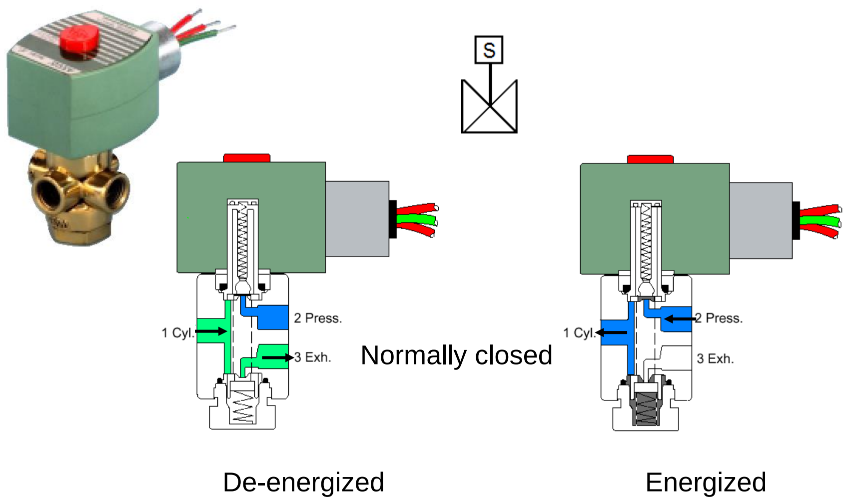
4 – Way
For double acting actuators

Solenoid valves are mainly used on actuated on-off valves, but sometimes with control valves to provide emergency shutdown action to control valves.

Three-way solenoid valves are used with spring return actuators, and four-way solenoid valves are used with double acting actuators.

Solenoid Valves

3 – Way for Spring Return Actuators



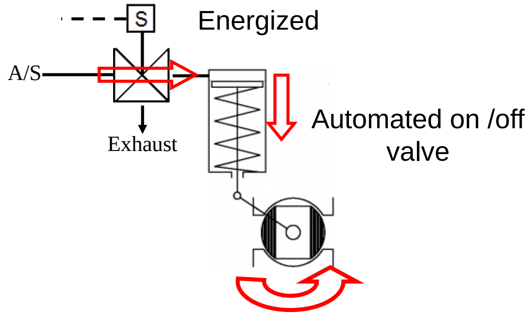
The most common configuration for solenoid valves on spring return actuators is the NORMALLY CLOSED configuration. The normally closed valve is one that when it is de-energized vents the actuator to atmosphere and when energized connects the air supply to the actuator.

Normally closed is the most common, because the valve actuator has the same failure action on loss of air as it does on loss of electricity.

Normally open solenoid valves are also available where the de-energized and energized actions are the opposite of what is shown here.

Solenoid Valves

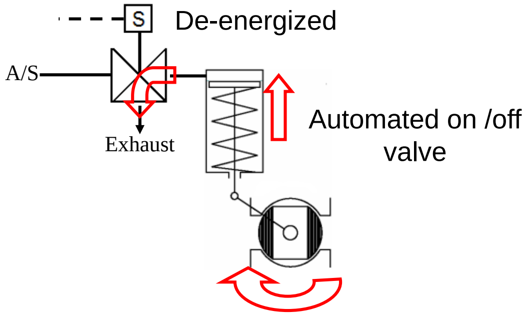
3 – Way for Spring Return Actuators



This figure shows schematically the flow through a 3-way normally closed solenoid when it is energized. In the energized state the air supply is directed into the actuator, opening the valve and compressing the spring.

Solenoid Valves

3 – Way for Spring Return Actuators

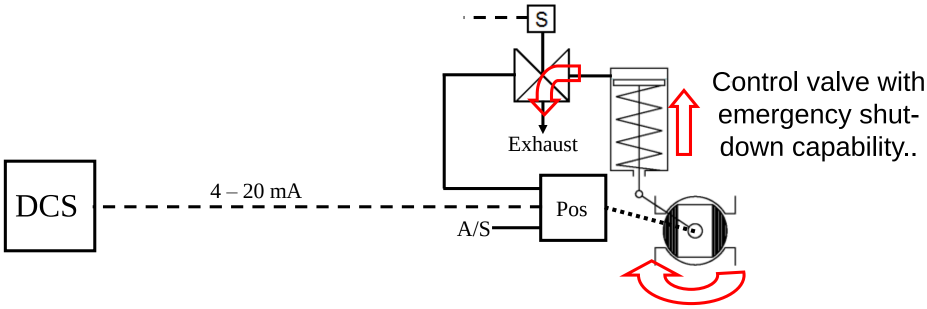


When the solenoid valve is de-energized, the air in the actuator is vented to atmosphere and the spring drives the valve to the closed position.

Although not shown here, valves with spring return actuators can also be configured for air to close and spring to open.

Solenoid Valves

3 – Way for Spring Return Actuators



Here we see a valve that is normally a control valve, but that can also be closed in an emergency by a signal from another source such as an emergency shutdown system.

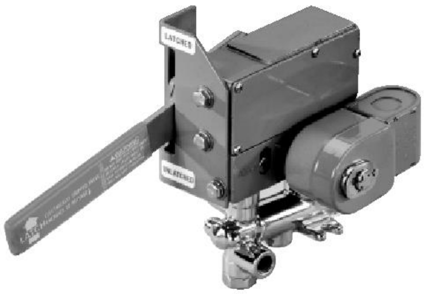
The 3-way solenoid valve normally directs the air from the positioner into the actuator, but when the solenoid is tripped by the emergency shutdown system, the air in the actuator is vented to atmosphere, and the control valve closes.

Some experts discourage relying on control valves to also serve as emergency shut down valves.

Solenoid Valves

3 – Way for Spring Return Actuators

Manual Reset



Operation alternatives:

- No voltage release
- Electrically tripped
- Free handle.

- Prevents inadvertent start-up
- Once tripped must be manually reset to automatic operation

Some critical processes must not be accidentally started up after being shut down.

The manual reset solenoid valves require that the control signal be first applied to the valve and then someone must go to the valve and physically reset the handle.

The most common configuration is the no voltage release option, where electric power energizes the solenoid, and the solenoid is tripped by turning off the power.

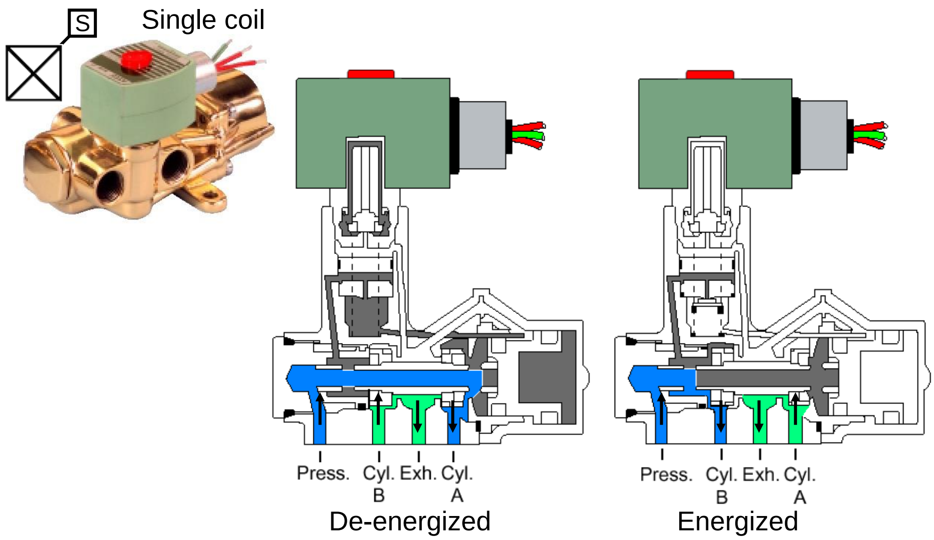
The electrically tripped option means that sending a momentary or continuous signal trips the valve.

Both of these options can be manually actuated by turning the handle, but they cannot be reset to automatic operation unless in the case of the electrically tripped valve, the solenoid is de-energized or in the case of the no voltage release valve the solenoid is energized.

Free handle means that the valve trips when the solenoid is de-energized and the handle will only cycle the valve if the solenoid is energized.

Solenoid Valves

4 – Way for Double Acting Actuators

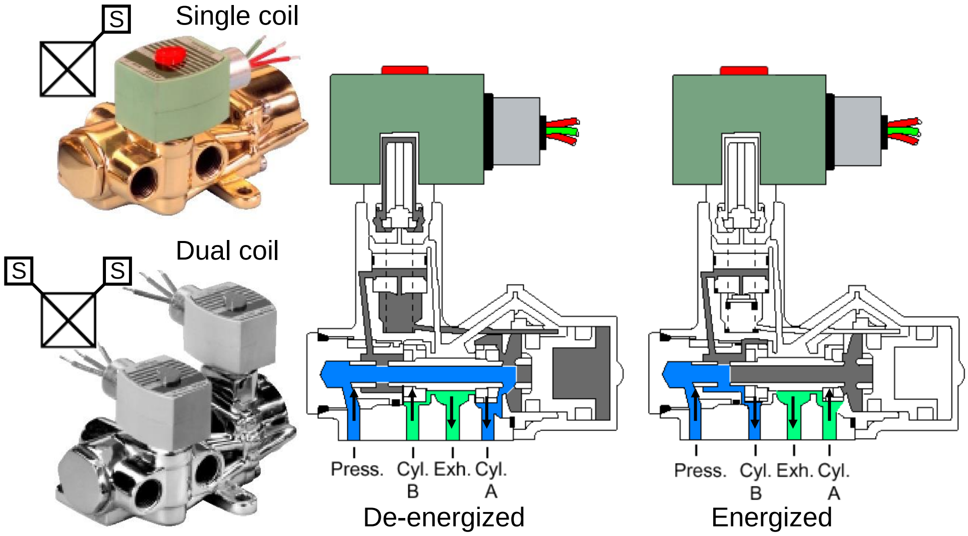


Four-way solenoids are required to actuate double acting actuators. The most common configuration is the single coil version which vents and pressurizes a double acting actuator as shown in the figure.

When the coil is de-energized, air is directed into one port of the actuator and the other actuator port is vented. When the solenoid is energized, the pressurized and vented ports of the actuator switch places.

Solenoid Valves

4 – Way for Double Acting Actuators



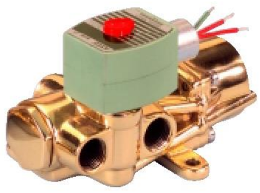
Less often, dual coil solenoid valves are used. Energizing one of the coils sets the valve for one flow path, and energizing the other coil switches the valve to the other flow path.

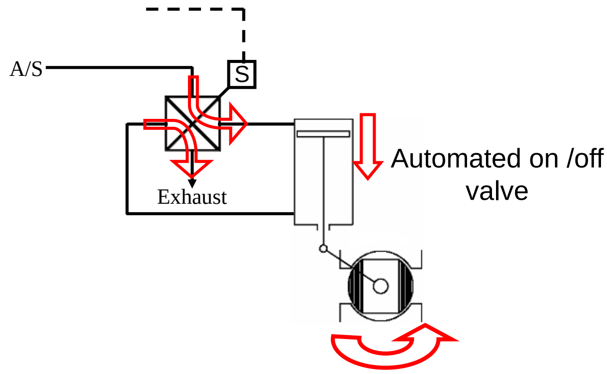
Only a momentary energization is required, however the coil can also be left energized continuously. However it is not allowed to energize both coils at the same time.

Dual coil solenoid valves are useful where electrical power is limited, such as systems than operate from solar cells, or in cases where the valve that is being controlled by the solenoid valve must remain in place on loss of electrical power.

Solenoid Valves

4 – Way for Double Acting Actuators

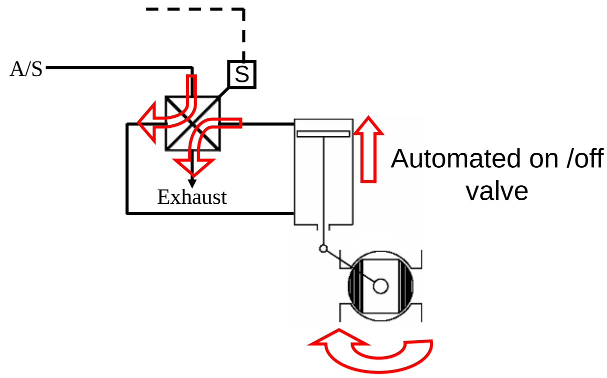




This figure shows schematically the flow path through a 4-way solenoid valve when the actuated valve is being opened. Air is directed into one side of the cylinder and vented from the other.

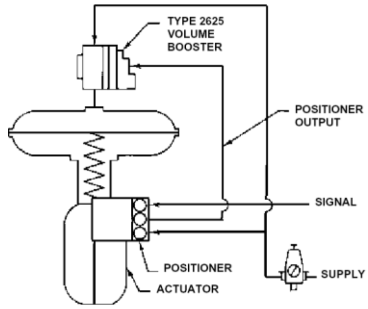
Solenoid Valves

4 – Way for Double Acting Actuators



This figure shows schematically the flow path through a 4-way solenoid valve when the actuated valve is being closed.

Volume Booster



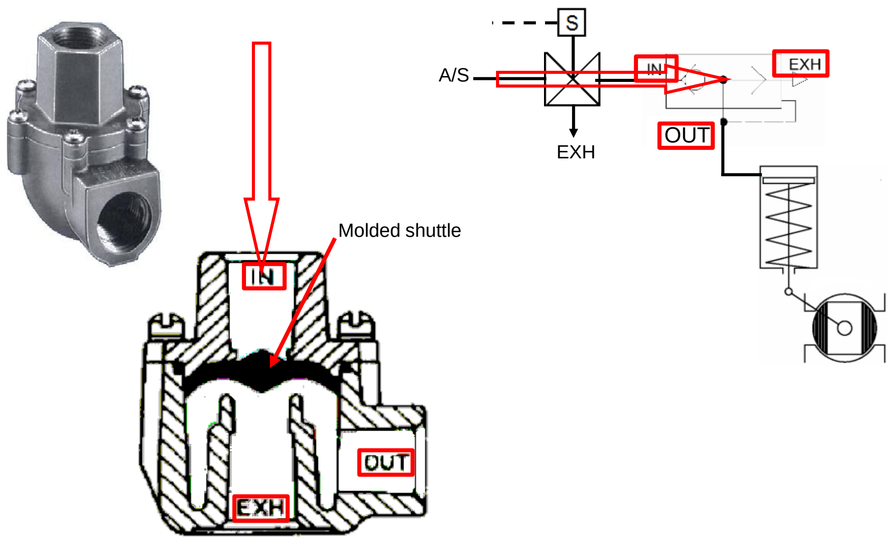
- 1 to1 high capacity pneumatic relay.
- Used when the positioner cannot supply enough air for large vales that must respond quickly
- Rarely needed…

A volume booster is a one to one high capacity pneumatic relay, meaning that it duplicates the input pressure at its output, but can supply a greater volume of air than a positioner.

They are used when a large valve needs to move very fast and the positioner can’t supply a high enough volume of air.

We don’t see volume boosters used very often.

Quick Exhaust Valve



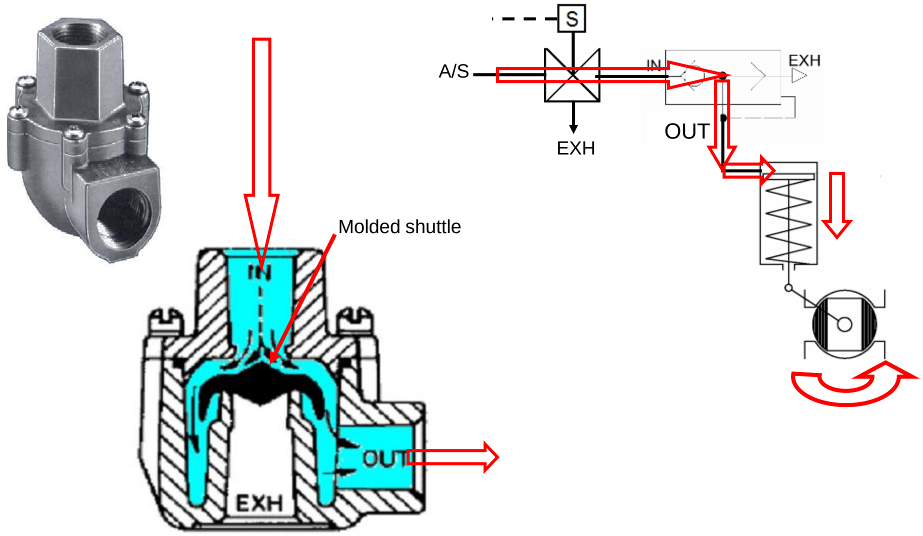
In some cases, such as emergency shut down valves, a valve is required to close very quickly. Some 3-way solenoid valves are designed to have a higher flow capacity in the venting direction than they have in the pressurizing direction.

If this does not result in fast enough operation, a quick exhaust valve can be added to the circuit as shown in the diagram.

The three ports, IN, OUT and EXHAUST are labeled in both the schematic diagram at the upper right and the cutaway drawing at the lower left.

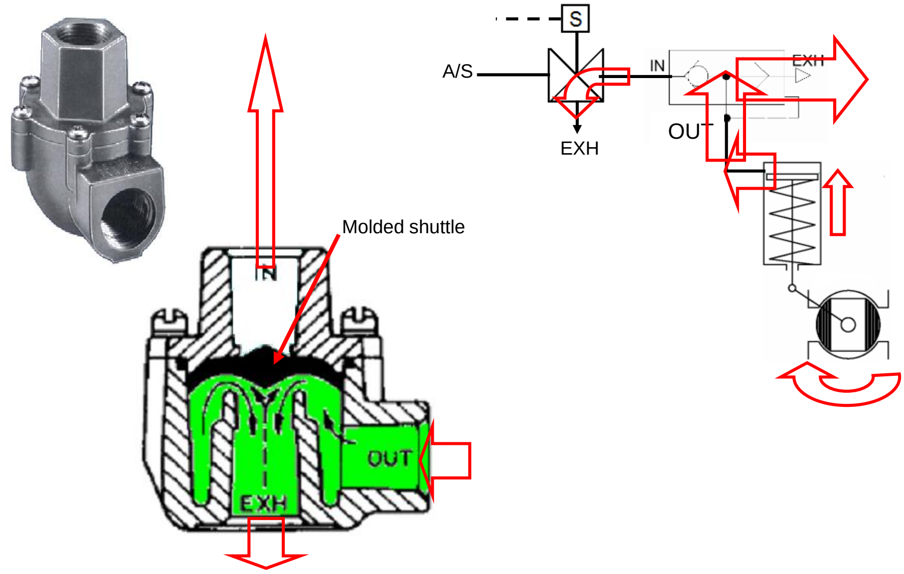
When the process valve is being opened, the solenoid is energized and the air supply goes through the solenoid valve.

Quick Exhaust Valve



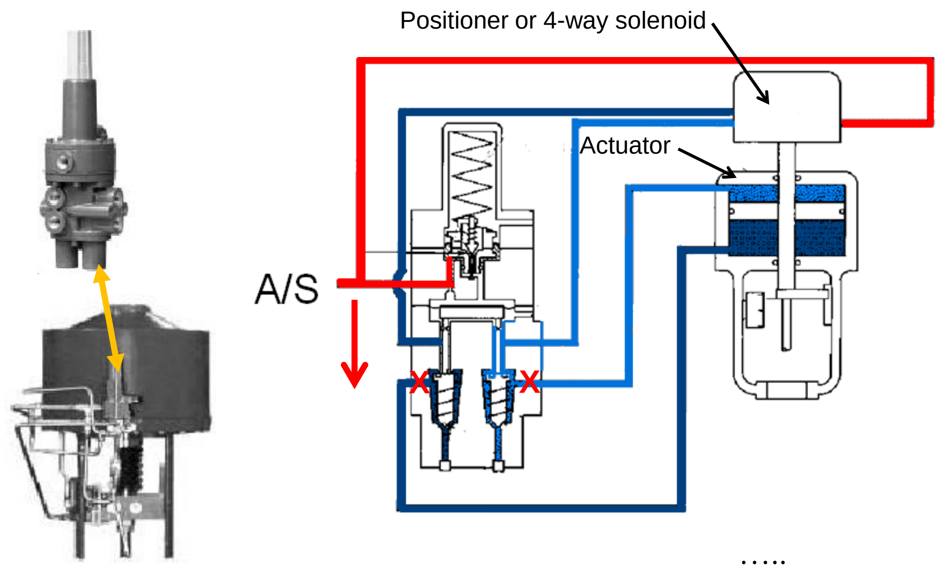
Pressure in the “IN” port of the quick exhaust valve is higher than in the “OUT” port and the “shuttle” flexes downward and flow through the quick exhaust valve is as shown in the cutaway view at the lower left-hand side of the slide. This opens the process valve.

Quick Exhaust Valve



When the process valve is called upon to close, the solenoid valve is deenergized and the solenoid starts to vent. This causes the pressure in the “IN” port of the quick exhaust valve to drop below the pressure in the process valve’s actuator and the “OUT” port of the quick exhaust valve causing the “shuttle” to flex upward opening a flow path from the “OUT” port to the exhaust port, abbreviated “EXH” in the diagram. Because the ports in the quick exhaust valve are large compared to the ports in a solenoid valve, the process valve’s actuator vents very rapidly, moving the valve to the spring position very quickly.

Lock-up Valve



Double acting actuators are often referred to as having a “fail in place” action, since if the air pressure on the pressurized side of the piston decays there is no restoring force on the other side to cause it to move.

In the real world the process usually exerts dynamic forces on the valve and in the absence of air pressure in the actuator these dynamic forces may be able to move the valve.

A lock-up valve, piped between the ports of the “control device”, which could be either a 4-way solenoid or a double acting positioner, senses the air supply pressure. When the air supply pressure drops below a preset value, the lock-up valve blocks off the actuator ports, blocking in whatever pressure was in the actuator at the time of the trip, effectively holding the actuator (and process valve) in place.