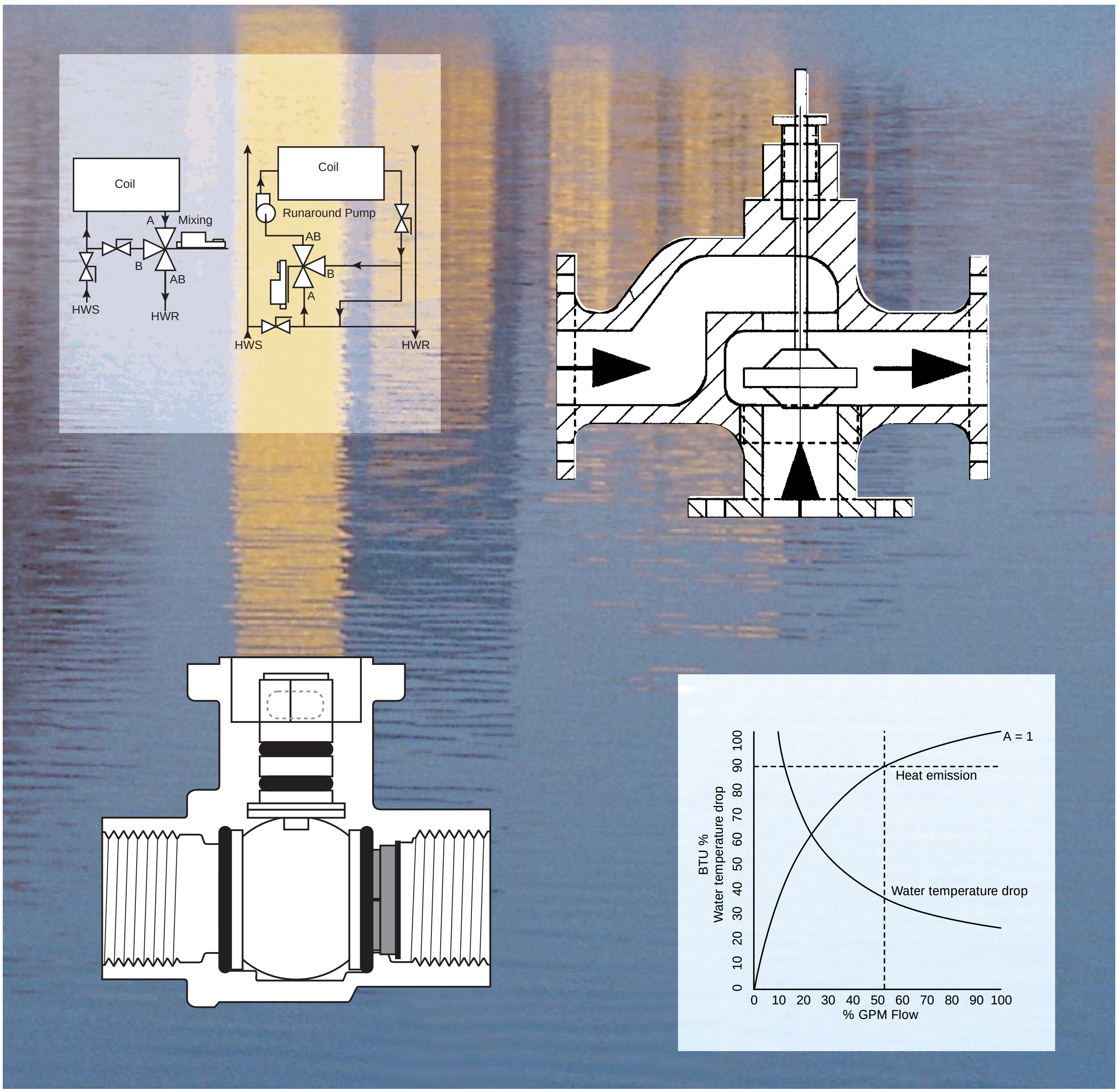
Actuators for heating, ventilation, air-conditioning

ELECTRONIC VALVE APPLICATIONS GUIDE

V4.2



BELIMO

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1. INTRODUCTION

The publication of this booklet coincides with the introduction of the Belimo Characterized Control Ball Valve. The section on ball valves in general and the Belimo Characterized Control Valve is expanded to give a full understanding of the problems with existing ball valves and the advantages of using the engineered approach of the special design of the BelimoControl Valve which is a modified ball valve.

This is the first new design in HVAC control valves since the inception of DDC control.

Valve sizing cannot be accurate without firm understanding of the hydronic system. While valves are the final control element, the balancing, coil sizing, coil response curve, and the piping losses all play a role in accurate sizing. A control valve cannot be properly sized without full analysis of the system. If the designer of the system requires high accuracy, then a rigorous analysis is required and the designer may have to size the valves.

Everything interacts in a hydronic system, so the valves cannot be treated as a simple subject by themselves. Instead, the whole hydronic system has to be taken into account. There is a good deal of data necessary for careful sizing and selection of valves, but when all of it is not available, the accuracy will suffer. There are times when the specification does not take as-built conditions fully into account. Valves may be missized by adherence to the specification and the design engineer should be consulted.

Every process, every coil, has a unique response. It is most difficult to fully study each and design a special characterized valve to control the process. But we can do some things to improve control.

1. We select a valve with an overall characteristic which most matches the process. If we cannot do full hydraulic analysis of the system, we can choose linear or equal percentage depending on which is closest to the response needed.
2. We select a valve with the proper capacity.
3. We balance the system.
4. We eliminate excessive pump head.
5. We select an actuator which is capable of good control and we tune the control loop.

The approach detailed above is common in process control. Systems work with good results.

It must be considered an error for the control engineer to unquestioningly accept a given control system and size and select valves in a stereotyped manner from a limited range of possibilities. A thorough analysis is needed.

Under any circumstance, this booklet should give a general knowledge of how the control valves and the hydronic system interacts. Hopefully, this will lead to good system design.

2. TYPES OF VALVES

Two, three, and four-way valves
Two-way valves have two ports and are used to control the flow in variable flow systems. *See Figure 1.* This is a 2-way globe valve.

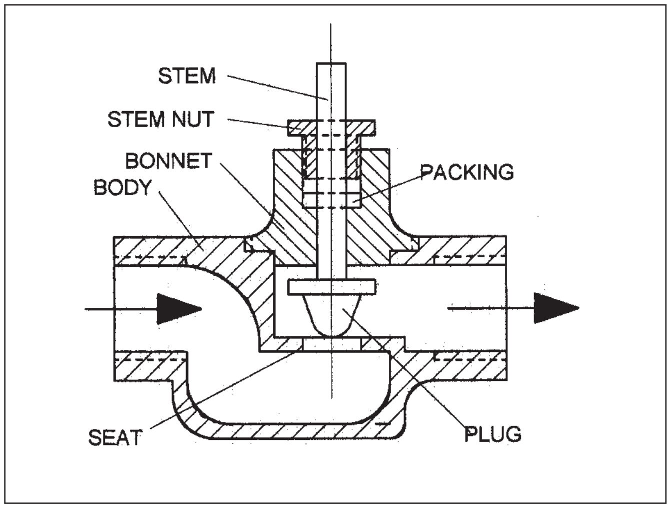

Figure 1 - Two-way globe valve

Three-way valves have three ports. One port is common to the other two. Three-way valves are used in constant flow systems. See *Figure 2* for a 3-way globe valve. Three-way valves can be piped for mixing or diverting service. Mixing function means that the flow can enter two ports and exit through the common port. Diverting function means that the flow enters the common port and exits through the other two ports. Damage may result if it is piped opposite to the intended function of the valve. The two are not interchangeable.

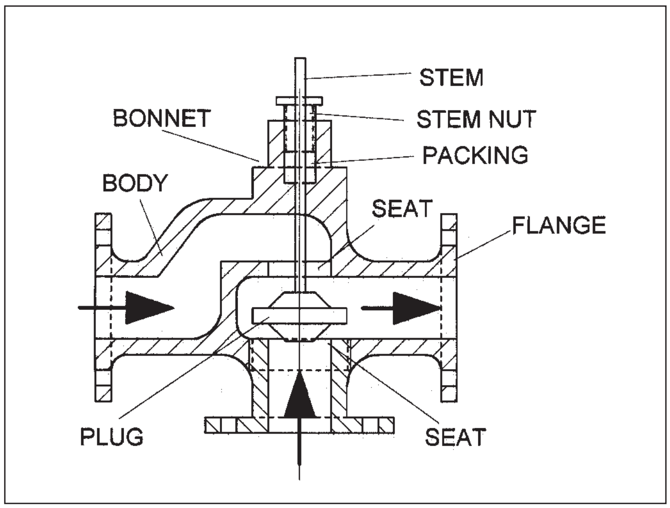

Figure 2 - Three-way globe valve

Four-way valves have four ports and usually are slide type valves. They can serve two constant flow loops, each with its own pump. Two ports are connected to the first loop (boiler) and two ports are connected to the second loop (peripheral system). These are common in Europe, but not in the Americas.

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Globe valves
Globe valves have a linear stem movement, which operates a plug against one seat (two-way valves) or between two seats (three-way valves). Regular two-way valves have one seat only. Three-way valves have one upper and one lower seat.

The relatively poor accuracy of the globe valve is discussed in Section 5 on rangeability.

In general globe valves are used in pipe sizes from $^1/_2$" to 6".

Standard Ball valves
Ball valves are made in both two and three-way configurations. They have an internal ball which has a hole drilled right through it. The flow is controlled by turning the ball 0 - 90 degrees versus the flow path through the body. They have high pressure bodies and very high close off capabilities.

Ball valves were used in the early 1960's in process control as control valves. They entered use in the HVAC market for control purposes when Belimo brought the direct coupled actuator to the market and the natural fit was recognized.

Three major variants are available: flat ball reduced port, round ball reduced port and standard full port. Many of these have been adapted for use as modulating valves by drilling bolt holes in the bonnet.

<u>Flat ball reduced port</u> valves have a flat on the rounded ball and the hole is drilled through the middle. See *Figures 3a and b.*

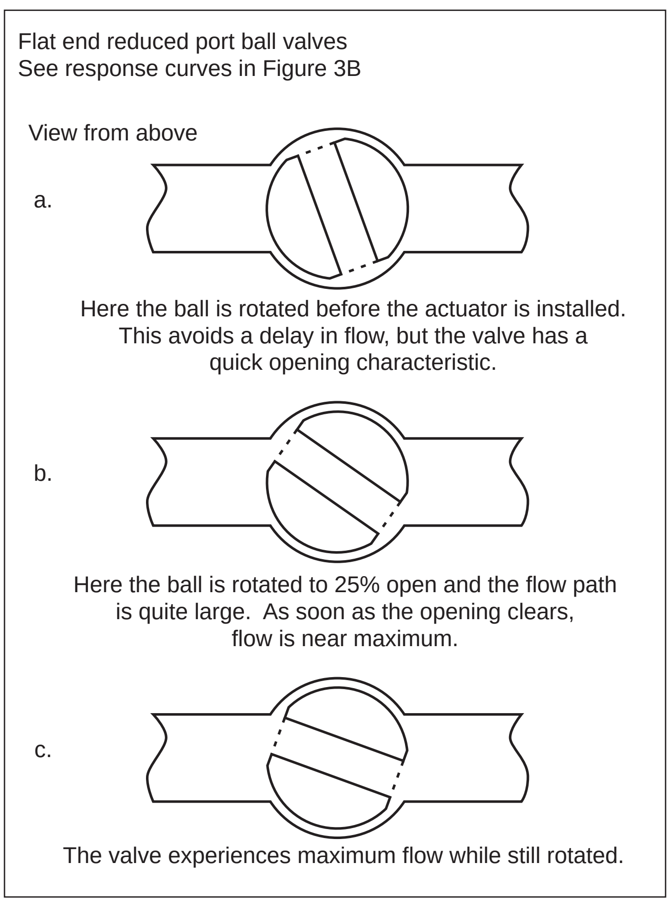

Figure 3a - Flat end reduced port ball valve

These valves have a poor flow response curve and should be used for 2-position control only. When they first rotate there is a delay before flow occurs while the hole moves to the area open to flow. Then the flow increases rapidly as the hole clears the restriction of the body. After that there is no increase in flow. Flow goes from zero to 100% over about 40° rotation. This is a quick opening valve.

<u>Round ball reduced port</u> valves do not have as bad a quick opening characteristic, but they have more delay. With the ball set up for a 90° rotation, they delay opening until about 20° rotated. By 50° open, they have reached full flow. In order to avoid the delayed response, it is possible to install the actuator with the ball turned about 10-20° nearer open. While this avoids the delay, the quick response remains, and the valve reaches full flow at about 40% open. This is a "delayed response, quick opening" valve. Again, it should be used for 2-position control, not modulation.

<u>Standard full port</u> ball valves are inexpensive and have been used for years for shut off. See *Figure 4a.*

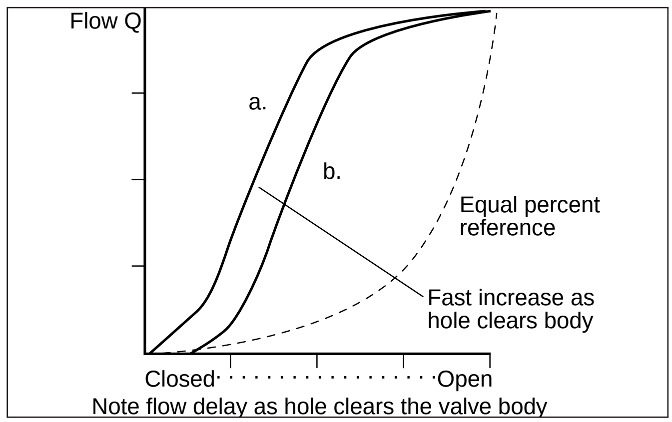

Figure 3b - Reduced port ball valve flow characteristic

The published flow curves have not proven to be accurate in testing. There is a delay in flow until the hole clears . The response curve is logarithmic (shallow equal percentage) along its middle portions, and then the flow flattens and does not increase appreciably during the last 30° of rotation. Reduced ports vary in size from about 60% of the pipe size down to quite small. The response curve worsens as the C_V goes down with the small ports. This is shown in *Figure 4b*.

In addition, the high capacity of the valve causes problems.

The problem is that the C_V of the ball valve is about 4 times that of the globe valve. This is fine for 2-position control. But the maximum C_V for modulating control of flow through coils is better matched by the globe (although most globes are still slightly too large). As a result, by the time the standard full port ball valve is about half open, the flow has reached the maximum design for the coil. The flow characteristic is still not correct. The rangeability is seriously reduced and actuator resolution is half that possible. Instead of a 2-10V operating voltage range, a 2.5V (due to delay in opening) -6V is likely.

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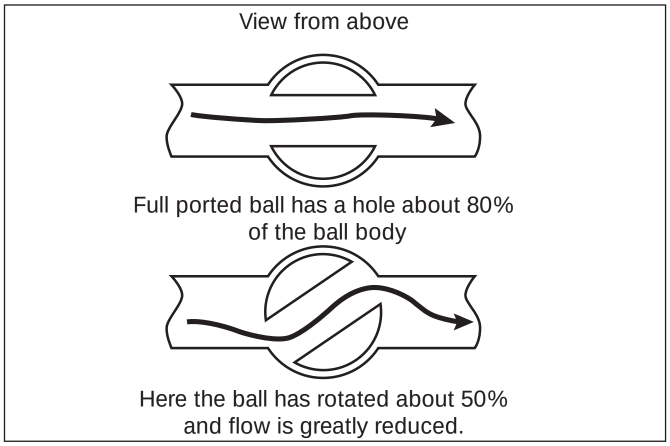

Figure 4a - Full ported ball valve

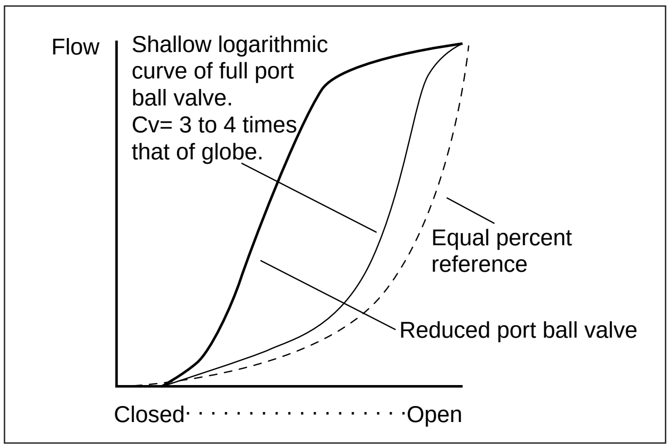

Figure 4b - Full port ball valve flow characteristic

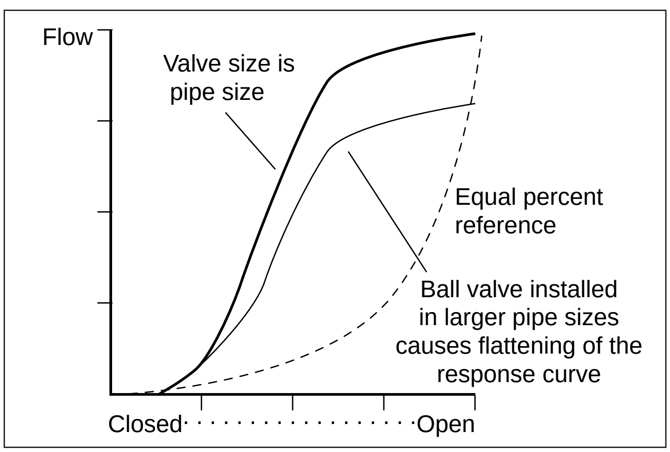

Figure 4c - Distortion of curve due to pipe size change

Standard ball valves are good for 2-position control only. When used in modulating control they have a high rangeability, but a limited range of signal output before they reach full flow.

In general standard ball valves are used in pipe sizes from $^{1}/_{2}$" to 3" in the HVAC industry.

There are two other factors which distort the flow curves shown in the figures here. One, if the valve is smaller than pipe size, then the curve is flattened. See *Figure 4c.* The full open flow is less, but the partial open flow is mostly unaffected. Two, the distortion in flow curves shown here for the ball valves are worse when installed due to authority considerations. This is covered in the sections which follow.

Belimo Characterized Control Ball Valves

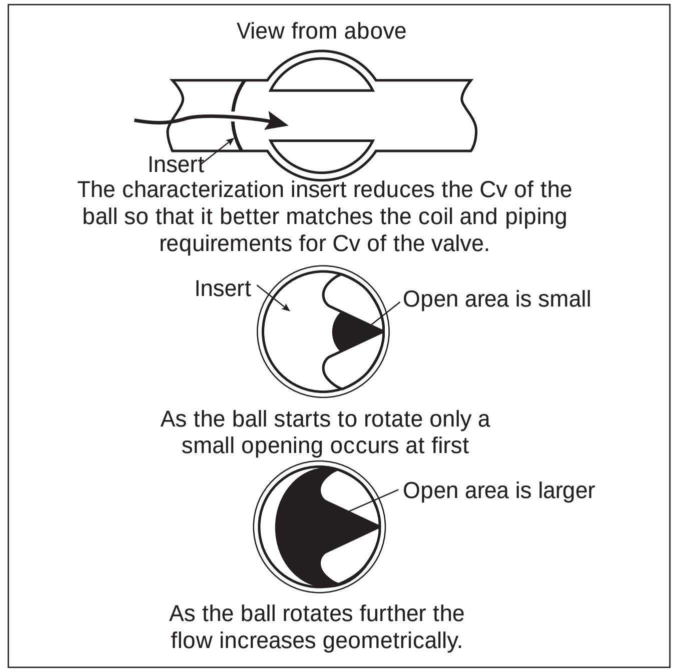

Figure 5a - Belimo Characterized Control Ball Valve

Belimo has designed special ball valves for control. These have characterized inserts or special shape balls. The flow characteristic can be true equal percentage like a globe valve. This is discussed further in the section on flow response which follows coil characteristics. The front cover and *figures 5a and b* show the Belimo characterized control ball valve. While getting low cost and the advantages of an industrial type valve, there is no sacrifice of controllability as with other ball valves.

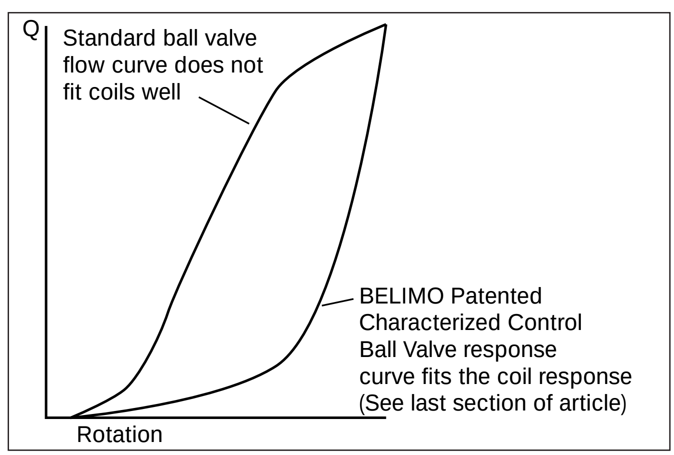

Figure 5b - Belimo characterized control ball flow characteristic

The Belimo innovation is low cost while maintaining quality better than HVAC commercial valves.

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Shoe valves
Rotary slide valves have a cylindrical inner surface, with openings in the circumference. A slide ("shoe") is operated by the turning motion of a stem, and covers or uncovers the openings, thereby controlling the flow. In process control this type of valve is referred to as a "trunion mounted characterized port ball valve."

Butterfly valves
Butterfly valves are two-way valves that use a disk to control the flow. The disk rotates 0-90 degrees inside a ring-shaped body, which usually is sandwiched between two pipe flanges. When the disk is parallel with the pipe, the valve is fully open. When the disk is perpendicular to the pipe, the valve is closed. See *Figure 6.*

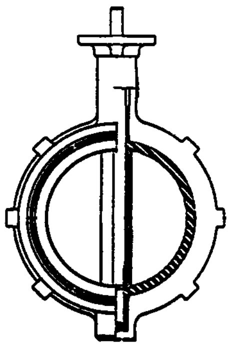

Fig. 6 - Butterfly valves

Often, the body of a butterfly valve has an inner lining of resilient material that provides a tight seal against the disk when the valve is closed. When a butterfly valve is partially open, dynamic forces will act upon the disk and produce a torque that tends to open the valve. The forces peak between 60° and 85° and drop to near zero when full open. This tendency is one reason why convention has been to limit the full opening of a butterfly valve to between 60° and 70°. It is important that the actuator is powerful enough to produce a sufficient torque to operate the valve from closed to fully open. Valves with a resilient seat requires a quite high torque to operate the disk near the closed position. Undercut discs reduce close off torque requirement and close off pressure rating while keeping leakage low below the reduced pressure rating. This is standard practice in HVAC where high close off pressures are not often required.

The butterfly valve has a flow characteristic similar to the full port ball valve or a modified parabolic. See *Figure 11.*

Two butterfly valves can be linked together so they operate as a three-way valve. They are mounted on a pipe-T and are linked so when one valve closes, the other valve opens.

Ball valves, butterfly valves and rotary slide valves are operated by a 0 - 90 degree movement. It is therefore easy to operate these valves by rotary electric actuators.

3. COIL CHARACTERISTICS

Heating Coils
Figure 7 shows the relationship between the heat emission and the flow of water through a typical heating coil. Different types of coils have different characteristics, but the basic "convex" shape remains essentially the same — the only difference is how pronounced the curvature. This depends upon the type of heat exchanger, the water side temperature drop, the air side temperature rise, and on the relative values of the water and the air.

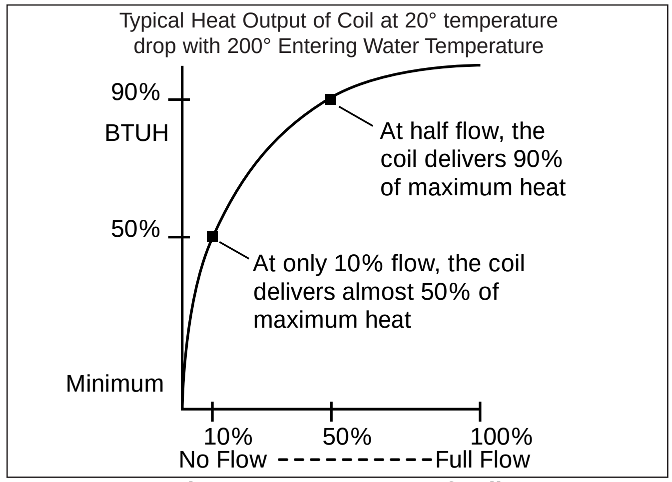

Figure 7 - Heat output of coil

The "convex" shape of the curve means that when the flow increases from zero, the heat emission increases at a high rate in the beginning, but as the flow is increased, the rate of increase decreases. The reason for this is that at small flows, the water takes a long time to pass the coil, so the temperature drop of the water will be large (effective use).

Conversely, when the flow is increased, the water spends less time inside the coil and the temperature drop of the water is less.

A coil is selected for a specific airflow and heat emission. A specific temperature drop of the water flow through the coil is produced only at these design conditions. This is the "design temperature drop". On the air side, there is a design temperature rise.

Figure 8 shows the different characteristics of coils with a design temperature drop of 10, 20 and 60°F respectively.

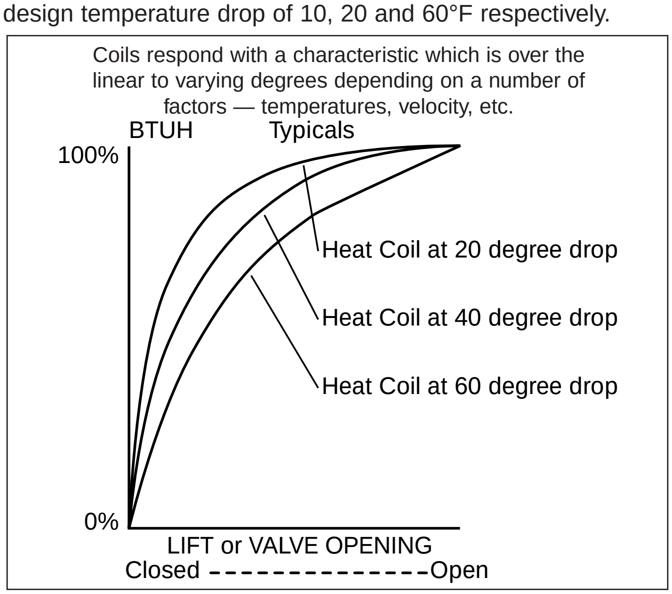

Figure 8 - Heat output of coil with varying temperatures

The coil response does not differ much with constant flow but varying temperatures are used. Thus the use of 3-way valves and runaround pumps is nearly the same as throttling the flow from 0 to 100% with 2-way valves. *Figure 25* has a coil emis-

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sion curve shown along with a temperature drop curve on the same graph.

Cooling coils
Figure 9 shows the characteristic response curve of a cooling coil. When studying the sensible heat emission versus the water flow, we find that the characteristics resembles a heating coil.

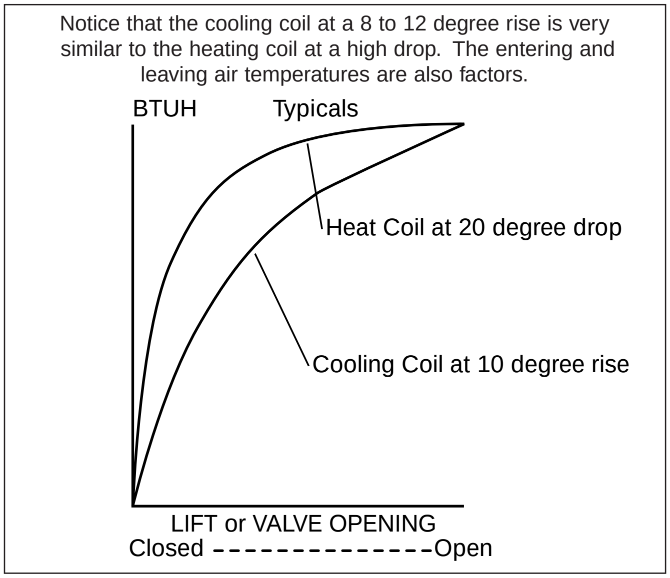

Figure 9 - Response curve of cooling coil

Total heat also includes the latent heat removal (moisture). Dehumidification is a very important aspect, but with respect to the stability aspects of the temperature control, it is the sensible heat curve that is of the determining factor. The curve is much closer to the linear than that of a typical heating coil. The water side drop is 10° instead of 20°, the air side change is from 75° to 55° instead of from 70° to maybe 120°.

For simplicity, the following text refers to heating coils. However, what is said also applies to cooling coils, except that heat is rejected and the temperature of the water passing through the coil increases. The curves however are different. Strictly speaking a typical cooling coil needs a different valve characteristic than a heating coil.

As we will see the control valve can be characterized in order to compensate for the non-linearities in the coil and process response.

4. THE FLOW COEFFICIENT

The correct sizing of the control valves is of the greatest importance for an HVAC system. Naturally, the valve must be large enough to supply the maximum required flow when fully open. However, it is very important when modulating control is used, that the valve is not oversized. When a valve is too large, the maximum required flow is already supplied when the valve is partially open. This means that just a fraction of the available stem movement is used. A small change of the stem results in a disproportionate large change in the heat output, especially when the valve begins to open. The system is therefore extremely sensitive at low and average loads, so stable control is hard to accomplish.

In order to size control valves the required flow coefficient must be calculated.

The flow coefficient is expressed as the C_V-value, which is defined as "the flow in GPM (US) of 60°F water through a fully open valve when the differential pressure across the valve is 1 psi".

By definition, $\mathbf{C_V = GPM/\sqrt{(\Delta P/g)}}$ or after rearranging, $\mathbf{GPM = C_V \sqrt{(\Delta P/g)}}$
Where g is the specific gravity of the fluid, water = 1
$\sqrt{\Delta P}$ is in psi

K_V is the metric counterpart to C_V. K_V of 100 = Cv of 116. Multiply the K_V value by 1.16 to obtain the C_V value. K_V is the m^3/h of water flowing through the valve at 100 kPa pressure drop.

C_V calculations
The C_V value is used for valves. It can be calculated for any component in a system. For example, a heat exchanger with a 4 psi pressure drop at 100 GPM has a $C_V = 100/\sqrt{4} = 50$.

C_V is a flow quantity. When 2 valves are piped in parallel the Total $C_V = C_V1 + C_V2$.

Usually in piping we use K loss coefficient factors to express resistance. K x Hv, the velocity head pressure, is the total pressure loss of a fitting. But C_V could be used for series systems if desired.

For capacities in series, if C_V is used, C_V Total can be found using

$$\left(\frac{1}{C_V1}\right)^2 + \left(\frac{1}{C_V2}\right)^2 + \ldots\ldots\ldots = \left(\frac{1}{C_V\text{Total}}\right)^2$$

This is derived from $C_V = 1/\sqrt{K}$ and K1 + K2 + … = K total for a series of pipe elements. Derivation can be found in ASHRAE S41.8 which uses R in place of K. K is more commonly found in fluid mechanics texts. Therefore

$$K = \left(\frac{1}{C_V}\right)^2$$

5. VALVE CHARACTERISTICS

Control
There is one assumption made during the discussion of matching coil to valve characteristic. That is that the control signal output and the actuator have linear characteristics. This is to say that a 1V signal increase results in the same rotation or lift of a valve regardless of its location on the signal range. Typically a 2 to 10 V signal is used in HVAC. The change from 2 to 3 V results in the same lift or rotation as the change from say 8 to 9 V. In addition the loop tuning constants, whether PI (D) or fuzzy logic generated, assume a mechanically linear process.

Notice that the output of a coil is decidedly non-linear. PI (D)

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was developed with more or less linear processes in mind. Loop tuning can compensate for non linearity, but the loop tuning time is excessive and many controllers do not have the ability to have multiple gain values. Even self tuning loops have difficulty as the response curve changes with other system variations. This will become clear as we progress.

There are 2 matters here which we should not confuse:

The C_V of a valve is the GPM which will flow at 1 psi pressure drop when the valve is full open. This is always published.

The GPM of flow at modulated positions is also measured at a 1 psi drop. These values determine the response curve. The values are also called C_V, but at a certain intermediate position. It is rare in HVAC for these values to be published. The manufacturer simply states "linear" or "equal percentage". In process control the values are almost always published, even if calculated rather than measured in testing. Process engineers size valves carefully using the C_V at modulated positions.

Characteristic curves
The valve characteristic is determined at laboratory conditions. The pressure drop across the valve is held constant at 1 psi and its flow quantity is measured. (This could be water, air, or other fluid.) The valve is opened at 10° steps and a curve is graphed. The ISA (Instrument Society of America) requires that three different pressures be used and the average of the values is the result published. Thus the C_V of a valve is an approximation, albeit a close one. See *Figure 10.* The small variations due to pressures in lab testing are inconsequential compared to authority and valve type variations. This is discussed in sections below.

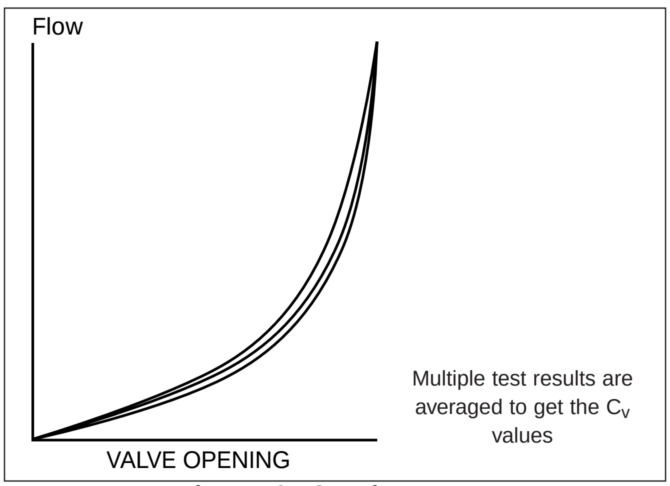

Figure 10 - C_V value curves

The characteristic is determined by the shape of the ports, ball, butterfly blade, or plugs of the valve. There are other characteristics than discussed below, but these are the main ones found in HVAC.

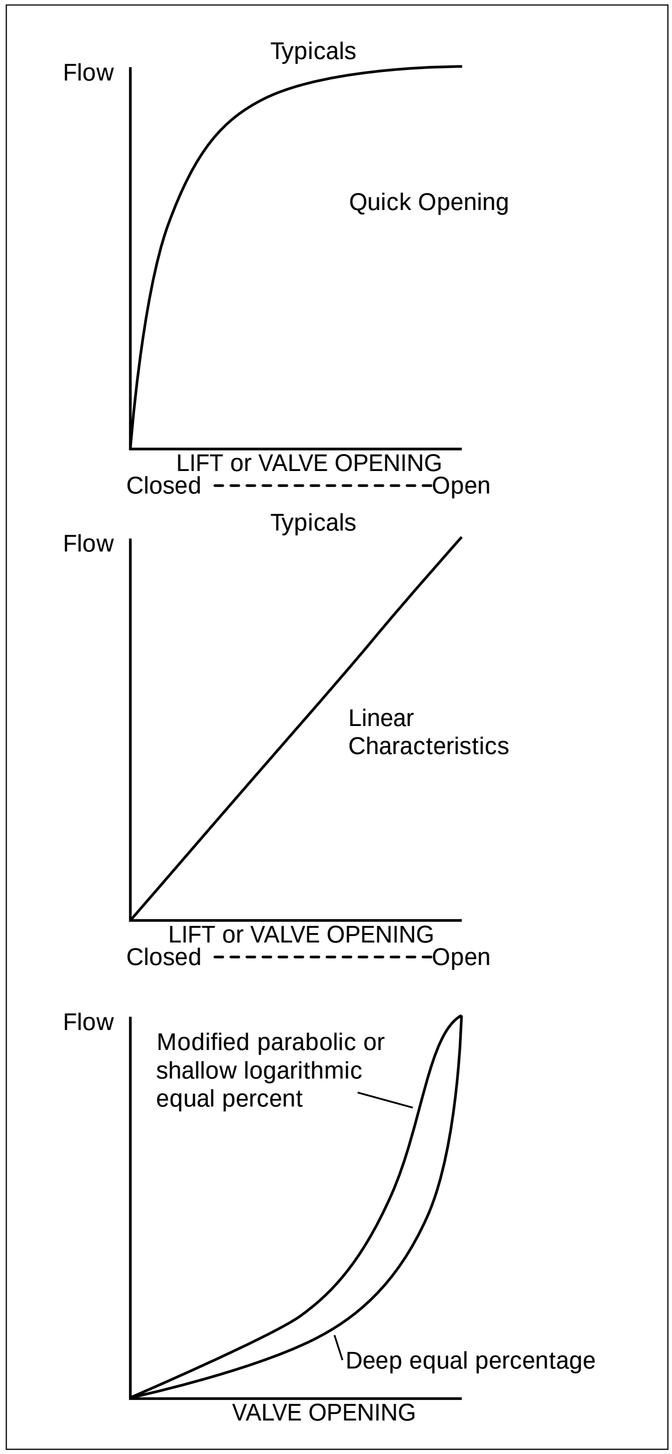

Figure 11 - Response curve of various valves

Quick Opening
Quick opening globe valves have a "plug" that is just a flat disc, which is operated against the seat. As soon as the disc lifts from the seat, the flow increases very quickly. This type of characteristics is suitable for on/off control. It gives a large flow capacity (Cv-value) compared to the valve size. See *Figure 11.* Flat end reduced port ball valves delay opening and then increase flow quickly. Globe valves can be manufactured with a quick opening characteristic.

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Linear
The flow is proportional to the position of the valve stem. It is also used with some three-way valves. Linear characteristics are used in some two-way valves for pressure or steam control. See *Figure 11.* Two-way globe valves can have a linear character but their use is limited. Three-way mixing and diverting valves are available as linear or equal percentage.

Modified parabolic
As shown in *Figure 11* this curve falls between linear and the traditional equal percentage characteristic. They have good modulating characteristics near closed, but become insensitive to further opening at near full open. They are also properly described as shallow equal percentage. A number of valve marketing brochures refer to this curve as equal percentage in efforts to draw comparison to the “real” equal percentage curve that globe valves provide.
Butterfly valves and standard and full ported ball valves have characteristics close to the modified parabolic curve.

Equal Percent
The equal percent characteristic gives a non-linear relationship between the flow and the stem position. At first, when the valve begins to open, the flow increases at a small rate, but as the valve is open further, the rate gradually increases. The valve curve is always under the linear. See *Figure 11.* As it happens, the coil curve is also equal percentage, but over the linear and complementary to the valve curve type.

The reason why this characteristic is called “equal percent” is that, when the valve is opened in equal percent increments, the flow increases by an equal percentage number over the previous value. Conversely, the flow decreases by an equal percentage number, when the valve is closed in equal percent increments.

There are a number of different curves which follow a formula which could be called equal percent. All are logarithmic curves. Some equal percentage curves are so shallow that they will not control a coil well.

In the following sequence, starting with 100 GPM when fully open, the flow is decreased by 30% for each 10% increment decrease of the opening.

%Open	100	90	80	70	60	50	40	30	20	10	0
GPM	100	70	49	34	24	17	12	8	6	4	(2.8)

This theoretical curve “aims” at 2.8 GPM for the 0% position. However, at the 0% position the valve needs to be closed. Therefore, the real curve is slightly different near the closed position. In addition leakage at 0% may exist. The flow follows the equal percent curve down to 4 GPM.

The percentages and flow quantities above are typical of the deep equal percentage curve of the globe valve, characterized control ball valve, and even of the opposed blade damper. A shallower curve occurs with standard ball valves, butterfly valves, and parallel blade dampers.

All equal percent characteristics are not created equal. The curve can be more or less pronounced depending upon the percentage number used.

The “concave” characteristics of the valve characteristics counteracts the “convex” nature of the coil. The intended net result is that the heat output becomes proportional to the stem position. Therefore, the equal percent characteristics is suitable for modulating control (proportional) of heating and cooling coils and other water based heat exchangers.
Most globe valves and Belimo characterized control ball valves have equal percentage characteristics.

Pipe Geometry
When a smaller valve than the pipe size is used, pipe reducers have to be used. The resultant C_V-value of the reducers & valve will be less than the nominal C_V-value of the valve. See *Figure 12.* F_p is the piping geometry factor.

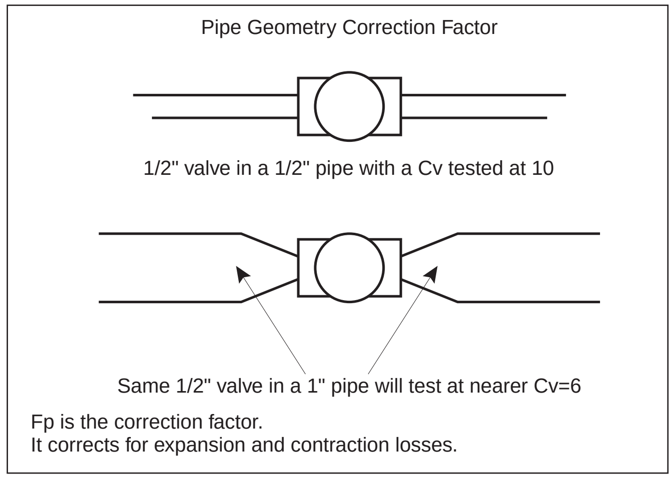

Figure 12 - Pipe geometry

The corrected C_V-value for different combinations of valves and pipe sizes are shown in tables for various types of valves in Belimo’s *Valve Sizing and Selection Guide Doc.*, a companion booklet to this.

The pipe geometry has a strong effect upon valves which have a large C_V-value compared to their size, such as full ported ball valves and butterfly valves. Most globe valves and reduced port ball valves are affected relatively little. The pipe geometry has its largest influence when the valve is fully open, but it is insignificant when the valve is almost closed. Therefore, the pipe geometry has a distorting effect upon the valve characteristics. $\mathbf{C_{VC} = F_p \times C_V}$ states that the corrected C_V equals the geometry factor times the C_V. $F_p = 1$ when the valve is the same as the pipe size.

The formula used to correct C_V is $F_p \times C_V = C_{VC}$.

$$\mathbf{F_p = 1 / \sqrt{\{(1 + [\,1.5\,\{1 - (d^2 / D^2)\}\,] / 890)\ (C_V / d^2)\ 2\}}}$$

C_V is the rated sizing coefficient without reducers.
d = Nominal valve size in inches
D = inside diameter of the pipe in inches
C_{VC} is the corrected C_V
It is a theoretical approximation and based on average concentric reducer tests.

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Figures 13 and 14 show examples of curves which are modified by pipe geometry. The curves of low capacity valves (globes and characterized balls) are pushed down only a small amount. They are pushed down very little at the top position only. The curve improves very little. The higher the capacity of the valve the more the curve is distorted at the open positions.

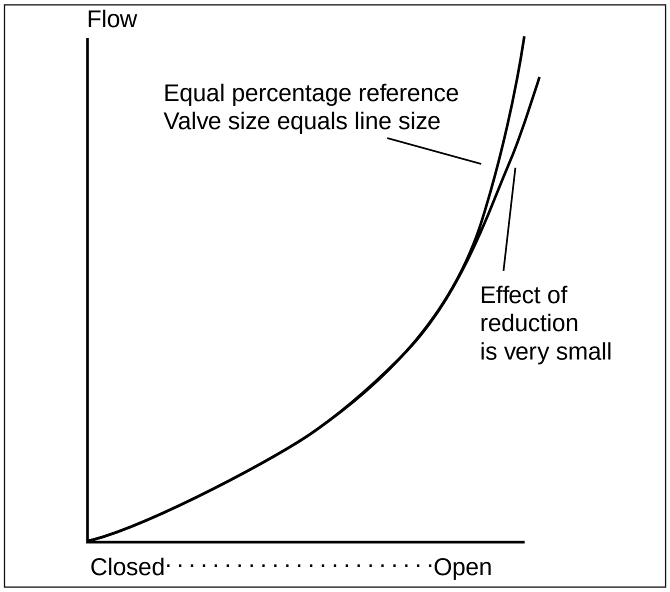

Figure 13 - Globe and characterized ball valve response curves due to geometry

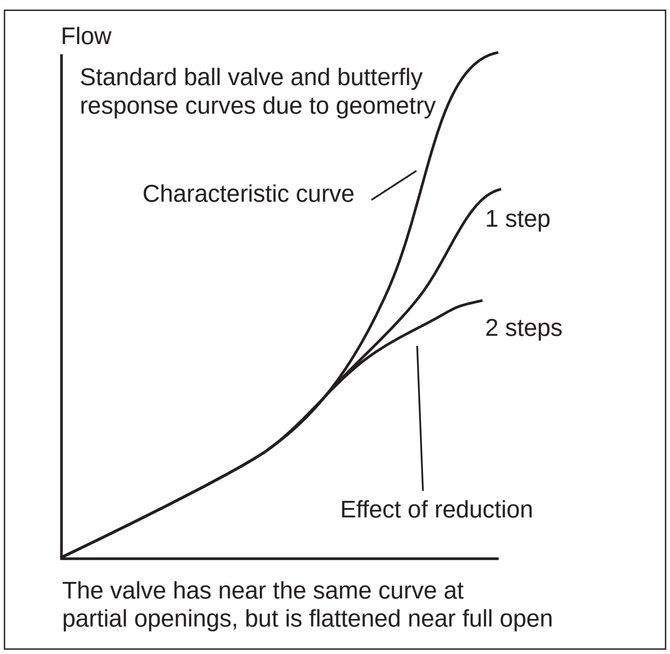

Figure 14 - Standard ball and butterfly response curves and effect of reduction

In *Figure 14*, note that the same coil output occurs when the valve is only half open. The reduction in Cv is inconsequential. The portion of the curve in which the system operates is unaffected by the reduction. The curve is not improved. The Belimo Characterized Control Ball Valve does not have this problem.

Resolution
Resolution is the number of positions an actuator will assume when the control signal is slowly changed 0 – 100%. You could also call it "positioning accuracy."

Rangeability Factor
Globe valves serve as a good example to explain valve authority. A globe valve has a contoured plug and a disk that operates against a seat. The largest diameter of the plug must be slightly smaller than the inside diameter of the seat. Otherwise the plug will get stuck in the seat. The clearance causes a minimum flow. Therefore, the contoured plug only can control the flow from a maximum flow down to a minimum flow. This is the "minimum controllable flow." When the plug is closed any further, the disk will stop the flow abruptly. The ratio between the full flow and the minimum controllable flow is the rangeability factor.

$$\text{Rangeability Factor RF} = \frac{\text{Maximum Flow}}{\text{Minimum Controllable Flow}}$$

It is important to realize that the rangeability factor is measured under laboratory conditions, with a constant differential pressure applied across the valve.

RANGEABILITY IS A CHARACTERISTIC OF THE VALVE ITSELF. It depends upon its design and manufacturing tolerances.

Turndown Ratio
The turndown ratio relates to installed valves only. It is the ratio between the maximum flow to the minimum controllable flow of a valve that is installed in a system.

$$\text{Turndown Ratio TR} = \frac{\text{Maximum Flow (installed)}}{\text{Minimum Controllable Flow}}$$

Example: In a laboratory, a constant differential pressure of 10 PSI is applied across the valve. We find that the maximum flow is 100 GPM, and that the minimum controllable flow is 2 GPM. Thus, the rangeability factory is RF = 100/2 = 50:1.

If we install this valve in series with a coil and apply 10 PSI across the valve and coil combination, we will find that the pressure drop across the coil reduces the pressure across the valve and thus the maximum flow. However, when the valve is almost closed, the pressure drop across the coil is zero and the full 10 PSI is applied across the valve. Therefore, the minimum controllable flow (2 GPM) will be the same as when there was no coil in series.

Let us say that the coil reduces the maximum flow to 70 GPM. The minimum controllable flow still is 2 GPM.

$$\text{TR} = \frac{70}{2} = 35:1$$

The turndown ratio is always smaller that the rangeability factor. Valve authority is covered in the next chapter. It is related to the turndown ratio. TR = RF x $\sqrt{A}$.

Many globe valves with commercial pneumatic actuation jump open. They seat with low leakage (typically .5% of C_v when new and 1% of C_v as the system ages) but they do not open smoothly. The minimum flow is .5% of C_v and then as soon as the pneumatic actuator moves at all they get 25% of flow. Control at low openings is impossible. 2 position control results at low loads. First off, then 25% open, then off, then 25% open, etc. is all that is possible.

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The actuator affects turndown. A Belimo actuator with a high accuracy has a 200:1 possible rangeability (8V span / .04V response). Positioning accuracy greater than 1% is unnecessary. A productive actuation is normally 2-3% of flow. But the Belimo can match the DDC and high valve accuracy which most electric and all commercial pneumatic actuators cannot do. Hunting and dithering are possible with too small increments. See section on control loop tuning.

Turndown is always less than the rangeability. Authority is related to turndown. Turndown ratio = Rangeability X √Authority. This is covered in the next section.
Given a rangeability = 30:1 and an authority of .5, the rangeability cannot be better than 30 x √.5 = 21 or 21:1. This is only 5% minimum flow at low loads.

The higher the turndown ratio is, the better the controllability will be.

6. VALVE AUTHORITY

When installed, the differential pressure across a control valve changes when it is operated between open and closed. When it is closed, the full differential pressure in the system acts upon the control valve alone. When the valve is fully open, the flow will cause a pressure drop in all the other parts of the system (coil, balancing valve, pipes etc.), so there will be less differential pressure across the control valve. Depending upon the system, variations in the pump head may add to the pressure variation.

The valve authority (A) is the ratio between the differential pressure across the fully open valve and the fully closed valve. Or:

A = Open Valve Pressure Drop/Closed Valve Pressure Drop

We can use A = 1 or A = 100% to say that the valve is the only pressure drop in a subcircuit. This is the inherent or intrinsic curve. As resistance is added in series with the valve the curve changes. Note that in a real installation the pressures in the supply and return vary with operation of other valves and that different types of valves have slightly different curve shapes.

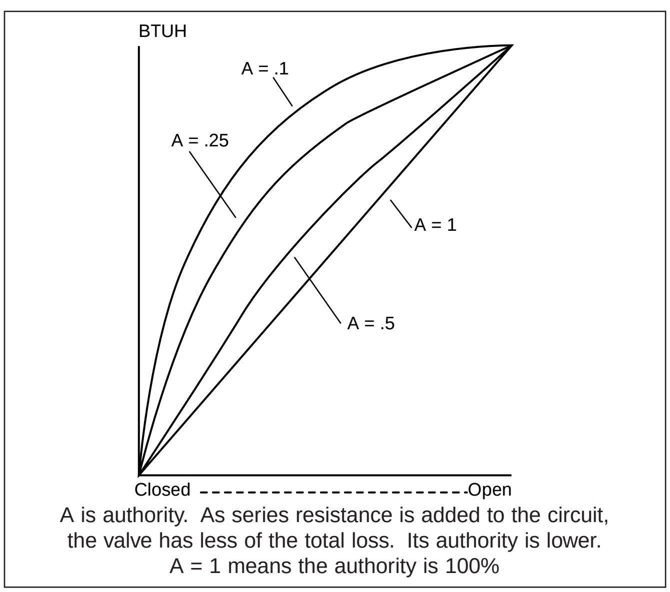

Figure 15 - Affect of valve authority on a linear valve

Figure 15 shows how the valve authority (A) affects a linear valve. Linear characteristic is not suitable for modulating control in water systems, but it serves as a good example to clearly show how the characteristic is changed as the valve authority changes.

Figure 16 shows how a low valve authority (A = .1 and A = .25) distorts an equal percent valve characteristic. When the valve authority is low, the smallest stem movement will result in a disproportionately large change in the heat output and stable control is difficult to accomplish.

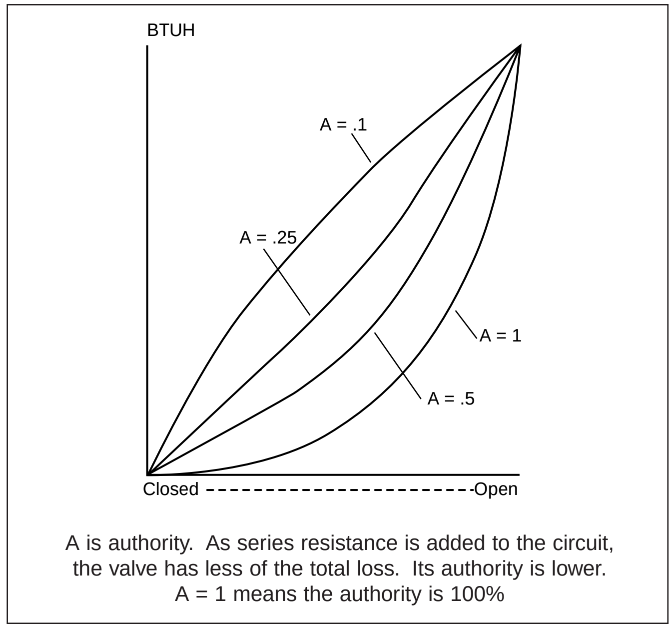

Figure 16 - Distortion of low valve authority on equal percent valve characteristics

Figure 17 shows an example of a valve with almost 100% authority while *Figure 18* shows a valve in a typical coil application with A = .4.

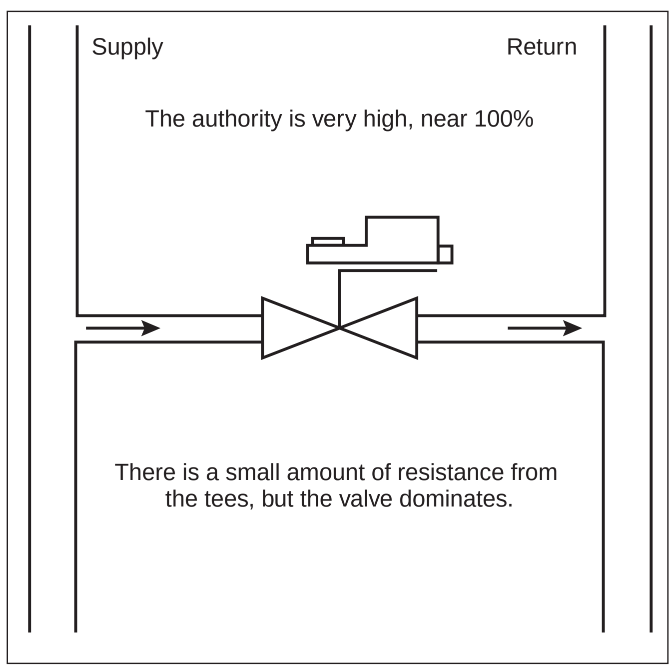

Figure 17 - High authority valve

The pressure variation across a control valve is very much dependent upon the type of system. The differential pressure gradually increases as the valve is operated from open to closed. This will distort the valve characteristics. The variation is very large when a single constant speed pump is used, and it results in a poor valve authority. Pressure control and variable speed pumping will improve the valve authority as such, and thereby they will help with controllability.

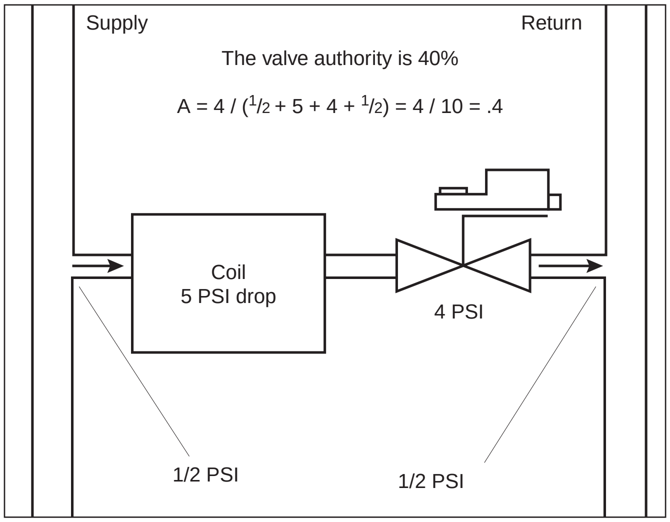

Figure 18 - Authority = .4

7. CONTROLLABILTY

The best control stability is accomplished when there is a linear relationship between the position of the valve stem and the heat output from the coil. Almost all control systems default to a linear signal, actuator, and loop tuning scenario.

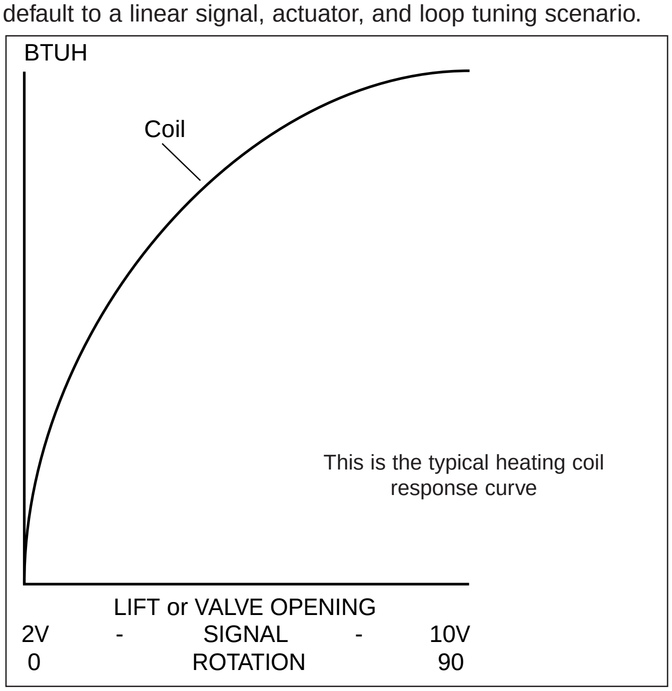

Figure 19 - Typical coil characteristic

Figure 19 shows the typical coil characteristic. *Figure 20* shows the deep equal percentage valve characteristic at A = 1. *Figure 21* shows the resulting relationship between the position of the valve stem and the heat output from the coil.

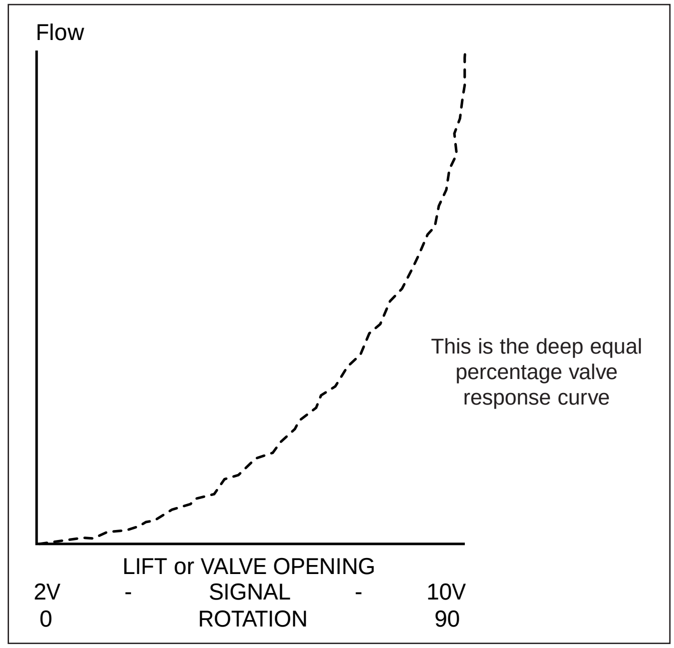

Figure 20 - Deep equal percentage valve response curve

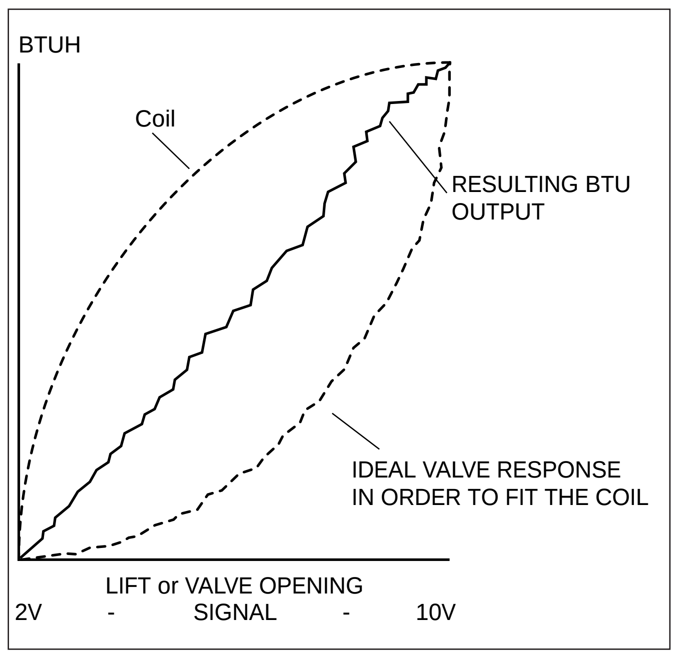

Figure 21 - Resulting BTU output

The coil characteristics is “convex” and an equal percent valve characteristics is “concave”. The two complement and counteract each other, so that the relationship between the control signal and the heat output is essentially linear. **This is only true as long as the valve characteristic is not distorted by a poor valve authority. That is, not distorted by too low an authority.**

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It is obvious that in order to get a stable control, a high valve authority (A) is desirable. This is demonstrated in *Figure 22.*

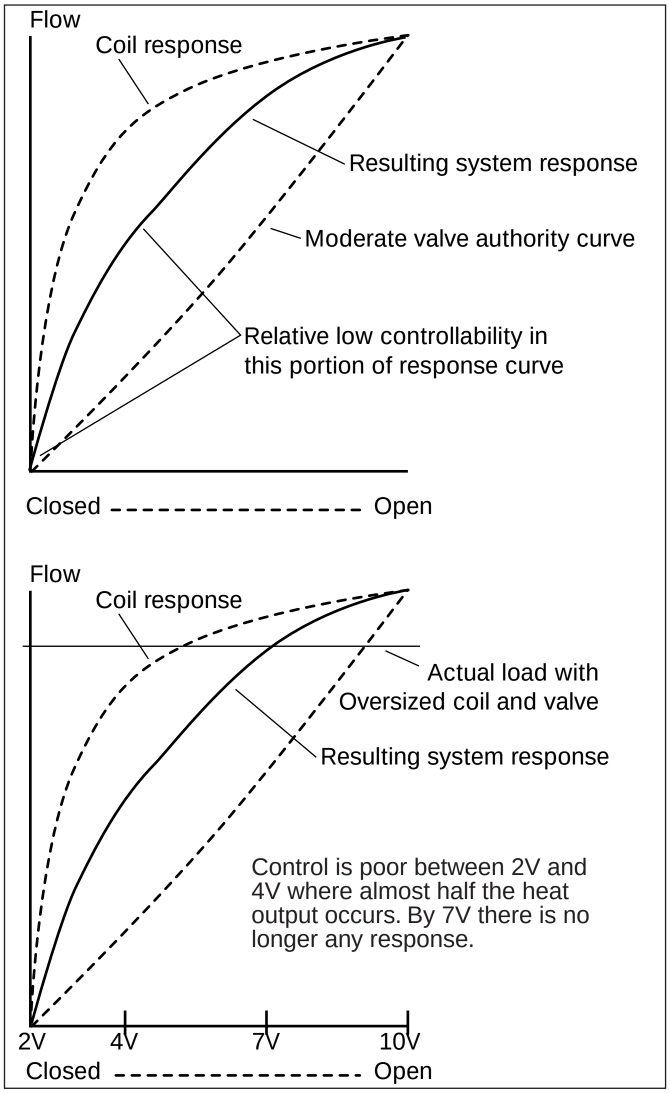

Figure 22 - Coil response

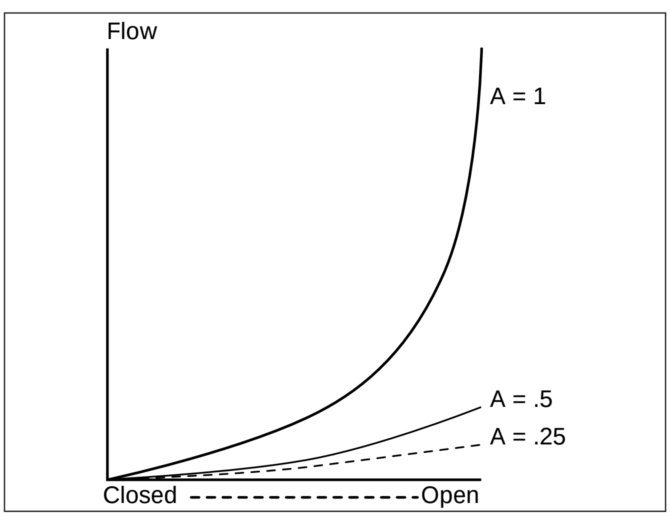

Figure 23 - Absolute value of flow

Figure 23 shows the authority curves in a different way than all the other drawings. The absolute value of the flow is shown for each curve. In all the other drawings the flow quantity is not adjusted since it is the linearity of the curve that is discussed.

Adding series resistance lowers the full open flow more than it lowers the midpoint flow. For this reason the curve changes gradually from concave to convex.

Actuator characterization

An actuator could be characterized also. Instead of the rotation being proportional to the signal it could follow a logarithmic curve. During the first volt of signal increase the actuator could move 3% of rotation; increasing in larger steps the last volt of signal increase could result in 25% of rotation.

Better is to tune the control loop to achieve this as necessary.

Reset of supply water temperature

The supply water temperature to the heating or cooling system is changed with respect to a parameter that is proportional to the load, usually the outdoor temperature. The reset function improves control when an oversized valve has been installed. When the temperature is reduced, the flow has to be increased in order to supply the same heating or cooling. Therefore, the control valves have to operate at a more open position. The reset schedule is the relationship between the outdoor temperature and the supply water temperature.

Example; the system has oversized coils, and the reset schedule is adjusted so the supply water temperature is 200F at the design outdoor temperature. This has the result that the control valves only open 50%. They operate on the steep part of the combined valve and coil response curve. If the reset schedule is changed, so the supply water temperature is lowered, then the flow has to be increased in order to meet the load. Therefore, the control valves have to operate at a more open position. This improves control.

Oversized heating coils will benefit from a reduced supply water temperature. For example a coil that is 25% oversized at 200F supply water temperature, will be right when the supply water temperature is reduced to about 190F.

The increased flow through the coils reduces the risk of freezing of heating coils, and the surface temperature of the coils is more even. The drawback is that the pumping cost increases.

There are limits on how the reset function can be used. It is the coil that requires the highest temperature that determines the reset schedule. Different coils serve different loads. Some coils may be very much oversized, while others are just right. All loads are not related to the outdoor temperature, for example interior zones. In cooling systems the reset is limited so the water is cold enough to provide dehumidification.

High performance System

The speed of the pump is controlled with respect to the position of the actuators, so one of the valves has to be almost 100% open to satisfy the load at this terminal. The reset and “high performance” functions have the effect that the control valves will open more and stay away from the near closed region where stability is a problem.

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Sequenced valves
Instead of using one valve, two equal percent valves can be piped in parallel and operated in sequence. Valve #2 should have twice a large C_V-value as valve #1. Several sequences of operation are possible. Valve #1 is modulated open while valve #2 is closed. Then valve #2 modulates open. Valve #1 could be closed as #2 opens only to reopen later when demand exists. One or 2 analog outputs could be used depending on the accuracy required. The combination of valves will have a very high rangeability. It is an inexpensive solution in many cases requiring large valves. Two smaller ones may be installed.

This solution is commonly used in steam systems, but has been used for hot water and air dampers also. It is essentially multistage control.

8. CONTROLLABILITY VS. PUMPING COSTS

To accomplish a high valve authority size the control valves for a high differential pressure. In a system with two-way valves, a valve authority of A=0.5 is needed. When the pressure drop across the control valves is as large as the total pressure drop of all the other parts (pipes, coils, balancing valves etc.) in the loop, the valve authority A=0.5 is relatively high. The distortion of the valve characteristic is quite small.

The controllability will be excellent, but at a cost. A pump must be selected, that can produce a sufficiently high pump head. The high pump head is energy consuming and results in quite high pumping costs.

In order to reduce the pumping costs, a lower valve authority must be accepted. A compromise should be found, that still gives an acceptable controllability. Modern controllers have advanced control algorithms, and can give a better control than older types of controllers. If we can accomplish stable control although the valve authority is not very high, then the pumping costs can be reduced.

Different types of hydronic systems have different characteristics with respect to the controllability, so the type of system is an important consideration when the pressure drop across the valves is determined. If a sophisticated DDC system is being used it may be possible to linearize the output using the control algorithm.

Eliminate the excessive pump head
Pipe sizing can be restrictive. The tradeoff between first cost and life cycle costs is always present, but the piping system is most difficult to replace in the future when reconstruction takes place.

The payback of trimming the impeller (or replacing the pump) is very good, because of the savings in the annual operating costs of the pump. Furthermore, it is wrong to use the main balancing valve to eliminate excessive pump head, because it is effective only at the maximum total flow. At reduced flow the main balancing valve will have almost no effect and the full pump head will act upon the control valves. The control valves will be subject to very large pressure variations, and the authority of all the control valves is low. The valve authority and controllability of the whole system is improved, when the excessive pump head is eliminated.

Some equipment remarks
The system curve
The system curve shows how the required pump head varies with the total flow in a system when all the control valves are open. It is a quadratic relationship between the flow and the pump head (differential pressure), so if one point of the system curve is known, the rest can be calculated. Calculate the C_V value of the known point. This is the C_V value of the whole system. Apply different flow values to this C_V value and calculate the corresponding differential pressure using the formula $Q = C_V \sqrt{\Delta P}$.

The pump curve
The pump curve shows how the pump head varies with the flow. Usually pump manufacture provides charts which show a number of pump curves at different pump speeds (RPM) or impeller diameters. The efficiency (%) is usually shown, so the best operating point can be determined.

The operating point
By plotting the system curve on the pump curve chart, the intersection point between the system and pump curves can be found. This is the “operating point”, which is the flow and pump head of the system. The “operating point” usually refers to fully open control valves in the system. When the control valves begin to close and reduce the flow, the operating point moves back along the pump curve. See *Figure 30.* When the speed or the diameter of the impeller is reduced, the operating point moves down along the system curve.

Constant flow systems are sized for an operating point at the point of highest efficiency. Variable flow systems are sized for the flat part of the pump curve.

The expansion tank absorbs the volume changes due to temperature variations of the water in the system. It also maintains the pressurization of the system, so cavitation is prevented. Some tanks have a pressurized bladder inside. The bladder should be sufficiently charged, so the correct pressurization is accomplished. The expansion tank and automatic fill valve should be connected close to the suction side of the pump. See *Figure 26.*

Pump location
The pump should be located just after the outlet of the boiler. Between the outlet of the boiler and the pump an automatic air separator should be located. The water leaving the boiler is very warm and the pressure is low at the suction side of the pump. The operating conditions for the air separator are therefore very favorable, because dissolved oxygen is released when the temperature is high and the pressure is low.

Expansion tank and fill valve
The expansion tank will maintain a constant pressure at the suction side of the pump. Therefore, the automatic fill valve is subject to a constant pressure, regardless of the operating conditions of the pump. Water will be added only if the pressure of the expansion tank drops, which indicates that there actually has been a loss of water in the system.

It is important that the automatic fill valve does not add water when it is not needed, because it will overfill the system. The relief valve will eventually remove the excess water. However, a cycle where the auto flow valve adds water and the relief

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valve removes it, must be avoided because oxygen rich water is repeatedly added to the system and corrosion will occur. Actually if the automatic fill valve is closed, for a period of time, leaks can be detected if the pressure keeps on dropping.

9. BALANCING

Balancing valves are essential. If there is a disturbance during normal operation and a control valve opens, then there can be a tremendous overflow (short circuit) at this terminal, if there is no balancing valve. The system pressure drops, and all the control valves begin to open and compete for the flow. Without balancing, a disturbance in one part of the system will spread to the whole system, and it will take some time before stability is restored. See *Figure 24* for locations of balancing valves in typical systems.

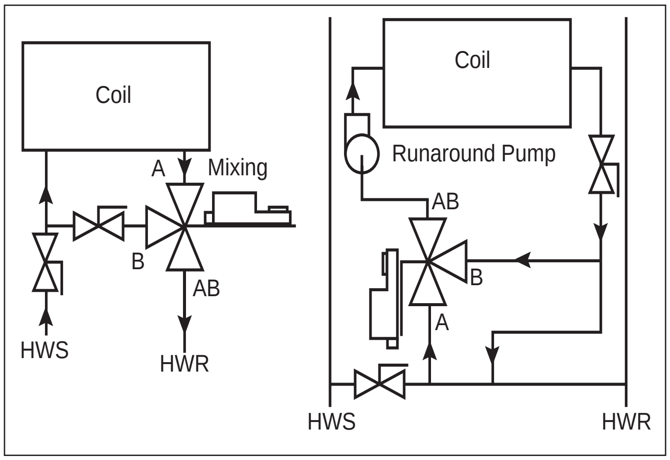

Figure 24 - Balancing valve locations

Most balancing valves also serve the purpose as shut-off (service or isolation) valves, so the additional cost is quite small. Actually, the balancing valves make it possible to confidently size the pump and control valves without any excessive safety margins. They can thereby represent a net savings in the installation cost.

The balancing valves are an essential diagnostic tool, and if the question should arise, can demonstrate that control valves with a reduced size actually provide the required flow.

Manual balancing valves can be adjusted to any position between open and closed. The valve body has two pressure ports so the differential pressure can be measured. The valve stem has a graduated scale so the stem position can be determined. A chart correlates the differential pressure and the stem position so the flow can be ascertained. The valve is adjusted so the desired flow is accomplished when the associated control valve is fully open. This is not so easily done, because there is an interaction between the balancing valves in a system. However, there are special balancing procedures that makes it possible — "the proportional method". Please refer to the manuals published by the manufacturers of balancing valves.

Automatic balancing valves are piped in series with the control valves, and automatically limit the flow to the desired maximum value. (Actually they are automatic flow limiting valves, but are usually referred to as "Automatic or Self Adjusting Balancing Valves".) Some types of automatic balancing valves have a fixed set point. Different inserts are available and can be interchanged to get the desired flow. Some valves have an adjustable set point.

Most automatic balancing valves can not provide shutoff. Therefore an additional shut off valve is needed. Automatic balancing valves do not require any special balancing procedure. However, the correct flow needs to be selected, adjusted, and verified.

The first step in balancing is to verify the required maximum flow through each unit (coil).

The second step, is to size the control valves correctly. Ideally, the control valve should be sized so it absorbs the majority of the available pressure at each terminal. The balancing valve absorbs the remaining pressure only, which should be relatively small, if the control valve is sized correctly. Unfortunately, it is hard for the contractor to size the control valves, because the needed information about the differential pressure in all parts of the system is usually not available.

If in order to get the desired flow, a balancing valve has to be adjusted to a near closed position, then it is obvious that either the control valve or balancing valve is too large. A balancing valve that needs to be adjusted for a high pressure drop, indicates that the control valve it serves is too large. It is the control valve that should have the large differential pressure, not the balancing valve. Also the balancing valves need to be sized correctly. If a balancing valve has to be adjusted to an almost closed position to get the correct flow, then it is too large. Oversized balancing valves are very hard to adjust and the accuracy is poor near the closed position. When small balancing valves are adjusted almost closed there is a risk that they will clog up with dirt. Use a finer strainer, and if possible, reduce the pump head, so both the control and balancing valves have to be more open.

Why balance?

Without balancing, the circuits with a low resistance will "short circuit" the loop, so the other circuits will not receive enough flow. This is a common problem during start-up in the morning. Some parts of the building receive a large flow, while other parts are deprived and lag behind. Therefore, it will take a needlessly long time to recover the normal operating temperature. Also, if there is a disturbance during normal operation and one control valve opens fully, the adjacent control valves are disturbed, and it will take some time before the system settles.

A prolonged overflow condition may cause erosion damage to the coils and piping. Without balancing, the total flow will be needlessly large.

Oversized pumps cause needlessly large pressure variations in the system, which is detrimental to the operation of the control valves.

Unbalanced systems have overflows, which means that the system operates with a low temperature differential between supply and return. This reduces the efficiency of chillers, high efficiency boilers and district heating or cooling systems. The balancing valves can also be used for diagnostic purposes.

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In a system using manual balancing valves, the balancing should be done so the balancing valve in the most resistive (remote) loop is adjusted for a low pressure drop (about 1 psi).

The last step in the balancing procedure is to adjust the main balancing valve (piped in series with the pump) so the total flow is correct. If the main balancing valve needs to be so much closed that there is a high pressure drop across it, then the pump head must be reduced. As the system ages it can be rebalanced and the main valve opened completely if necessary.

The pump head should be reduced, so much that only about 1 psi remains at the main balancing valve (just enough to measure the flow accurately). Any more is a waste. Variable speed pumping reduces the need for building up pressure and then destroying it with a valve. It also reduces life cycle cost considerably.

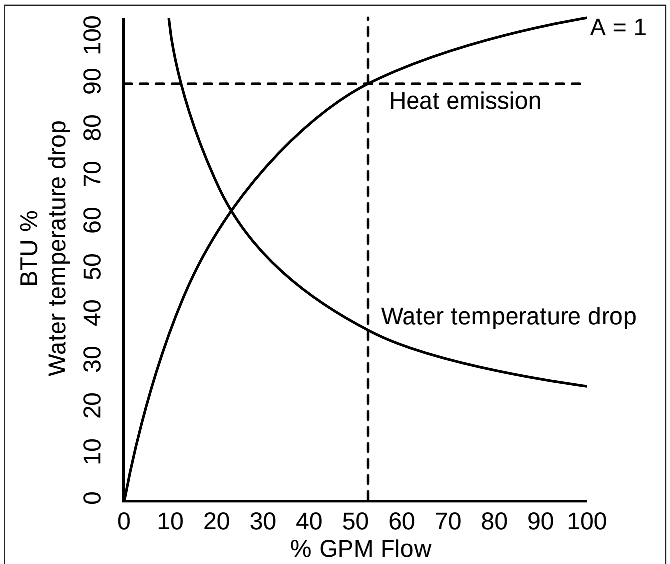
When automatic balancing valves are used, and if the pump head is excessive, the automatic balancing valves will operate well above the low end of their operating range. The pump head should be reduced so at least one of the automatic balancing valves operates at the low end of its operating range. (This is about 2 psi if the operating range is 2 - 32 psi.)

When the calculated load falls between two coil sizes, it is natural that the next larger size is selected. Therefore, most coils are more or less oversized for the application. The needed maximum flow is very much dependent upon the installed coil and the operating conditions. Therefore, the coil manufacturer should be consulted to determine the true maximum flow requirements. To do this charts or software are available.

Example
The specification calls for a 90,000 BTUH load and a coil with a 20 F design temperature drop. (BTUH = BTU/hour. LBSH = LBS/hour. GPH = gallons/hour, 1 Gallon = 8.333 LBS)

1 BTU raises the temperature of 1 LB water 1F. Therefore, 90,000 BTUH/20F = 4,500 LBSH/8.33 = 540 GPH/60 = 9 GPM. This is the specified flow. The nearest standard coil, has a design temperature drop of 20 F at a 100,000 BTUH load. This coil is installed, but it means that we are going to use only 90,000/100,000 = 90% of the capacity of the installed coil. See *Figure 25* and the 90% BTUH line. It corresponds to a 36F temperature drop. Let us apply that to the specified load. 90,000/36F = 2,500 LBSH/8.33 = 300 GPH/60 = 5 GPM. This is the actual flow. The actual flow is 5/9 = 56% of the specified flow. As can be seen, a 10% oversized coil results in water flow reduction to 56%.

Balancing is thus seen as very important. Note that if the balancing valve is used to reduce pressure the authority of the control valve decreases. It would be better to carefully analyze the entire system initially. The control valves may be oversized when balancing reduces pressure a great deal.
If deferred maintenance of the system is expected, then the efficiency of the coil may decrease due to dirt on the air side and the resulting lower air flow. Some leeway should exist.



Typical coil heat emission vs. water flow and water temperature drop vs. water flow

Figure 25 - BTU% v. %GPM flow

It is very important that the flow calculation is based upon the actual operating conditions, otherwise the pump, pipes, control and balancing valves will be sized for a needlessly large flow. This is a worthwhile effort, because the difference can be very large. The potential savings can be large, and most importantly, the operation of the system will be much improved.

Differential Pressure
Information on the pump head is usually available in most specifications. This is a very valuable data, because it has been calculated to overcome all the different pressure losses in the system. In other words, the specified pump head tells us what the resistance of the system is.

The following text describes a method, where the pressure drop across two-way valves is calculated as a percentage of the pump head. The percentage number is dependent upon the type of system, and is chosen so a good compromise between controllability and required pump head is found.

There is nothing “scientific” about the percentage numbers in the following text. Higher numbers could be used, as long the flow requirements can be satisfied.

10. Selecting the pressure drop of the valve

On-Off Control.
See *Figure 26.*
The valve sizing when on-off control is used is very simple. Valves with the same size as the pipe are normally used. The valve characteristics is not an issue, but valves with a large C_V-value compared to the size is preferred. Balancing valves are important. The pressure variations can be very large depending upon how many valves are open at the same time. Therefore, automatic balancing valves are preferred

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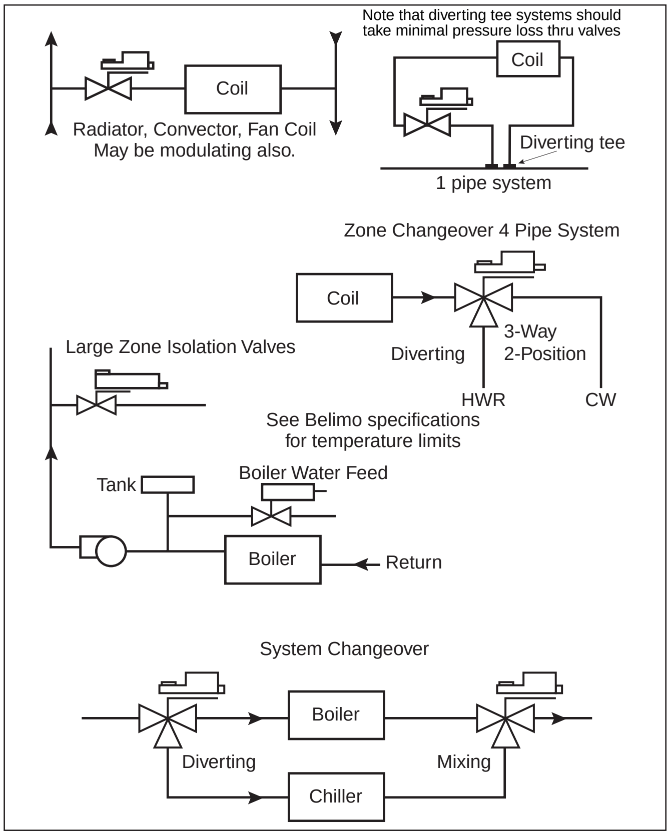

Figure 26 - Two and three-way valves in 2-position application

Sometimes a valve that is one size reduction in pipe can be used. 10% of the pump head is the maximum loss allowed for sizing the C_v. Another rule sometimes used is to allow no more than a 1 psi loss through the valve. Too few valves used with 2-position control can cause pump pressure problems. See the next section on diversity.

Diversity

Diversity refers to the variation in load on a system. There may be a number of valves all of which are never full open or closed. Some loads may be at 100% while others are at 25%. The closing of one valve has little effect on the pump head and system curve. With no diversity, say only 2 valves, operation of one has a major effect on the pump and system curve. A large chiller system could have problems with surging (unloading the chiller abruptly with accompanying vibration) with no diversity.

With low diversity the closing of a valve leads to an increase in pressure drop across it. At a low amount of opening the pressure increase causes the flow to stay high.

Authority variation due to pressure variation

When looking at any piping, coil, and valve it should be observed that the authority of the valve is usually constant. However, the authority concept depends on relatively constant pressure drop as the valve closes. Some increase is not important. If diversity does not exist, closing the valve may destroy the normal curve since pressure increases.

Constant flow system

See Figure 27.

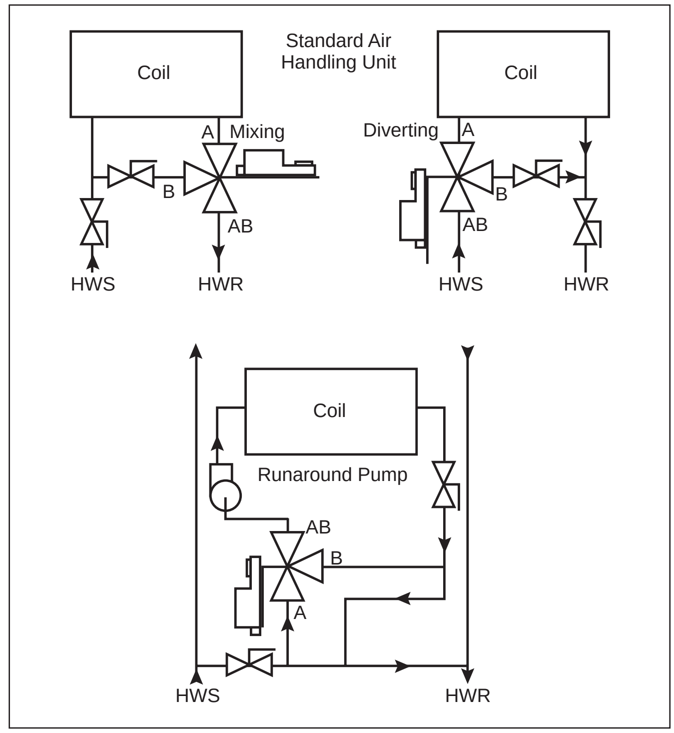

Figure 27 - Constant flow systems and three-way valves at the terminals

Constant flow systems use three-way valves at the terminals. The valves should be sized for slightly more than the pressure drop of the coil. If this information is not available, use 4 psi. The valve is sized for a C_v which produces this drop at the specified flow rate using C_v = GPM / √ P. The drop could go as high as 9 psi if necessary. Since the square root of the pressure drop is used, the range from 4 to 9 psi only gives a multiplier of from 2 to 3 against the C_v.

Without hard data from the design engineer it is difficult to state what this pressure loss is. Coils are manufactured with less than 1 psi drop (2.3') to as much as 6 psi (14') at the velocities possible. Although it is not always true, reheat coils tend to have low drops and air handling units tend to have higher drops. Especially cooling coils.

The goal is to get the authority near 50% or a little more to mechanically linearize the process. The coil characteristic response is quite similar to that of the variable flow system – it is significantly over the linear.

Valves should be equal percentage.

The lower drawing in *Figure 27* shows a coil with a secondary circulating pump. It provides many important advantages. It provides a constant flow through the coil. The flow is always turbulent. The risk of freezing is reduced. The temperature over the surface of the coil is more even, so the leaving air temperature is more uniform. It is possible to adjust the secondary flow, so the capacity of oversized coils is reduced. The pressure drop across the coil is taken care of by the secondary pump. Therefore, the head of the main pump can be

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reduced. Dirt will not collect inside the tubes. The heat transfer curve with a secondary circulating pump is more linear. If the coil is oversized, it is possible to increase the secondary flow and thereby decrease the water temperature drop across the coil. This reduces the capacity of the coil, and the primary flow will be increased.

Variable flow - one pump

See *Figures 28a & b.* Variable flow systems use two-way valves at the terminals. One main pump serves the whole system. It operates at a constant speed. The pump head varies significantly with the load. As any valve closes, the head goes up riding on the pump curve. The control valves should be sized to be 50% of the loss in the circuit. This could be 50% of the specified pump head assuming the pump is sized correctly.

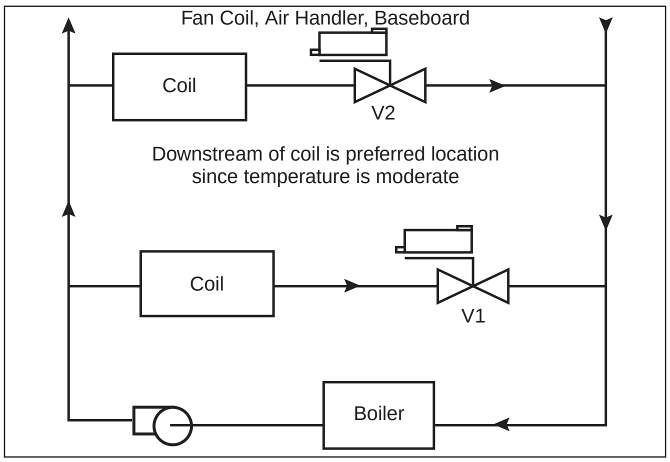

Figure 28a - Variable flow systems and two-way valves at terminals

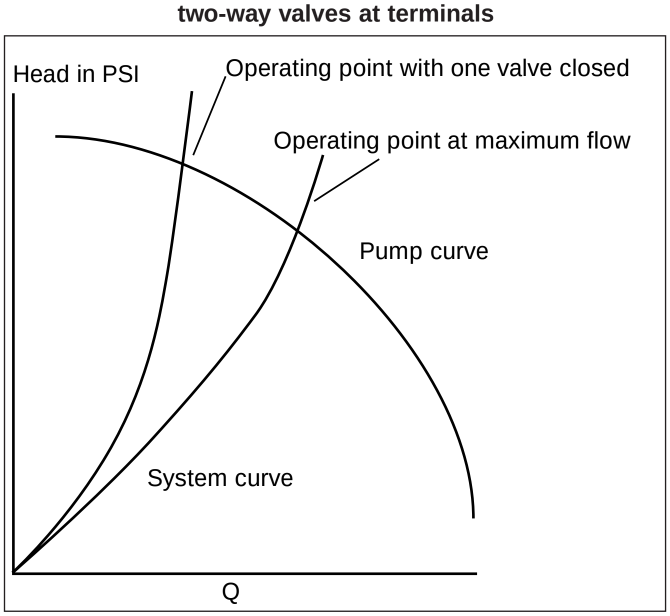

Figure 28b - Variable flow systems pump curve

The default rule of thumb is to use 4 psi to 9 psi for the valve sizing. No less than this will produce authority that will control well.

Equal percentage valves should be used.

The differences in coil response curve could be taken into consideration. The far over the linear 20° heating coil design drop will respond differently than the shallow over the linear 10° cooling coil design drop. A ball or butterfly has a shallow equal percentage curve when the entire range of 90° is not used. These would match against the shallow coil curve.

In any event, the tendency to oversize the valve indicates that a deep equal percentage valve curve is best. Not only does it match the coil curve, but when first opening the resolution is much better. At the near full open areas the control is not as good and may not be used anyway. At near closed when the system pressure may rise, the curve has good resolution.

There must be diversity in the system to keep the pressure from rising too high, otherwise a bypass must be used.

Variable volume or flow
Staged pumps, variable speed pumping or primary - secondary pumping

This reduces the variation in the pump head between maximum and minimum flow. The valves should be sized as stated above. As long as the authority of the valve remains constant, the reduced flow will be inconsequential.

When an air handling unit is shut off, do not open the control valve fully. Instead, open it no more than what is necessary to provide a desired minimum flow. Otherwise the variable speed system will run at maximum speed when the air handling units are off and energy is wasted.

Variable flow system - "high performance system"

See *Figure 29.* Except that the differential pressure sensor is removed it is similar to the above system, but instead of using a differential pressure controller, the speed of the pump is controlled with respect to the position of the actuators.

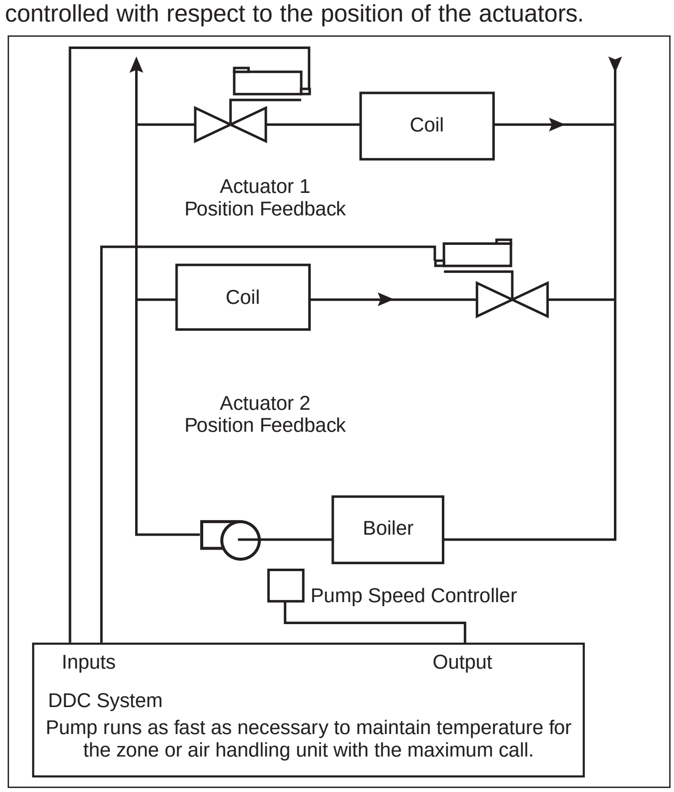

Figure 29 - Variable flow high performance system

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The speed of the pump is controlled (reduced) so at least one of the valves has to be almost 100% open to satisfy the load.

Because the control valves will operate nearly fully open, the stability will be very good, even if the valves are oversized. However, "high performance" systems can sometimes be problematic, and it is possible that it is necessary to change to differential pressure control. (It is prudent to install nipples for the pressure sensors regardless if they are used initially or not.)

"High performance" systems require an advanced DDC installation, that can analyze and quickly communicate the load signals. (Warning, one anomalous load signal may upset the whole system.) Do not let the valve open fully, when an air handing unit is shut off. Otherwise the variable speed system will run at maximum speed. Energy is wasted, and the pump head will be very high, so the valves will probably hunt. The valve should be no more open than what is necessary to provide a desired minimum flow.

Selecting the valve
Calculate the C_V value using the formula $C_V = GPM / \sqrt{\Delta P}$ and apply the relevant flow (GPM) and differential pressure (ΔP). Control valves are available with a limited number of C_V values only. Chances are that the calculated value falls in between two standard C_V values.

Select cooling valves for the next <u>higher</u> C_V. The cooling temperature drop is typically only 12° to 8° which makes them quite sensitive to undersizing.

If you have to use the originally specified flow, select a heating valve with the next <u>smaller</u> C_V-value. When you can use a corrected flow that reflects the actual conditions, select a valve with the next <u>higher</u> C_V-value.
Section 11 goes into details about decreasing the valve size and shows that there is little danger when applying heating valves.

This assumes that specified data is oversized but that corrected analysis gives correct data.

In general when estimating modulating valves at terminal units and air handling units will use valves one size smaller than the pipe. Two position applications and some special applications use line size valves.

When the C_{VC}'s of available valves do not meet the requirements there is one simple way to solve the problem. An adjustable start and span actuator or a limit on signal output to the actuator is needed.

Charts for C_{VC} at rotated positions for the ball and butterfly valves appear in the last section of this manual. Globes do not change significantly.

The 90% open ball or butterfly has about the same C_{VC} as a one size reduction in pipe size. The 80% open ball or butterfly has about the same C_{VC} as a two size reduction in pipe size. A 90% open butterfly reduced 1 pipe size has about the same C_{VC} as an 80% open valve. Combinations are possible.

Limiting the full open rotation to avoid cavitation which would result from too severe a reduction in valve size is a common practice. These charts allow finding the size valve to use.

THE GOLDEN RULE IS NEVER REDUCE THE VALVE TO BELOW 1/2 THE LINE SIZE. When reducing, use pipe and/or valve supports since reducers weaken the structural strength of the assembly. Do not risk mechanical integrity.

Bypass applications
There are 2 common bypass applications shown in *Figures 30a and b*. These must each be analyzed individually to decide on how to size the valve. For example, the system bypass. Its purpose is to keep the pump differential from rising too high. A limit is selected, typically 10-20% above the operating point pressure. Above this the flow is restricted and the pressures are deemed too high. When the differential pressure rises, the controller modulates the bypass valve open.

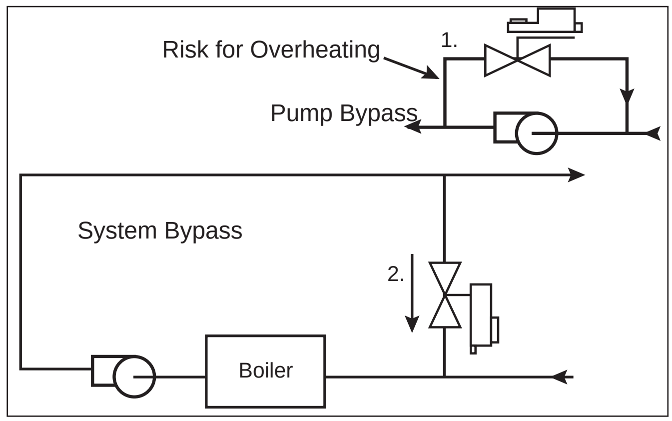

Figure 30a - Bypass applications

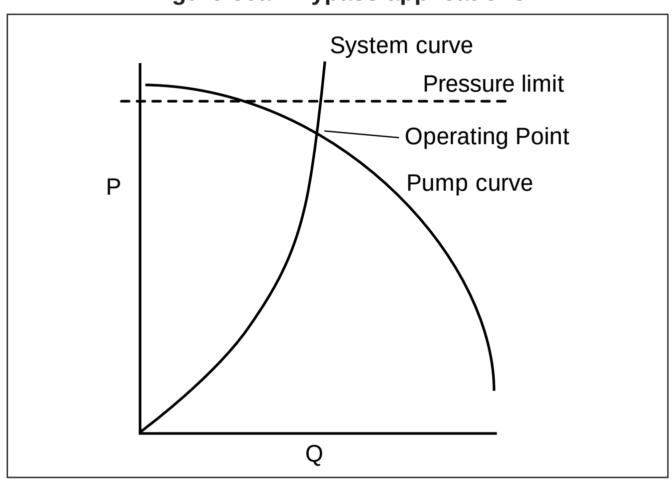

Figure 30b - Pressure limit and operating point

For example, let the setpoint be 20 psi. Let the pump capacity at design operating point be 200 GPM. And assume that the bypass must be able to pass 100 GPM of water since 100 GPM will always be flowing since some valves will always be open to some degree.

Then the valve is sized for 100 GPM, the difference, at a 20 psi differential. This is quite different from the requirements for air handling unit and reheat coils.

The design engineer should be consulted if the specification does not indicate what sizing parameters to use for a bypass valve.

Figure 31 shows the common 3-way valve applications which control temperature not flow. Sizing techniques are different in these applications. In the cooling tower application the valve will respond to the needs of the chiller's condenser by maintaining about 80 degree water. The valve should be as linear as possible. Linear globes are available, but butterfly valves are often used here and they are shallow equal percentage.

More important is that the outlet pressure to the spray nozzles (if used) be high enough. The inlet pressure and the nozzle and piping pressure losses must be known to size the valve. A balancing valve should be placed in the circuit going to the sump to equalize the pressure loss to that going to the spray.

This valve may not be sized without a known pressure drop. Failure to verify the allowable drop will lead to problems. Estimate one pipe size less for biding purposes, but do not engineer the valve without verification.

Figure 31 also shows a perimeter loop reset or boiler bypass which is another 3-way application controlling temperature. The flow is constant but the temperature is varied by mixing return with supply water. The valve is linear and is sized line size. A balancing valve should add the same drop as the boiler circuit adds if it is above a few psi.

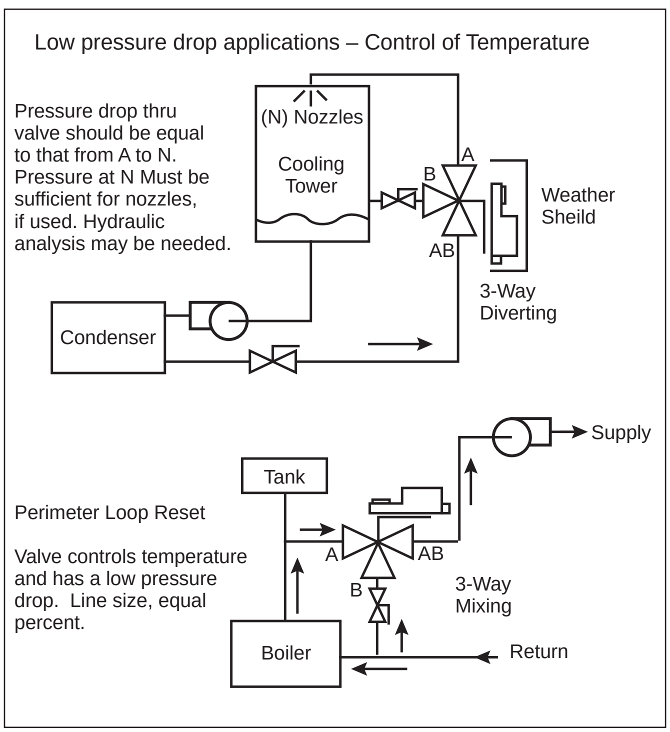

Figure 31 - Three-way valve applications controlling temperature

11. Consequences of increasing the differential pressure

If we size the control valves for a larger differential pressure, in accordance with the above rules, we will get smaller valves and an improved valve authority (A). This is of course good, but can we satisfy the maximum flow requirements? The consequences of an increased pressure drop over the control valves will depend upon the circumstances in each case.

Alternate 1 – Coils are sized correctly, not oversized, and actual flow is as specified

This is an unlikely example. It means that the installed coils are exactly as specified, without any oversizing at all. Nevertheless, this is a very interesting example, that highlights the inherent nature of hydronic systems.

Although the valves are sized for a higher than the originally specified pressure drop, all of the flow requirements will essentially be satisfied in most systems. This may sound strange at first. Instinctively, it would make sense that, if for example, the pressure drop for a valve is doubled, then the flow goes down to 50%. However, this is not the case.

First, because of the quadratic relation ship between flow and pressure, the C_V-value of a valve is reduced by just 29% and not 50% when the dimensioning pressure drop is doubled. Secondly, the control valves are not the only resistance to the flow in the whole loop. In comparison to the rest of the loop, the increase in resistance of the control valves is relatively small, so the effect upon the whole loop is minimal.
Thirdly, an increase in the total resistance moves the operating point back on the pump curve. This increases the pump head, counteracting some of the increased resistance, so the resulting flow reduction is about half of what it otherwise would have been. The system does have some self correction.

Lastly, it is only the few valves in the least favored circuits that are "at risk". The majority of the valves are located closer to the pump, and are subject to a higher differential pressure.

It is extremely unlikely that a valve ever needs to be replaced.

Example
For simplicity, the following example has only one coil, control valve, balancing valve, piping, boiler and pump. This of course is unrealistic, but nothing is gained by a complicated example with multiple loops and valves. The conclusion will be essentially the same.

System Operating Point Q=100 GPM,
Pressure Drop = 16 psi.

System C_V = GPM/$\sqrt{\Delta P}$ = 100/$\sqrt{16}$ = 25
Total "Original" valve. Q = 100 GPM, ΔP = 3 psi. C_V = 58
System Pressure Drop, minus control valve 16-3 = 13 psi.
System without control valve C_V = 100/$\sqrt{13}$ = 27.7
"Original" valve is replaced with a valve with Δp = 6 psi
"Replacement" valve Q = 100 GPM, ΔP = 6 psi. C_V = 40.8

New system C_V = $\left(\frac{1}{40.8}\right)^2 + \left(\frac{1}{27.7}\right)^2 = \left(\frac{1}{C_V 2}\right)^2$

C_V Total = 22.9

The new system C_V is 22.9, which is 91.9% of the original value (C_V=25), so it would seem that the flow will be 8.1% less than specified. However, the operating point backs up on the pump curve, and increases the pump head. With a typical pump, the new operating point will give a flow that is about 95% of the specified value.

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The increase in pressure drop across the control valve went from 3 to 6 psi, but the reduction in the flow is only 5%. When applied to the heat transfer curve of a coil, *(*see *Figure 25)* a 5% reduction of the flow will not result in a noticeable reduction in the heat emission.

This is just one example, but the increase in pressure drop must be quite large before any significant reduction in the heat output will take place.

Alternate 2 – Most coils are slightly oversized, but we stay with the originally specified flow for selecting the valves and pump.

As we know from Alternative #1, the valves can supply 95% of the specified flow, which is more than what the oversized coils will need. Actually the valves are probably larger than necessary.

Alternate 3 – The size of the valve is reduced to reflect the oversized coils, but the original specification for the pump is used.

The valves are correctly sized, but the pump is needlessly large. Therefore, flow requirement can easily be met. In most cases if not all, the valves are larger than necessary.

Alternate 4 – The size of the valve is reduced to reflect the oversized coils and the specification for the pump is corrected with a smaller pump.

Both the valves and the pump are correctly sized, so the flow requirement can be met.

It is possible for a valve to be six times too large using the traditional sizing methods

Example:
A coil with a 20F temperature drop is specified.
The load is 1,000,000 BTUH.
The calculated flow is 1,000,000/20F = 50,000 LBSH
50,000 LBSH/8.33 = 6,005 GPH
6,005 GPH/60 = 100 GPM
The pressure drop is 4 psi, so $C_V = 100 / \sqrt{4} = 50$.

Suppose that the installed coil is 20% too large. This means that the actual temperature drop is 50F instead of 20F. (See *Figure 25)*. Look at 80% BTUH. The actually needed flow at 50F temperature drop will be 1,000,000/50F = 20,000 LBSH/ 8.33 = 2,402 GPH/60 = 40 GPM.

This is 40% of the specified flow of 100 GPM.

At 40% flow the pipe losses is very small and the pump head is higher, so the differential pressure across the valve has increased to 25 psi. (See the diagram in *Figure 28*, and you will see that this is possible.)

The correct C_V value should be based upon 40 GPM and 25 psi.

$C_V = 40/\sqrt{25} = 8$.

$C_V = 50$ is 50/8 = 6.25 times larger than $C_V = 8$.
It is true. When the traditional sizing rules are used, the valves are often many times too large, so they can satisfy the maximum load already when they are 1/3 open.

Direct return or reverse return
The Figures here show direct return systems. The direct return system, has pathways of different length through the different loops (loop = pump - riser - coil - return - boiler - pump). The loops closest to the pump are “favored”, and the differential pressure across these terminals is larger than at the remote terminals. Proper balancing is therefore especially important.

Reverse return requires extra piping. The advantage is that the loops are of the same length. Contrary to popular belief, this does not mean that the piping network is “self balancing”. Without balancing, the first and last terminals will get an overflow, while the mid terminals will get less. Reverse return is an improvement over direct return, but balancing valves are still needed. Direct return with automatic balancing valves is a less expensive alternative. The piping is less, which is a savings in itself and because the pipe losses are smaller, a smaller pump can be used.

In addition, because of the lower pipe losses, it is easier to achieve a higher valve authority in direct return systems.

12. Determining the required valve pressure ratings

The rated close-off pressure for a combination of actuator, linkage and valve is the **differential** pressure it can close against. This pressure must be higher than the maximum differential pressure the valve will be subject to in the installation.

In a variable flow system (2-way valves) without pressure control, the close-off pressure is the maximum pressure the pump can produce. We do not subtract flow losses between the pump and the valve. When no water is flowing, the valve sees the full pump pressure.

In a constant flow system (3-way valves), the differential pressure across the valve is essentially constant. In order to get a safety margin, the close-off pressure is calculated 1.5 times the differential pressure. Since water is always flowing, the valve will never see the full pump pressure.

There are variable flow systems which use 3-way valves to provide minimum flow at the furthest terminals (10-15%). These three-way valves are subject to a variable differential pressure. The maximum differential pressure is the pump head minus the pressure drop across the balancing valve. These valves see more of the pump pressure as the 2-way valves close down.

Valves have a number of pressure conditions which must be considered. **They have 2 ratings — 1) body and stem static rating, and 2) disc and seat close-off rating.**

Static rating is the amount of total pressure which the body and stem seal must hold against without leaking.

Close off is the differential across the disc against which the valve can hold without leaking.

Steam valves should be selected for the boiler rating. For

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example, if an 18 psi boiler pressure is maintained, they the valve must hold against 18 psi (and 260°F).

Water valves require a bit more consideration. In *Figure 10* there are several conditions which must be evaluated to determine the pressure ratings required of the valve.

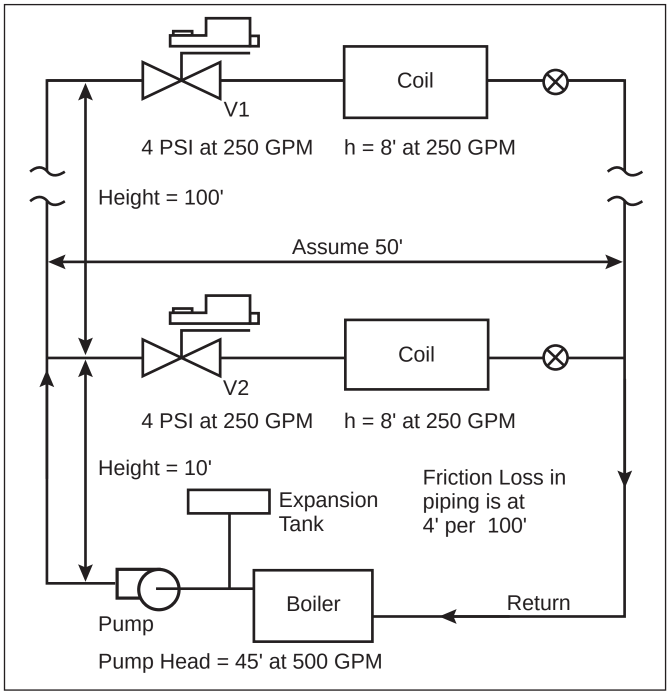

Figure 32 - Boiler, pump and piping system

Static head
The pressure on the pump when the system is off is the weight of the column of water above it.

V1 has almost no pressure on it. It is on the top floor, and there is no piping above it. The system has a fill pressure which is typically 20 psi.

V2 has 100' of water pressure on it. We do not know the diameter of the pipe, but it does not matter. We know that there is 1 psi per 2.3' column height. Thus 100' is 100/2.3 = 44 psi static pressure. The column of water is on both sides of the valve, but this is inconsequential.

Fill pressure
By applying a fill pressure that is 20 psi higher than the static pressure, a sufficient pressurization is achieved. This gives 44 + 20 = 64 psi for V2 in the example above. V1 would have only the fill pressure of 20 psi.

Pump pressure
When the pump is running and the valves are full open, then the head at the valve inlet is the sum of the various heads.

There is about 130' of pipe on the way to V1, and the piping loss typical average is 4' per 100' of run. The loss is about 5'.

V1 has the pressure of the pump or 45' less the piping losses of 5' on the way to the valve or 40'. This is 40/2.3 - 17 psi of pump pressure.

V2 has about 25' of pipe between it and the pump for 1' of friction loss. 45 - 1 = 44' of pump head or 44/2.3 = 19 psi of pump pressure.

Total pressure
The total for V1 will be 17 psi pump + 20 psi fill gives 37 psi. The total for V2 will be 44 psi column height + 19 psi pump + fill pressure of 20 psi = 83 psi. (The balancing valve will be taking a good deal of pressure loss in this circuit.)

Dead head pressure
Many systems do not have supply to return bypass pressure control. Both valves could be near closed and take full pump pressure.

When V1 and V2 are both closed, the whole pump pressure appears at the inlet of V1 and V2. In this system there is no pressure control, and it would be possible for the pump to dead head. There is usually some system pressure control. Three-way valves or a supply to return bypass would accomplish this. The pump curve becomes important in this case. Typically the pump curve has from 30 to 45 degree slope. At no or very low flow, the pump pressure could rise to say double the design operating point or 90'. The valves would have to withstand pump pressure of 90/2.3 or 40 psi. The pump curve must be examined. It is rare to allow the pumps to deadhead against the valves. The design engineer has the data necessary to specify this pressure.

Thus the body static pressure rating is the sum of the column, fill, and pump pressures. The fill pressure and any column heights must be added to the pump pressure.

If 3-way valves were used, or if there were enough valves and a load situation to provide diversity, the pump pressure would never rise to this point because it would be relieved by the bypass or other valves.

Close-off pressure
Close off is the maximum differential which will appear across the valve disc and seat. It is necessary to choose the worst condition.

The **system off** condition has no differential across the valve. The weight of the water on each size balances; the fill pressure appears on both sides.

In **normal operation** with both valves full open, the pressures do not include the height of the columns of water; the supply and return cancel out. The fill pressure is seen on both sides of the valve also. The only pressure is the pump differential head, less the friction loss on the way to the valve. Both valves have a given 4 psi drop when full open.

If one valve were closed and the other open, the pump pressure would rise as flow volume went down. (This affects the flow quality of the partially open valve.) The pressure rise can only be found by examination of the pump curve.

If pressure control exists, then the pressure may not increase or will increase to a specified level. That level is the differential or close off pressure in this case.

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As **both valves close** against full pump pressure, the worst close off condition exists. As assumed above, this could be near 40 psi pump pressure.

This is the differential or close off pressure in this case. The deadhead condition exists. Although rare, some systems experience this condition.

Typically the ANSI 125 class is sufficient since the typical pressures met in HVAC systems are in the order of 30 psi. Tall buildings have the same close off as low rises, but the static pressures could be high.

The rated close-off pressure for a combination of actuator, linkage and valve is the **differential** pressure it can close against. This pressure must be higher than the maximum differential pressure the valve will be subject to in the installation.

In a variable flow system (2-way valves) without pressure control, the close-off pressure is the maximum pressure the pump can produce. We do not subtract flow losses between the pump and the valve. When no water is flowing, the valve sees the full pump pressure.

In a constant flow system (3-way valves), the differential pressure across the valve is essentially constant. In order to get a safety margin, the close-off pressure is calculated 1.5 times the differential pressure. Since water is always flowing, the valve will never see the full pump pressure. Since the valves above are sized for a 4 psi drop, 6 psi is the close off required.

There are variable flow systems which use 3-way valves to provide minimum flow at the furthest terminals (10-15%.) These three-way valves are subject to a variable differential pressure. The maximum differential pressure is the pump head minus the pressure drop across the balancing valve. These valves see more of the pump pressure as the 2-way valves close down.

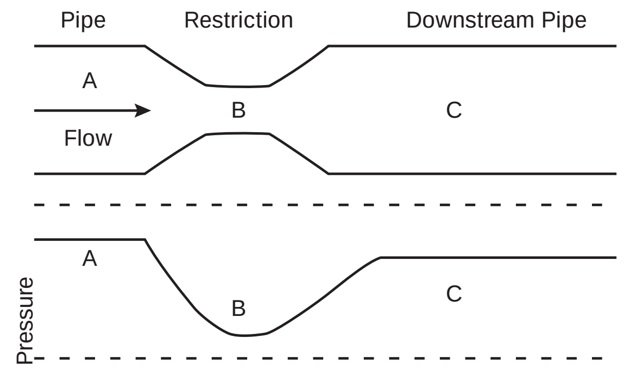
In a typical globe valve, the differential pressure acts upon the plug and produces a lifting force. The force is calculated by multiplying the area of the seat with the differential pressure. The actuator must be strong enough to overcome the lifting force and the friction in the box packing.

Ball valves are operated by a quarter turn motion. The seats are pretensioned and exerts a certain force upon the ball. The required torque is therefore the same regardless of the differential pressure up to a certain value (which usually is quite high). When this value is exceeded, the torque increases in proportion to the differential pressure. In addition the geometry of the ball valve is such that pressures are mostly balanced. They have very high close off ratings.

Butterfly valves have high dynamic forces when modulating. Dynamic forces acts upon the disk and produce either positive or negative torque depending upon the position of the disk. When closed the pressure on the 2 half sides of the blade are mostly balanced but the rubber seals require a high torque at close-off.

Globe valves require a special linkage so a linear stem movement is accomplished. The force must be sufficient to produce the required close-off pressure. In order to overcome the friction in the box-packing of the valve, about 15 lbs. should be subtracted from the rated force of the linkage. Calculate the area of the seat. It is usually the same as the valve size. For example; 1.5" valve. Area = $\prod r^2$ therefore 3.14 x $(1.5/2)^2$ = 1.77 sq. in. The rated force of the linkage is 150 lbs. Close-Off Pressure = (150 - 15)/1.77 = 85 psi.

Cavitation



At A the pressure is higher than at B or C
At B the pressure drops due to high velocity
At C the pressure again goes up. If it goes above a certain point and air was released, then cavitation occurs.

Figure 33 - Cavitation

Cavitation is the formation of vapor bubbles in areas of low pressure in a piping system. It can occur whenever a restriction and expansion occurs; it is not limited to valves and pumps.

When water passes through the restriction in a valve it increases velocity and velocity pressure and lowers its static pressure. When it again decreases its velocity the static pressure goes back up. When the static is low air entrained in the water can be released and form bubbles. Then the bubbles are again crushed when the static pressure increases. The bubbles implode in a pointed shape. Some hit the sides of the pipe as far away as 20 pipe diameters from the valve. They can eat away the pipe. The sound is not hissing, it is more like gravel flowing inside the pipe.

Cavitation can be predicted by solving this equation:

Maximum allowable $\Delta P = F_L^2 (P1 - F_F \times VP)$

F_L is the liquid pressure recovery factor (see Chart 1).
P1 is the inlet pressure psia
F_F is the liquid critical pressure ratio factor (.96 for water)
VP is the vapor pressure in psia at inlet temperature (see *Chart 2*).

From the vapor pressure charts it is seen that hot water cavitates easier.

F_L is a function of individual valves. It also varies with the amount open – decreasing as the valve opens. Thus cavitation occurs easiest when a valve is full open.

For all practical purposes these values can be used:

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Chart 1 F_L		
F_L	Type valve	Amount Open
1	Globes	All positions
1	Control Ball	All positions
.75	Butterflies	10° open
.65	"	70° open
.5	"	90° open
.9	Standard Ball	10° open
.75	"	70° open
.5	"	90° open

Chart 2 Vapor pressure of water	
°F	psia
32	.09
40	.12
50	.18
75	.43
100	.95
125	1.9
150	3.7
175	6.7
190	9.3
200	11.5
210	14.1
212	14.7

Normally cavitation is not a problem as long as the pressure drop is kept low. Closed systems present few problems because the outlet pressure stays high. The problem application is open systems – the primary one being cooling tower bypasses.

The pressure drop must be kept below the ΔP calculated above.

Cavitation is eliminated by increasing the pressure in the system. If the pressure is increased sufficiently, the lowest pressure inside the valve will exceed the vapor pressure, no steam bubbles are formed and cavitation is avoided. The system pressure gets lower higher up in a building. Therefore, valves located high up in a building, is especially vulnerable to cavitation. The expansion tank (pressure tank) should be sufficiently charged, so the pressure always is high enough to prevent cavitation in all parts of the system.

In addition to increasing system pressure there are valve remedies for cavitation. High velocity is the origin of the problems and decreasing it may be possible. A ball or butterfly valve need not be fully opened. By limiting opening the velocity is decreased, of course at a higher pressure loss. These valves have such high capacity anyway that this is no problem.

13. Miscellaneous water valve considerations

Leakage

Leakage through control valves should be avoided, because it can be very costly. The leakage depends upon the type of valve and the differential pressure. It is therefore expressed as a percentage of the rated C_V value.

Example:

Rated C_V value = 400. Leakage 0.5%. Differential pressure 25 psi. Leakage = 400 x 0.005 x $\sqrt{25}$ = 10 GPM.

A very large leakage will occur if the actuator is not strong enough to close the valve, or if the linkage is not correctly adjusted.

Materials

The following will give some understanding why the choice of materials is so important.

There is dissolved oxygen in water, which makes it aggressive. In a closed hydronic system there is some corrosion at first when the system is filled, but the corrosion stops after all of the dissolved oxygen has been absorbed. The water is no longer perfectly clean, but it is not aggressive. This makes it possible to use cast iron and regular brass in the valves. It is therefore important that leaks are eliminated, because when new water is added, there will be some corrosion.

Regular brass (copper & zinc) may be subject to “de-zinc-ification” if it is used in water with dissolved oxygen or certain additives. The zinc molecules are leached out, leaving a porous copper surface. There are special brass alloys, usually with an extra high copper content, that are “dezincification” resistant. Bronze (copper, tin, zinc) valves are used in domestic hot water systems, because the potable water supply, has lots of dissolved oxygen.

The Belimo Characterized Control Ball Valve is forged brass. This is lighter and stronger than cast valves. They can be used with common additives without problems.

Cooling tower water is in contact with air, so it has much dissolved oxygen which is very aggressive. Water evaporates, leaving the minerals behind, so the mineral content is very high (calcium carbonate, magnesium silicate, rust, etc.). This together with airborne dirt may form deposits on the valves, which commonly are butterfly valves. Aluminum-bronze, rubber coated or all plastic valves are preferred, for the cooling tower applications.

Galvanic action between dissimilar metals can also occur. Movement is from iron to bronze and the small amount of bronze in a valve will not affect the high amount of iron in piping.

Temperatures

If so used butterfly valves must have seals meant for high temperature steam service. Viton seats are normally used with stainless disks.

If a valve is being used for alternate hot and cold service it must be able to handle the extremes. Just as boiler or chiller thermal shock from too fast a medium temperature change

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occurs, so a valve can be damaged also.
Temperature limitations exist in some common applications: below 35° chilled water and steam boiler feed valves which are typically at 200° or a bit higher. The actuator must be mounted using a thermal isolation linkage.
Condensation of water may occur in the actuator when its temperature drops below the dew point of the surrounding air. Water will gradually fill the actuator. Some of the Belimo actuators have semi-permeable membranes and all are NEMA 2 rated, but this is not absolute protection. A few precautions will ensure long life of the actuator. The valve is of course in no danger here.

There are a few cautions necessary when feed valves for steam boilers are selected.

First, the temperature of the water is often above 200°F and the valve is frequently mounted near a hot boiler. Life expectancy is reduced. The actuator will experience ambient above its 122°F limit unless special precautions are taken. It is best to locate the valve away from the boiler or condensate tank.

The hot pipe must be insulated and a heat shield may be necessary. The valve may open slowly, but should close quickly to avoid overfilling; a spring return actuator may be necessary. The settings of the float switches may allow slow action in both open and close operation. Overfilling is possible if a slow valve is used and the float switch is set for a fast response. If a slow actuator is used, change the float switch to compensate.

Belimo actuators are thermally isolated from the valve in all cases. This increases the life span of the actuator.

14. Steam valve sizing

Introduction
Steam systems have a boiler which produces steam which fills the pipes of the system. Pumps are not needed. The supply steam goes through the valve, then the coil. When the steam cools it condenses back into water which gathers in the trap. From there it flows through the condensate pipes back to the boiler. In large systems a receiver holds the water until a boiler water level switch turns a pump on and the condensate goes back into the boiler.

There are a number of variations but this is the system used most in commercial applications. See *Figure 34.*

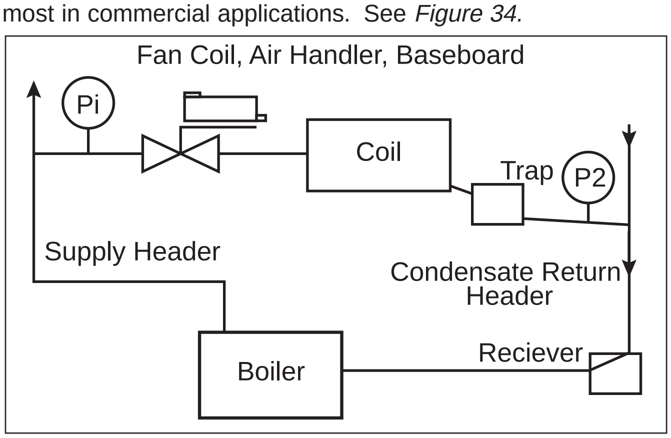

Figure 34 - Steam valve applications

P1 means Inlet Pressure.
P2 means Outlet Pressure.
This is the return header pressure after the trap.

There are some systems with vacuum in the return which require slightly different calculations. Typically P2 is zero, that is atmospheric pressure.

P1 - P2 is the differential pressure across the valve, coil, and trap. h = P1 - P2

"h" is the valve pressure drop and is used to size the valve. Low pressure systems, less than 15 psi, are sized using gauge pressure or psig – pounds per square inch gauge. High pressure systems are sized using absolute pressure or psia — sometimes called atmospheric pressure. Inside the pipes the gauge pressure is less than it would be if exposed to atmosphere. Absolute pressure adds the pressure of the atmosphere to the gauge pressure.

psia = psig + 14.7 psi.

or Pounds per square inch absolute = Pounds per square inch gauge + 14.7 psi which is the weight of the atmosphere of the earth.

High pressure systems require valves with stainless steel trim. Low pressure valves may use stainless steel trim. Erosion of the seat and disc due to high velocity steam when the valve is near closed is always a possibility. This is referred to as wire draw or tunneling. Wire draw in bronze takes place above a 30 psi pressure drop. Wire draw in stainless steel takes place above a 50 psi pressure drop. Low pressure steam systems can use bronze trim without problems.

It is important to distinguish between low and high pressure applications in order to size correctly.

This formula is used to size valves:

$$C_V = (W * \sqrt{V}) / (63.3 * \sqrt{h}) * Y$$

W = #/hr It is sometimes written as Q
V = specific volume using psig
h = pressure drop. This is P1.
Y is an expansion factor. It varies and is typically equal to .8.

Chart 3
Specific Volume Ft³ / lb.

psig	psia	√ V
0	14.7	5.2
10	25	4
20	35	3.4
30	45	3.1
40	55	2.8
50	65	2.6
60	75	2.4
70	85	2.3
80	95	2.1
100	115	2
120	135	1.8
140	155	1.7
160	175	1.6
200	215	1.5
300	315	1.2
400	415	1.1

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2 Position control of high and low pressure steam
For 2 position control the steam valve is sized for as low a pressure drop as possible. A line sized valve is used in most cases. 10% of P1 is also a valid method for both high and low pressure systems.

Modulating control of low pressure steam

LOW PRESSURE < 15 psi
For modulation, 80% of the difference between the inlet and outlet pressures is used as the valve pressure drop. Assuming atmospheric pressure at the outlet, this means 80% of the inlet pressure can be used as the drop. (Some vacuum systems may use more than this drop.)

Use

h = 80% (P1 - P2)

to size the valve.

Equal percentage characteristic is generally best for low pressure steam. As the valve closes the inlet pressure increases. The reduction in flow will not be as great as might be expected. An equal percentage valve counteracts this tendency.

Steam coils themselves respond linearly with an increase in flow. Unlike water coil response, steam coils are linear.

Modulating control of high pressure steam

HIGH PRESSURE > 15 psi
The maximum flow through a valve occurs when the drop is about 42% of the absolute inlet pressure. After this there is no increase in flow. psia = psig +14.7 .

Use

h = .42 psia

to size valves.
Note that the outlet pressure is not used in the calculation.

High pressure steam valves are best using linear response. The density of steam does increase at low opening, but not enough to matter. The limit on flow velocity and the linear coil response result in a linear valve operating best.

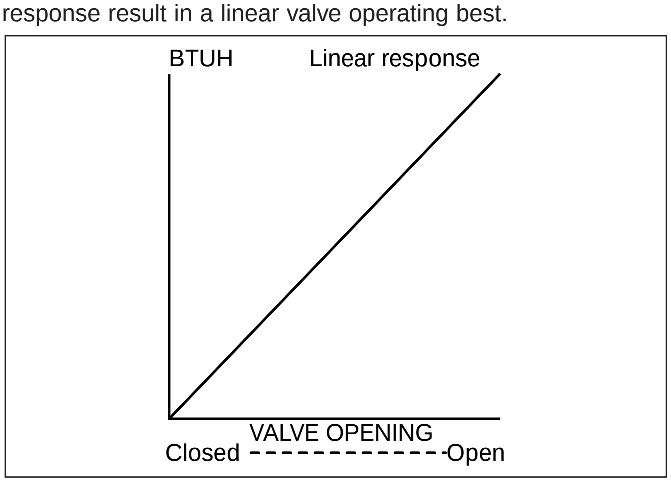

Figure 35a - Steam Coil Response

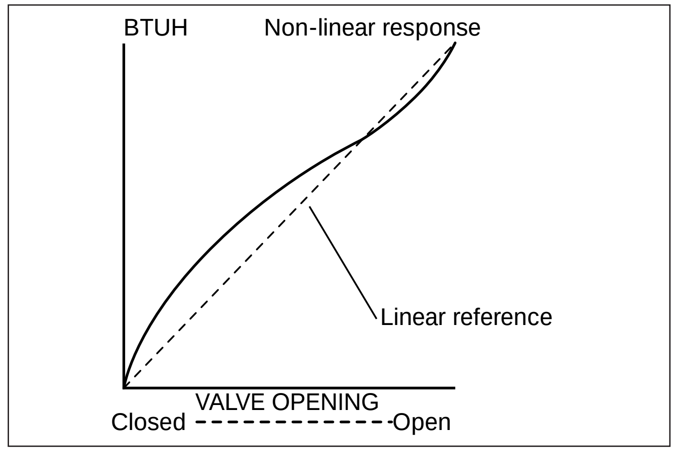

Figure 35b - Steam flow

Choked flow at sonic velocity
When steam or any compressible fluid is flowing through a restriction (valve), the flow quantity increases as the downstream pressure is reduced. The reduction in pressure is sensed as a wave traveling back up the stream. This wave travels at the speed of sound. The reduction below the point where the flow travels at the speed of sound is impossible because the wave cannot travel back to the valve.

This is called choked flow. It is the reason why high pressure steam cannot increase flow above .42 psia.

If you work out the numbers at about 15 psig the .42 psia value = .8 of P1 - P2. Cavitation is not involved in this process and does not occur.

Sizing the valve
After deciding what pressure drop, h, to use, it is then necessary to find the correct C_V by using a chart. Use of the formula is possible, but difficult.

Charts giving quantity of flow at various Cv and steam pressures are found in the Belimo Valve Sizing and Selection Guide Doc V3.1.

Superheat and saturated steam
The charts and formulae above are given for saturated steam. Just after steam boils the water content is as high as possible. If the steam is heated further the water content is less in proportion and the heat output is less than those given in the charts based on the formula above. When this occurs the C_V must be corrected.

The formula for correction with superheated steam is New C_V = C_V calculated x (1 + [.0007 x F superheat]). For each degree of superheat increase the required C_V by 1 plus superheat x .0007.

15. Summary of valve sizing criteria

Sizing information:
Pipe size
Flow rate in GPM or # of steam
Density of medium, assumed to be water = 1 in this article.
Pressure drop through coil and piping at design

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Desired pressure drop through valve in full open position
System and valve pressure ratings

Selection information
Allowable leakage
Turndown required
Temperature of medium and ambient
Environment (Nema 4 required, etc.)
Space constraints
Actuator control signal definition and any special needs

Characterized ball valves are designed for control purposes. The standard ball valve is not designed for modulating control.

16. Actuation

Control signal type
One of the first considerations is to be sure that the actuator has the ability to position accurately so that the control is accurate. Alone actuator accuracy cannot compensate for improper valve sizing or bad control loop tuning, but given those, it is essential.

Belimo actuators vary in their accuracy depending on the control application and actuator type used. Most control uses 2 to 10V signal output from the DDC or analog controller.

Resolution is the smallest change in signal that is required to make a fine incremental movement of an actuator. In general the positioning is within .1V using an 8V span from 2V to 10V (.2 mA for 4 – 20 mA control). Some actuators use a .03V resolution. Others use .2V change required to reverse direction and .04V change required to move (reposition) if the signal change is in the same direction – increase or decrease – as the last change. This is a form of signal anti-oscillation protection.

Assuming about .1V accuracy gives an 80:1 positioning accuracy. Repeatability is the same. There is no hysteresis. The circuits are all positive positioning. The actuators can be stalled at any intermediate position from full open to closed without damaging the motor. Overload protection exists on all actuators.

3-point floating control can be quite accurate depending on the performance of the controller. The time to drive from full closed to open is generally 1 minute to 2.5 minutes. This allows a lot of time for the controller to position the actuator carefully, but most controllers use simple algorithms for control and do not receive loop tuning attention. This is particularly the case with reheat valves where large numbers are installed and time to tune each loop is not given in the original installation contract.

3 point floating control is not positive positioning unless position feedback is used.

PWM control can be as accurate as 2-10V if loops are tuned. The actuators are positive positioning.
Various other signal start and span voltages can be used. Control accuracy varies depending on the controller. The actuators have the same resolution accuracy regardless of signal span. This means that use of a 6-9V signal gives a 3V span with .1V resolution. 30:1 is the highest system rangeability possible.

The **system rangeability** is a function of the worst component's accuracy in the chain of control – sensor, lag times, software, D to A output, actuator, and valve are all involved. A ball valve has a high rangeability. Repeatability may be more important than overall response characteristic. This must be examined on a case by case basis.

Adjustable start and span actuators allow sequencing actuators from same control signal. The heating valve is full open at 2V and closes at just short of 6V. The cooling valve then starts to open at a bit over 6V and is full open at 10V.

This saves an analog output from the DDC system and frequently the number of points are restricted. However this limits the rangeability. It is superior to use separate analog outputs for each valve. In particular when oversizing valves is typical and by half open the valve is at full flow, the accuracy degrades below an acceptable level.

A variation on the above is to use 2 heating valves, either steam or hot or chilled water, for the same coil. Then the valves are staged open. The first valve is sized for about 1/3 the design and the 2nd stage is 2/3. If high accuracy at partial load is the biggest issue the first stage could even be less of the proportion.

A variation on this, which takes a bit of special programming, is to modulate the 1/3 valve at low loads. Then at medium loads, when it is full open and more heat (cold) is required, the 2/3 valve is modulated and the 1/3 valve is driven closed. Then again at high loads, the 2/3 remains full open and the 1/3 valve is modulated. This produces very high accuracy.

The practice is common for steam valves, but it has been used for any fluid flow control including dampers and air flow. Programming cannot use simple PI control during the change over periods since coordination of the stages is necessary when one is closing and the other opening.

Torque and force
For globe valves, the method of calculating close off force is area of disc times pressure of medium plus any internal resistance of packing, seals, and bonnet. Globe valve linkages translate rotary torque into force. The lift of Belimo valves is either 5/8" or 1" Force generated is equal to the product of torque x radius length of arm. A rack and pinion linkage with a 1" effective radius and a force of a 133 in-lb. actuator will be about 133 in-lb./ 1" = 133 lb.

Valves 3/4" and below will have over 250 psi close off available. It is not until large valves or high differentials are used that close off ratings become a concern.

Butterfly and ball valves with a direct coupled actuator use torque not force for the calculation. The actuator torque is the torque delivered. No translation to force is necessary. See *Figure 36.*

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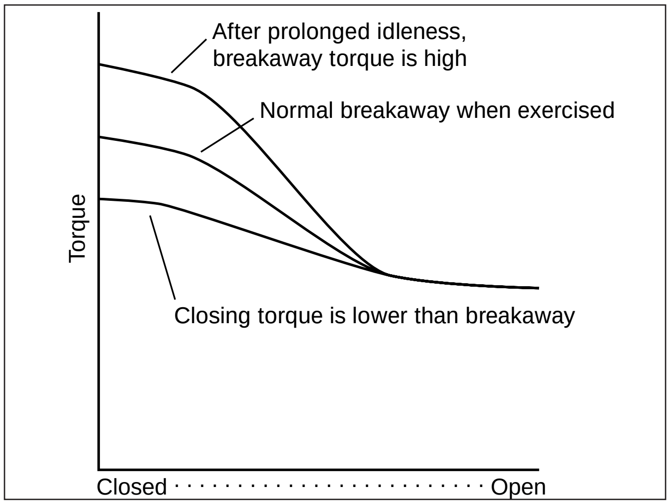

Figure 36 - Torque requirements for ball and butterfly valves

The torque requirement for ball valves depends on stem torque (tightness of stem nut), ball-seat torque caused by friction, and system pressure. System pressure pushes the ball into the seat and increases torque requirement. Closing torque is normally 20% less than opening with soft seats. After prolonged disuse Teflon creep occurs. The Teflon slowly creeps into pores of the metal ball and increases breakaway torque the first time the ball again rotates. For this reason it is recommended that the breakaway torque be at least 20% higher than normal operating torque. This has been included in Belimo products.

Dirty, abrasive water can increase torque load by seeping into seat materials. The problem application is cooling towers. The exposure to dirt is constant and special care in keeping strainers clean is sometimes necessary. A 30 mesh strainer is standard and sufficient in most cases.

When two butterflies are connected to a tee and a linkage is used, the linkage geometry can modify the torque delivery at different angles of rotation. Actuators can be stacked on top of each other or one put on each butterfly and a linkage installed to allow the actuators to help each other. Rather than 2 Belimo's, it is possible to install a larger process actuator in many cases, but the cost is higher.

The methods for calculating linkage torque-force relationships is given in the Belimo Mounting Methods Guide for dampers at the very end of that manual.

The close off pressures for Belimo valves and actuators are listed in the Valve Product Guides.

Limiting Ball or Butterfly Valve Opening to Control C_V

CHART 4

Valve Size	Pipe Size	C_Vc 100% Open	ROTATION in percent 90	80	70	60	50	40	30	20	10
2	2	210	147	96.6	63	44	27	16	8.4	4.2	2.1
2	2.5	166	133	90	60	43	24	16	8.2	4.1	2.0
2	3	134	113	82	57	42	24	16	8.2	4.1	2.0
2	4	111	98	75	54	41	23	16	8.2	4.1	2.0

Chart 4 is typical of standard ball or butterfly valves; it can be used for several purposes.

Examine the first line. A 2" valve is installed in a 2" pipe. The Cv as normally stated is 210; this is a full ported valve. This is the flow in GPM at a 1 psi pressure drop. When the valve is rotated 10° closed to only 90° open, it will pass 147 GPM at a 1 psi pressure drop. Note that at 60° open it passes 44 GPM which is the same as an equal percentage globe valve would pass full open. As it happens, the tendency to cavitate is now reduced to that of a globe.

If cavitation is a problem the valve full opening rotation can be limited. It is standard to modulate a butterfly open to 70° but no further. The response curve above 70° flattens out and is not part of an equal percentage curve. The tendency to cavitate is reduced. See *Chart 1.* The F_L , Liquid pressure recovery factor increases by 50% when a butterfly is only 70° open compared to full open. This means the pressure drop can be increased by 50% without cavitation.

More commonly if the Cv of a valve is too high, it need not be opened all the way. Increasing it more open than necessary will not have a great impact on capacity, But the resolution is better and the valve responds quicker. It does not have to close from open to, say 70° before impacting the heat output. The curve is better for control purposes in this area.

The above remarks are most appropriate to standard ball and butterfly valves. A full open standard ball or butterfly valve will cavitate at 1/4 the pressure drop compared to a globe or Characterized Control Ball Valve.

By using a 2V start but limiting the opening to say 7V maximum, the valve opening is limited. The maximum voltage is calculated using an 8V span and 90 degrees of rotation of the valve to find the maximum needed. The Belimo adjustable start and span actuators are not necessary for the above application. Resolution is best using a 2V to 10V actuator.

The best way is to use the characterized control ball valve line. The curves are equal percentage and the maximum Cv are nearer those required for most applications. Then a full 2V to 10V signal range is used and accuracy is highest.

Control Loop Tuning

The Belimo actuator has a life-span in the millions of actuations. This is quite sufficient for a 20 year life. Untuned oscillating (hunting) loops can cause a million actuations in a few years and result in premature failure. In addition a poorly tuned loop produces bad temperature control.

The opposite of an oscillating loop is a sluggish loop. Movement does not occur soon enough or rotation is too little to effect flows and droop or overshoot is extreme.

The trade off between tight control and the number and size of actuator movements is not a difficult decision making process. Space temperature changes occur very slowly. When a conference room fills with people it takes at least 5 minutes for a 1 degree space change. It may take another 5 minutes for a wall sensor to register the change. Fast actuation with many adjustments is not necessary.

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Mixed air is another very stable process. There are rarely rapid changes in outdoor air or average return air temperatures. A bare averaging thermistor sensor may respond in as short a time as 30 seconds to a 1 degree change; encapsulated sensors may take 2 minutes.

During start-up periods more actuations may be necessary to find correct operating points. During these periods the space is not occupied and tight control is unnecessary. Time exists to find the correct valve positions and many fast responses are unnecessary.

Control loops should cause productive actuation with each movement of the actuator. In process control a 2 to 3 degree movement with each actuation is considered standard. Oscillation is constant changes in position. Hunting is wide, constant swings in control signal. Dithering is small, unproductive, sometimes imperceptible, changes in signal. For floating actuators time of pulse instead of signal voltage is the control output, but the result is the same.

Proportional (and integral and derivative if used) constants should be set so that continuous oscillation of actuators does not occur.

Actuation should not occur before the effects of previous actuation have had time to affect the sensor. This may be 10 minutes for a space sensor or 1 minute for a mixed air sensor. Movement of the valve actuator when space temperature is within .5 degree of setpoint or when discharge air is within 1 degree of setpoint is not productive. This may require some logic statements which not all controllers can achieve. In any event, after a correction, at least one time constant should occur before another actuation occurs.

When 3-point floating control is used the total run time of the actuator should be entered in the program logic. When the actuator is at either the full open or full closed position, continual pulsing of actuator against end stops (end stop dithering) should not occur. Actuator should either stop and hold or drive continuously against the end stop.

Each control loop should be individually tuned. A variety of methods exist to stop oscillation. Scan times may be increased. Gains may be decreased (throttling range increased). Where the control loop is not fully adjustable, it may be necessary to find a method particular to the individual controller. Adjusting or resetting the deadband is a possibility. Derivative control is usually very difficult to tune. P + I control is sufficient. In all events the actuator must be protected from premature failure due to destructive oscillation.

Another problem occurs with many very small actuations. The brushes used with DC motors can get dust buildup between them and the motor if the movements are too small. A 3% movement of a 60 second motor is 2 seconds. This is productive from the view of good controllability and keeps the Belimo from stopping due to a high resistance brush short. (This problem can be recognized by tapping the actuator and observing that it starts to work again.) A number of actuators use brush type motors.

GOOD CONTROL LOOP TUNING PROMOTES GOOD TEMPERATURE CONTROL AND LONG EQUIPMENT LIFE.

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18. Glossary

Air separator
Device placed at high point of system to remove dissolved air from the water so potential for oxidation of metals, air pockets and cavitation are reduced.

Alternate Hot and Cold Water Use
Some valves are used for both hot and cold water depending on the season. If so the construction must be such that the changing temperatures do not cause thermal shock – cracking a component — and thereby destroying the valve.

Angle Body
While most control valves are straight through flow, some are right angled or L shaped. These are used primarily for radiation.

Angle of Opening
The degrees or percent of opening of a disc or ball in a rotary valve. Globes are not rotary.

ANSI Class
Temperature and pressure ratings of a valve.

Authority
The ratio of the wide open pressure loss through a valve to the system pressure loss (including the valve) across the subcircuit in which the valve is installed.

Automatic flow control valve
Valve connected in series with the control valve and coil to limit the flow, so it can not increase above the adjusted maximum value. It will be fully open when the flow is less than the setpoint. It is also called; “automatic balancing valve” or “self-adjusting dynamic balancing valve”.

Ball valve
Rotary valve whose turning element is a ball with a whole drilled through it. Several types are used. Standard port and reduced port ball valves were originally made for balancing and shut off purposes. Characterized port valves have shaped flow passages to produce flow characteristics like those of other control valves for modulating control.

Body
The outer case of the valve which contains the water, steam, or air medium.

Body Rating – nominal, actual
Maximum allowable system pressure, within a specified temperature range. The nominal rating is the ANSI class. In some cases the actual rating varies. Check all specs to be sure the ratings meet the service.

Boiler
Hot water or steam generator.

Booster pump
Small circulating pump often used as a secondary pump or a runaround pump.

Bonnet thread
The bonnet has thread to which a nut is often attached to allow mounting of the linkage. In retrofit applications the thread type must often be determined to allow selection of adapters to mount the Belimo linkage.

BTUH
British Thermal Units per Hour. Amount of heat necessary to raise one gallon of water 1°F.

Bubbletight
Valve with a low leakage, for example a butterfly valve with resilient liner.

Butterfly valve
Single bladed rotary valve usually for high capacity control. The flow characteristic is shallow logarithmic. In modulating control the valve is usually opened a maximum of 70°.

Bypass
A pipe path which bypasses water around a boiler, chiller, or valve and coil.

Capacity Index
See Flow Coefficient.

Cavitation
In pumps and valves. Air entrained in water is released when a restriction causes static pressure to be lowered. The air bubbles implode when the water has passed the restriction and velocity decreases and the pressure again increases. The imploding vapor bubbles erode the surrounding material and cause an intense noise.

Characterized control ball valve
A ball valve which incorporates a characterizing disk. Reduces the C_V value to approximately the same C_V value as a globe valve of the same size. This disk determines the valve characteristic. Disks with differently dimentioned openings are available, so a number of C_V values are available for each valve size.

Characterized port
A valve port which modifies the flow characteristic for modulation. Equal percentage characteristics are usually produced.

Choked flow
When velocity of steam reaches the speed of sound it can go no faster and flow is referred to as “choked.”

Chrome plating
Ball valves are frequently chrome plated. Stainless is recommended for modulating control.

Circulating pump
Small pump used in residential, light commercial, or coil runaround service.

Close Off Pressure
The maximum differential pressure against which a valve can close off and keep any water from passing through.

Common Port
The port in a diverting or mixing valve which is common to the other two.

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Composite seat
Soft seat, often Teflon, used instead of metal. Leakage is reduced from the firm contact with the disc.

Compressible flow
Steam is a compressible fluid while water is not. Air is compressible but not at the velocities found in HVAC service.

Contoured Plug
A plug shaped to achieve a particular flow characteristic.

Corrosion
Hard water and other chemicals in water will gradually eat away metals in valves and piping elements.

De-coupled
A part of a hydronic system is de-coupled from the rest of the system by an "equalizer bridge" between supply and return. Variations in the differential pressure in the primary circuit will not be transmitted to the secondary circuit.

Density
Mass per unit volume.

Differential pressure
The pressure differential between two points in a hydronic system.

Direct Acting
A valve which increases flow with an increase in signal.

Disc Rotation
A butterfly rotates the disc. A globe lifts the disc.

Diverting
A valve with 2 inlets and one outlet is mixing. A valve with one inlet and 2 outlets is diverting.

Double Seated
A valve with 2 seats and discs. By balancing the pressure across the discs torque requirement is very low. Leakage is high since getting the seats to touch the discs simultaneously is difficult. Mixing and diverting valves are forms of double seated valves.

Dynamic Pressure
Pressure due to flow of water as opposed to its weight.

Equalizer bridge
A bypass pipe between supply and return which de-couples two different parts of a system.

Erosion
The loss of material due to cavitation or a too high flow velocity. Generally velocity limited to 6 fps in copper and 10 fps in steel.

Expansion tank
A tank which maintains the pressurization of the system. It absorbs the volume changes due to temperature variations of the water. Some expansion tanks have a pressurized bladder inside.

EPDM
Material used in seats, particularly butterfly valves.

Equal percentage
Logarithmic response curve. In valves usually refers to a deep curve. Globe valves and characterized ball valves have equal percentage curves. Standard ball valves and butterfly valves have shallow logarithmic curves, not normally considered to be equal percentage.

Flanged
Larger valves use flanges to bolt to pipes instead of screw threads. Heavy duty.

Flared
Flare fittings are used in direct copper to copper valve connection. Small zone and terminal valves may have flared connections.

Flow coefficient
C_V. GPM of 60°F water that flows through a valve at a 1 psi differential pressure. Usually thought of as full open valve flow, but can be used at partial openings also and in process such use is standard.

Kv. Cubic meters/hour, m^3/h of water, flowing through a valve with 100 kPa differential pressure.

Flow resistance
Resistance to the flow in pipes, fittings, coils, valves, etc.. It is often expressed as a pressure drop, although not theoretically correct. K is the loss coefficient. $\Delta Pt = K \times Pv$

F_p
Piping geometry factor. See pipe geometry.

Guide
Portion of valve which guides the stem and disc onto the seat.

"h"
Pressure or pressure difference.

Hysteresis
The required change in the control signal to reverse the movement of an actuator. The "dead zone" is the hysteresis. Often expressed in percentage of the control signal full span.

High Pressure
Pressures above 15psi in steam.

Inherent characteristic
The valve flow response characteristic or curve when the pressure drop across the valve is maintained constant. See Installed Characteristic.

Installed Characteristic
When a valve is installed with series losses the inherent characteristic changes becoming shallower with a small series loss or over the linear with a lot of series loss.

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Isolation valve
Manual valves before and after a control valve or other pipe element which allow disconnection without water leakage.

K
The loss coefficient in water system elements. (C is used in air systems.)

Leakage
Water leakage through the seat and disc of the valve. Could be a leaky stem, but not normally used in this context. The leakage inside a shut valve is usually shown as a percentage of the C_v value.

Lift (stroke, travel, percent open)
In a globe, the inches of travel necessary to lift the disc off the seat and go to full open. Typically 1/2” to 1 1/2”

Line size
A valve whose inlet and outlet are the same size as the pipe.

Linear characteristic
A valve response curve is linear if an equal amount of lift, rotation, or signal change always produces the same change in flow quantity.

Low Pressure
In steam, pressures below 15 psi.

Manual balancing valve
Circuit balancing valves (cbv) are used to adjust the flow resistance in the different parts of a hydronic system, to obtain the desired flow distribution. It is manually adjusted and has a calibrated stem/hand-wheel. The valve has two ports to facilitate measurement of the differential pressure across the valve.

Materials
Brass, Bronze, Chrome plated, Cast Iron, Composites, EPDM, Stainless Steel, Teflon, TEFZEL, Viton, Zinc, etc. Materials used in valve construction.

Medium
Water, steam, air, etc. which run through the valve.

Most resistive circuit
The loop with the largest pressure drop.

Normally open
Valve that is open when the signal to the actuator is low or when power is lost.

Normally closed
Valve that is closed when the signal to the actuator is low or when power is lost.

NPT
National Pipe Thread. Screwed end valves are sometimes referred to as NPT. Plumbing pipe uses inside diameter.

Pipe losses
The pressure drop along a certain length of pipe. It is dependent upon the distance, size, flow velocity and the inside surface roughness. The resistance of pipe bends, fittings etc. are often expressed as “equivalent pipe length” which is added to the pipe losses. K value loss coefficients are more accurate.

Pipe geometry
Typically, a properly sized control valve is smaller than the pipe it is connected to. Pipe reducers are used and the change in the cross section area, results in a lower C_v value. This is especially true for ball and butterfly valves (globe valves and Belimo’s characterized ball valves are affected very little). The control valve manufacturer should supply a table showing the “corrected” C_v values for the different valve and pipe sizes. C_vc is the corrected C_v. $C_vc = F_p \times C_v$ where F_p is the piping geometry factor.

Pressure
Usually refers to the pressure differential between the inside of a pipe or vessel, and the atmosphere. It is the gauge pressure that is shown on a regular pressure gauge (psig).

Pressure drop
The pressure differential across a resistance to the flow, for example a valve.

Pull pump
Coil pump. Also called runaround pump.

Pump head
The pressure increase between the inlet and outlet of a pump.

Pump performance curves
The relation between the flow and pump head at different impeller diameters. There are also curves showing the efficiency % and horse power at different operating points.

Radiator Valve
Either manual or self contained valves used to control flow through a radiator.

Rangeability
The flow through a control valve follows a certain characteristic down to a “minimum controllable flow” where the flow abruptly changes and the valve closes. The rangeability factor is the ratio between the maximum flow and the minimum controllable flow. Laboratory valve characteristic. See turndown.

Reverse Acting
Valve which decreases flow with an increase in signal. Typically a function of the actuator.

Reset schedule
The supply water temperature to the heating or cooling system is changed with respect to a parameter that is proportional to the load, usually the outdoor temperature.

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Resolution
The finest positioning change of an actuator when discreet changes of the control signal are made. Electronic actuators have a far greater resolution than commercial grade pneumatic actuators.

Runaround pump
Coil pump. Also called pull pump.

Seating Torque
Torque necessary to close valve.

Secondary pumps
Pumps used to provide circulation in secondary loops that are de-coupled from the main loop.

Self Contained Valve
Valve with built in operator. Used for baseboard, radiation.

Service temperature range
High and low temperatures which a valve can withstand and still function properly.

Single seated
Most valves have single seats as opposed to the double seated valves.

Soft Seat
Resilient seat versus the metal to metal or hard composite.

Stainless steel
Strong hard steel which withstands corrosion and high temperatures.

Static Pressure Rating
Pressure rating of a valve against weight of water against the body walls.

Stem travel
See lift.

Stroke
See lift.

Sweat
Soldered connections to valve inlet and outlet.

System curve
A curve showing the relationship between flow and pressure in a system served by a pump. An exponential curve representing the system pressure loss versus the flow quantity.

Throttling
Proportional or modulating.

Trim
Inside components of valve which have contact with medium. The seat, plug, stem, and ball are all trim components.

Turndown ratio
The turndown ratio relates to an installed valve. It is the ratio between the full flow and the minimum “controllable” flow of the valve. The turndown ratio always is less than the rangeability factor because it is dependent upon both the rangeability factor of the valve itself, and its valve authority. In addition the full flow required is usually less than the capacity of the valve.

Union valve
Valve with a built in union coupling on one fitting end to facilitate installation.

V-port
Shaped Characterized ball. Parabolic.

Variable speed pumping
The pump is operated at a variable speed by a variable speed drive, which can be adjusted manually, or controlled automatically. It reduces the operating costs of the pump significantly and improves pressure control.

Welded End
High pressure end fitting on process valves.

Wire draw
Erosion of the plug or seat of a valve due to high velocity steam when the valve is near closed. Also called tunneling. The damage looks like a sharp cut along the side of the plug or seat.

Bronze wire draws at 30 psi ΔP. Stainless at 50 psi ΔP. A 50% safety factor recommended.

Zone Valve
Small valves to control zones. Can be on-off or modulating.

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19. TABLES & Formulas

TABLE 1.
Inside Flow Area in inches

Size	Steel - 40	Copper K
1/2	0.3	0.22
3/4	0.53	0.436
1	0.86	0.78
1-1/4	1.5	1.22
1-1/2	2.04	1.72
2	3.36	3
2-1/2	4.8	4.66
3	7.4	6.6
4	12.7	11.7

TABLE 2.
GPM at typical velocities

Ft/Sec	3		4		5		8		10	
	Steel-40	Copper	Steel-40	Copper	Steel-40	Copper	Steel-40	Copper	Steel-40	Copper
1/2”	2.8	2	3.7	2.7	4.7	3.4	7.5	—	9.3	—
3/4”	4.9	4	6.5	85.4	8.2	6.8	13.1	—	16.4	—
1”	8	7.3	10.7	9.7	13.4	12.1	21.4	—	26.7	—
1 1/4”	14	11.4	18.7	15.2	23.3	19	37.3	—	46.6	—
1 1/2”	19	16	25.4	21.4	31.7	26.7	50.7	—	63.4	—
2”	31	28	41.8	37.3	52.2	46.6	83.6	—	104.4	—
2 1/2”	45	43	60	58	75	72	119	—	149	—
3”	69	61	92	82	115	103	184	—	230	—
4”	118	109	158	145	197	181	316	—	395	—

For others divide any flow by the velocity to get V at 1 ft/sec and interpolate or add.

Heat Transfer Formulas
Air
Sensible Heat, BTUH = CFM X 1.085 X ΔT
Latent Heat, BTUH = CFM X .68 X grains of moisture per lb standard air
= CFM X 4.5 X (Entering Air Enthalpy – Leaving Air Enthalpy)
Total Heat (BTUH) = CFM X 60/Cu.Ft./LB X (Entering Air Enthalpy – Leaving Air Enthalpy)

Water Coils
GPM = BTUH air side load / BTUH water side sensible capacity
GPM = BTUH / ΔT X 60 X 8.34 lb/gal
GPM = CFM X 1.08 X ΔT air / ΔT water X 500
(Sensible, 500 is K factor)
= D Enthalpy X CFM X .075 X 60 / ΔT X 500
(Total cooling)
Alternately, GPM = CFM X BTU/lb dry air/113 X ΔT water side

1.085 = .24BTU/#AIR-°F X 60min/hr x 1# air/13.4 cu. ft.

For other conditions, substitute 13.4 with other specific volume

Water to water heat exchanger
BTUH supply side = GPM X 500 X ΔT water load side
Note that 500 in formulas above could be replaced by a constant factor K.

TABLE 3.

Constant K for heat exchangers and coils	
Water Temp F	**K**
60	500
100	496
150	490
180	487
200	484
250	479
300	473

1.08 is a scaling constant. It is derived from .24BTU/lb air °F X 60 min / hr x 1lb air / 13.4 cu.ft

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Radiation
Radiators and convectors are sized according to equivalent direct radiation or EDR.

TABLE 4.

Average Temp BTUH / EDR		
°F	Cast Iron (Room Ambient)	Convector (65°F inlet)
215	240	240
200	209	205
190	187	183
180	167	162
170	148	141
160	130	120
130	76	70
100	30	27

When EDR is known, calculate the steam supply quantity needed, W
W = .24 EDR where .24 is # steam/ unit EDR

Steam Coils
W = Steam in lb / hr = BTUH air side load / Latent heat of steam
= CFM X 60 X .075 X .24 X DT / 970
= CFM X ΔT / 890
= BTUH air / 1000 BTU/lb steam
Approximate heat of vaporization of steam = 1000BTU/lb

TABLE 5.

Saturated Steam Pressure – Temperature Table				
psig	**psia**	**T °F**	**T °C**	**kPa gauge**
0	14.7	212	100	100
1	15.7	215	102	107
2	16.7	219	104	114
3	17.7	220	104	120
5	19.7	227	108	134
7	21.7	232	111	148
10	24.7	239	115	168
15	29.7	250	121	202
20	34.7	259	126	236
25	39.7	267	131	270
30	44.7	274	134	304
35	49.7	280	138	338
40	54.7	287	142	372
50	64.7	298	148	440

Steam to water heat exchanger
W = Steam in lb / hr = GPM X ΔT X .5
(.5 = 8.34 lb water / gal X 60 min/hr X lb steam/1000 BTU X BTU/lb water-°F)

Steam Jet Humidification
Q = 4.5 x ΔW lb/hr moisture

Where ΔW = (lb of moisture per lb of air leaving)/hr – (lb of moisture per lb of air entering)

Valve C_V is then sized directly from charts for the boiler pressure and the # per hour needed. Use the same sizing rules already given for low and high pressure systems.

TABLE 6.

Properties of Saturated Steam			
Vacuum in. Mercury	**Steam Temp or Boiling Point °F**	**Specific Volume V**	**$\sqrt{V}$**
		cu.ft./lb	
29	77	705	26.6
20	161	75	8.7
12	187	43	6.6
6	201	33	5.7
2	208	29	5.4
psig			
0	212	27	5.2
2	219	24	4.9
4	224	21.4	4.6
5	227	20.4	4.5
8	234	17.9	4.2
10	239	16.5	4.1
12	244	15.4	3.9
15	250	13.9	3.7
20	259	12	3.5
25	267	10.6	3.3
30	274	9.4	3.1
40	287	7.8	2.8
50	298	6.7	2.6

Belimo Actuators are not rated above 250°F.

Superheat
The formula for correction with superheated steam is New C_V = C_V calculated x (1 + [.0007 x F superheat]). For each degree of superheat increase the required C_V by 1 plus superheat x .0007.

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Metric System
c = specific thermal capacity = kJ/kgK and for water is 4.196 ≈ 4.2
4.2 [kJ/kgK] = **4'200** [J/kgK] = **4'200** [Ws/kgK]
Joule = 1 watt-second
kw = kilowatt
N = kg-m/s^2 = .225 lb force
Pa = N /m^2
Nominal thermal power Q_{100} is in kW
Q = heat flow in kW
Q = m x c x Δt with m in kg/s and c in kJ / kg-K
Q = V x r x c x Δt with V in m3 /s and c in kJ / kg-K
Q = 4190 x m^3 /s x ΔT
Q = 1.2 X L/s X Δh
Where Δh = enthalpy difference in kJ/kg dry air; requires metric psychometric chart.
Sp. Gr. = Specific gravity, water = 1
Specific Heat Water = 4190 J/kg-K
V = Flow (liquid) = m^3/h, m^3/s
p = Density = kg/m^3

1 m water = 9.9 kPa
1 mm Hg = 133.3 kPa
100,000 Pa = 1 bar

Valve Sizing
$K_{VS}= V_{100} / \sqrt{\Delta P_{V100}}$ by definition

ΔP_{V100} at 1 bar
V_{100} in m^3/h
$\Delta P_{V100} = K_{VS} = (V_{100} / K_{VS})^2$

$K_{VS} = V_{100} / \sqrt{\Delta P_{V100}}$
is analogous to
$C_v = GPM/\sqrt{\Delta P}$

Example:
Given: V_{100} = 4 m^3/h and ΔP_{V100} = 0.3 bar
$K_{VS} = V_{100} / \sqrt{\Delta P_{V100}}$
$K_{VS} = 4/\sqrt{.3} = 7.3$
A valve with a K_{VS} = 7.3 m^3/h is required.

Conversions
1 bar = 14.7 psi = 29.9 in. Hg = 33.9 ft. w.g.
1 BTU = 3.14 watts
1 °C = (°F - 32°)/1.8
1 kJ/kg = .43 Btu/lb dry air
1 joule = .738 ft-lb = .00095 Btu
1 kg = 2.2 lb
1 kg/m^2 = .0624 lb/ft^2
1000kg / m^3 = 62.4 lb/ft^3
1 m/s = 197 fpm
1m = 3.28 ft
1m^3/hr = 4.4 GPM
1 N = kg-m/s^2 = 8.85 lb force
1 Pa = .000145 psi = .004 in. w.g. = .004 in. w.g.
1 kPa = .145 psi = .296 in Hg
1 W = J/s = 3.41 Btu/hr
1m^3/hr = 4.4 GPM

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Comparison of Various Valves at Rotated Positions with Fp Factors.											
Valve Size	Pipe Size	C_v / C_vc 100% open	C_vc ROTATION in percent 90	80	70	60	50	40	30	20	10
Ball Valves											
1/2"	1/2" to 1"	(1) 1	0.88	0.68	0.48	0.35	0.25	0.15	0.1	0.07	0.02
1/2"	1/2"	(2) 9.8	6.9	4.5	2.9	2.1	1.3	0.8	0.4	0.15	0.02
1/2"	3/4"	7.4	5.9	4.4	2.9	2.1	1.2	0.8	0.4	0.15	0.02
1/2"	1"	6.3	5.3	3.8	2.6	1.9	1.2	0.8	0.4	0.15	0.02
2"	2"	210	147	96	63	44	27	17	8	4	2
2"	2.5"	166	133	90	58	42	26	17	8	4	2
2"	3"	134	113	82	56	38	25	17	8	4	2
2"	4"	111	98	75	53	37	25	17	8	4	2
Butterfly Valves											
2"	2"	166	151	123	96	71	50	33	20	10	5
2"	2.5"	141	128	104	82	61	40	30	20	10	5
2"	3"	121	110	90	70	52	38	30	20	10	5
3"	3"	340	309	252	197	146	100	65	30	15	8
3"	4"	282	257	209	164	121	90	60	30	15	8
3"	5"	241	219	180	150	111	80	55	30	15	8
Belimo Characterized Control Ball Valves and Globe Valves (3)											
1/2"	1/2" to 1"	1	0.7	0.49	0.34	0.22	0.17	0.11	0.07	0.05	0.01
1"	1" to 1.5"	10	7	5	3	2	2	1	0.7	0.5	0.25
2"	2" to 2.5"	40	28	20	14	9	7	4	3	2	1

This chart is for study and comparison purposes. Not all C_v's are representative of Belimo products.

(1) Note that use of 1 for full C_v gives the decimal equivalents of the C_v's at the rotated positions.

(2) Note the large amount of reduction in C_v at the full open positions, but no effect at near closed positions.

(3) There is negligible effect on the C_v at reduced pipe sizes for low capacity valves.

Inlet conditions can reduce C_v. Elbows or fittings near the valve inlet can cause reduced Cv.

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Application Information
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Doc. V4.2 Valve Applications Guide

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