2.
Automatic control valves

2.0 Ease of control

Hydronic balancing has a very important role to play in all aspects of temperature control, both heating and cooling, in buildings. A typical example that illustrates this is an office building heated with hot water radiators. A boiler produces hot water which is circulated to the radiators all over the building. A control system controls the supply water temperature versus the outdoor temperature, according to a "reset curve." The reset curve increases the set-point for the supply water temperature, in proportion to the drop in the outdoor temperature.

The supply water temperature is continuously adjusted so the right amount of heat is always supplied to compensate for the heat losses as the outdoor temperature changes. Unfortunately, there will not be a homogeneous distribution of the heat to the different parts of the building. The parts of the hydronic system closest to the boiler and circulating pump will probably be the "most favored circuits," and the parts at the furthest end will tend to be the "least favored circuits." The favored circuits will get excessive flows, and will provide too much heating. The least favored circuits will get too little. Parts of the building will be overheated, and other parts will be too cold.

This will cause complaints that usually are solved by increasing the adjustment of the controller, until the supply of heat is also satisfied in the coldest rooms. This means that all the other rooms will be overheated, and energy is wasted.

2.1 The correct solution

The correct solution is to use balancing valves, and adjust the resistance in the different parts of the system so the desired distribution of the flow is accomplished. Now, all the rooms will be evenly heated and the controller can be adjusted to provide a supply water temperature which is no warmer than is necessary to reach and maintain the desired indoor temperature. The result is substantial energy savings and improved comfort.

Without balancing valves it is hard to analyze the function of a system, and there is a risk that incorrect conclusions are made. In many cases boilers, pumps, valves, etc., have been replaced with larger units, when the real problem was the lack of balancing.

2.2 Temperature control and energy management

Hydronic balancing is a prerequisite for a proper automatic temperature control and energy management system. Hydronic balancing is laying the foundation for the control of the hydronic system. It makes it possible for the two systems to cooperate and provide energy savings without sacrificing comfort. Without hydronic balancing the inherent characteristics of the hydronic system will fight the efforts of the control system to save energy.

Hydronic balancing will save energy by evenly distributing the supply of heating and cooling to the different parts of a building, and will also eliminate excessive flows in the most favored parts of the hydronic system. Therefore, it reduces the total circulated volume.

The pumping cost can be significantly reduced, and because the flow and pressure in all parts of the system are predictable, savings can be made by installing valves, fittings and other components that are not any larger than needed.

Also important is the ability to measure the flow in all the different parts of the hydronic system. By doing this, a system can be analyzed and any problem can be detected. Without balancing valves the hydronic system will be an anonymous compilation of pipes, pumps, coils, valves, etc., because the thermometers alone will not give a clear picture of what is going on.

2.3 Control valves

Control valves are really nothing more than manual valves with an actuator, usually either an air-operated diaphragm or an electric motor. Control valves vary the supply of heating or cooling in response to signals from a controller, resulting in the desired temperature being reached and held. Most control valves are either two-way or three-way valves.

2.3.1 Two-way valves

The valves are operated by an actuator that provides either a linear or rotary stem movement, depending upon the type of valve. The most common two-way valves are globe valves, which have a linear stem movement. Globe valves have a stem which, using a linear movement, operates a plug up and down against a seat inside the valve body. Single seated two-way valves usually provide a tight shut-off, using normal actuators. However, if the valve size and differential pressure are large, the lifting force that acts upon the plug can be so large that the actuator is not strong enough to close the valve.

A double seated valve can solve this problem. It has two plugs mounted on one stem, and acts against two seats. The flow pattern inside the valve body is such that the differential pressure acts on the two plugs in opposing directions so the forces are, to a great extent, cancelled. Unfortunately, it is hard to produce a double seated valve where the distance between the seats is exactly the same as the distance between the plugs. The result is that when the first seat and plug have closed, there still is a small distance remaining for the other plug and seat, and there will be leakage through the valve.

An alternative to the double seated valve is the balanced two-way valve. It has one seat and a single plug with one portion shaped as a piston, which operates inside a cylinder. A hole is drilled from the bottom of the plug to the top of the piston, so the two sides are subject to the same pressure, balancing the forces. Even if there is some leakage between the plug and cylinder, the leakage from a balanced valve is less than from a double seated valve.

Ball valves and butterfly valves are operated by a 90 degree rotary stem motion. When fully open they give a small resistance to the flow in comparison to the valve size. Ball valves give a tight shut-off. Butterfly valves have a disc operating from parallel to the flow (open) to perpendicular to the flow (closed).

2.3.2 Three-way valves

Three-way valves are either globe valves operated with a linear stem movement, or rotary type valves operated with a 90° or 180° movement. The rotary types are ball, shoe and disc three-way valves. The most common valve by far is the globe valve.

Mixing and diverting plug valves have different internal designs. The reason for this is that the force caused by the differential pressure on the plug should be directed away from the seat. If the force is instead directed against the seat, and the plug is operated near the seat, there is a risk that the plug suddenly will be pulled against the seat. The flow is abruptly interrupted, which causes a shock wave and recoil, which lifts the plug from the seat so the flow starts and a new cycle is repeated. The result is a continuous chattering, which can be quite loud. The same is true for two-way valves that are installed in reverse.

All valves should be installed in accordance with the arrows marked on the valve body, indicating the correct direction of the flow.

2.3.3 Control valve sizing

The sizing of control valves is of great importance to the performance of a control system. Too small a valve will not satisfy the maximum flow requirement. Too large a valve will provide the maximum needed flow when it is only partially open. The problem is that only a portion of the range of the valves will be utilized, and the smallest change of the opening of the valves will have a too large an effect, which makes it hard to accomplish stable control.

Therefore valve sizes should be carefully calculated, so the valves pass no more than the desired flow at the available differential pressure.

The Cv coefficient is defined as the number of USGPM of 60°F water that flows through a fully open valve with a 1 psi differential pressure applied across the valve. The following formula is used to calculate the Cv coefficient.

$\Delta P(PSI) = R \times G^n \times d$
R = Hydraulic resistance of the element
d = denisty of the liquid, with d = 1 for water at 60°F
G = The flow of the liquid in USGPM
For turbulant flows, the coefficient "n" is around 1.85.

For a valve,

$n = 2$ and $USGPM = \sqrt{\dfrac{\Delta P(PSI)}{(R \times d)}}$

If we replace $\dfrac{1}{\sqrt{R}}$ by a coefficient Cv, we obtain the formula:

$$USGPM = Cv\sqrt{\Delta P(PSI)/d}$$

$USGPM = Cv\sqrt{\Delta P(PSI)}$ assuming d= 1 for water

Valve Coefficients Cv and Kv

	ΔP in PSI	ΔP in Ft. hd	ΔP in bar	ΔP in kPa
Cv	$Cv = \dfrac{USGPM}{\sqrt{\Delta P(PSI)}}$	$Cv = 1.52 \times \dfrac{USGPM}{\sqrt{\Delta P(Ft.\ hd)}}$	$Cv = 1.16 \times \dfrac{m^3h}{\sqrt{\Delta P(bar)}}$	$Cv = 42 \times \dfrac{L/S}{\sqrt{\Delta P(kPa)}}$
Kv	$Kv = .86 \times \dfrac{USGPM}{\sqrt{\Delta P(PSI)}}$	$Kv = 1.32 \times \dfrac{USGPM}{\sqrt{\Delta P(Ft.\ hd)}}$	$Kv = 1.16 \times \dfrac{m^3h}{\sqrt{\Delta P(bar)}}$	$Kv = 36 \times \dfrac{L/S}{\sqrt{\Delta P(kPa)}}$
Pressure Drop	$\Delta P(PSI) = \left(\dfrac{USGPM}{Cv}\right)^2$	$\Delta P(Ft.\ hd) = \left(\dfrac{USGPM}{.658Cv}\right)^2$	$\Delta P(bar) = \left(1.16\dfrac{m3h}{Cv}\right)^2$	$\Delta P(kPa) = \left(42 \times \dfrac{L/S}{Cv}\right)$
Flow	$USGPM = Cv \times \sqrt{\Delta P(PSI)}$	$USGPM = .658Cv \times \sqrt{\Delta P(Ft.\ hd)}$	$m^3h = .86Cv \times \sqrt{\Delta P(bar)}$	$L/S = \dfrac{Cv}{42} \times \sqrt{\Delta P(kPa)}$

Cv = 1.16Kv   Kv = .66Cv

$$Cv = \frac{USGPM}{\sqrt{\Delta P(PSI)}}, \ \Delta P = \frac{Cv^2}{USGPM}$$

$$USGPM = 0.66 \times Cv\Delta P(ft); \ Cv = \frac{1.5\ USGPM}{\sqrt{\Delta P}}$$

$$\Delta P(PSI) = \left(\frac{USGPM}{0.66 \times Cv}\right)^2$$

The European counterpart of the Cv value is the Kv value, which is based upon the metric system. The following formula is used to calculate the Kv value formula.

$$Kv = \frac{m^3/hour}{\sqrt{\Delta P(bars)}}$$

$$Cv = 1.16Kv$$

$$Kv = .86Cv$$

The manufacturers of control valves provide charts and tables showing the relationship between Cv values, flow and differential pressure. These offer a convenient alternative to the formulas, because the charts also indicate which Cv values are available. In order to calculate the Cv value, the maximum flow and corresponding differential pressure must be known. Information about the flow is easily available data from the controlled heat exchanger or coil. The differential pressure should only be determined after an analysis of the hydronic system.

However, the following can be used as a general rule. The total differential pressure is distributed among the different elements (control valve, coil, balancing valve, shut-off valve, etc.) that are connected between supply and return. A modulating two-way control valve should be responsible for as large a portion of the differential pressure (pressure drop) as possible. At least 50% of the total differential pressure should occur across the control valve, in order to make a good control function possible. This rule also applies to modulating three-way valves, but under certain circumstances other criteria can be applied. When on/off control is used, any valve size can be used, as long as it is large enough to pass the needed flow.

2.3.4 Valves in parallel

The total Cv value of valves piped in parallel is arrived at by adding the Cv values of the valves.

2.3.5 Close-off pressure

The maximum close-off pressure is the highest pressure a valve can close against. As the differential pressure increases, the force (globe valves) or the torque (rotary valves) required to close the valve increases. The maximum close-off pressure is the highest pressure the actuator is capable of, in order to operate a given valve size. The larger the valve size, the smaller the maximum close-off pressure will be.

2.3.6 Maximum operating pressure differential

In order to limit the maximum velocity through a valve, the specified maximum differential pressure must not be exceeded when the valve is open. A velocity that is too high may cause vibrations, erosion and noise.

2.3.7 Leakage

The leakage through a valve is expressed as a percentage of the Cv value. For example, a double seated two-way valve with a Cv value = 100 and a leakage of 0.5% has a leakage corresponding to a Cv value of 0.5. This may not sound like much, but if the close-off pressure is = 16 psi, as much as 2 USGPM will leak through. A coil utilizes small flows very efficiently, so the losses can be substantial. Leakage will also occur if an actuator is improperly adjusted or if it cannot provide a force strong enough to close the valve.

2.3.8 Normally closed or normally open valves

Two-way valves can have a "trim" (plug and seat) that closes when the stem is pushed down. This is referred to as a "Normally open" valve. A valve that opens when the stem is pushed down is referred to as a "Normally closed" valve.

2.3.9 Problems caused by over-sized control valves

A control valve that is larger than needed will:
- Cause control problems (Hunting)
- Have a needlessly low close-off pressure
- Have a larger leakage
- Cost more
- May require a larger and more expensive actuator

2.4 Valve cavitation

Cavitation is the forming and imploding of vapor bubbles in a liquid due to decreased and increased pressure as the liquid flows through a restriction. When the pressure increases above the vapor pressure, the bubbles implode, which creates very high shock waves which erode the surrounding material and cause an intense noise. This is what can happen inside a valve between the plug and seat, under certain conditions.

Cavitation depends upon the design of the valve, the pressure before (P1) and after (P2) the valve and the vapor pressure (Pv).

2.5 Valve characteristics

The valve characteristics define how a valve changes the flow when the stem is operated 0% to 100% and the differential pressure is constant. The flow is controlled by specially contoured plugs or ports. In the following text the different shapes of ports are used as example only.

There are several common valve characteristics.

Quick opening
A disc is operated against a seat, and as soon as the disk begins to lift the flow increases rapidly.

A 10% opening can give more than 50% of the maximum flow. This type of valve is used for on/off control, and provides large Cv values relative to the size of the valve body.

Linear
There is a linear relationship between the variation of the flow, and the stem movement, 0% to 100%. The port has parallel sides and the exposed area is varied in proportion to the stem movement.

Quadratic
The plug has a V-shape, with straight sides which varies the exposed area of the opening quadratically.

For example, when the valve is 30% open the flow is 9%, 50% open the flow is 25% and at 70% open there is 49% flow. At 90% open the flow is 81%. The quadratic characteristic is a compromise between linear and equal percent.

Equal percent.
It has a more pronounced curvature than the quadratic characteristic, and therefore requires a V-port with specially shaped sides instead of straight sides. The equal percent characteristic gives an equal percentage change over the previous flow for equal increment of stem movement. For example, a 30% opening could produce 10 USGPM, and an additional 10 percentage point increase in the opening to 40% increases the flow to 16 USGPM, which is a 60% increase. If there is then an additional 10 percentage point increase of the opening to 50%, there will be a 60% increase over the previous flow, from 16 USGPM to 25.6 USGPM.

This characteristic compensates and counteracts the non-linear heat emission curve of coils and heat exchangers, and is therefore used for modulating control.

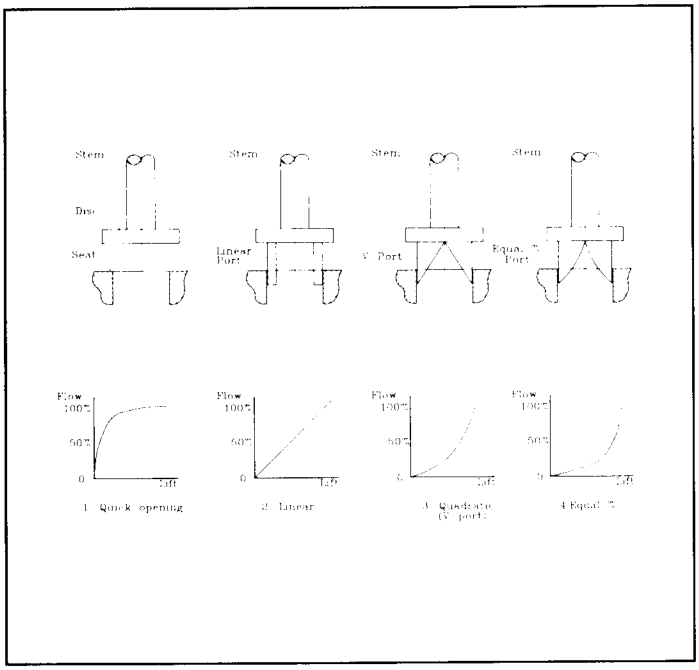

Fig. 2 Control valve seats

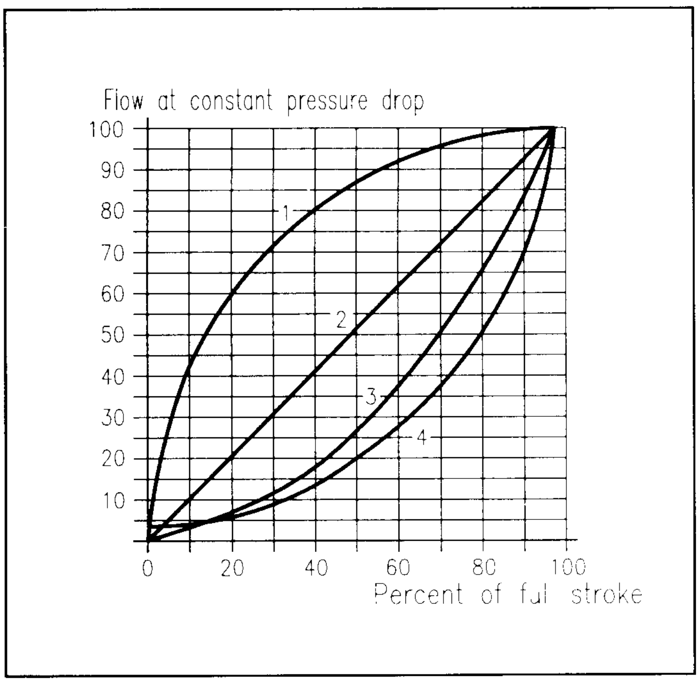

Fig. 3. Typical flow characteristics of valves
1. Quick opening 2. Linear 3. Quadratic
4. Equal percentage

The flow characteristic is defined as a constant differential pressure across the valve. However, when the valve is installed, the differential pressure will change as the flow changes. This will distort the flow characteristic, so it no longer has exactly the same shape as the theoretical curve.

Figure 4a shows a control valve piped in series with other elements such as coil, pipes, hand valve and a balancing valve.

Figure 4b shows how the differential pressure is distributed between the elements of the circuit when the valve is fully open. In the following text the differential pressure between the supply and return is constant regardless of the flow.

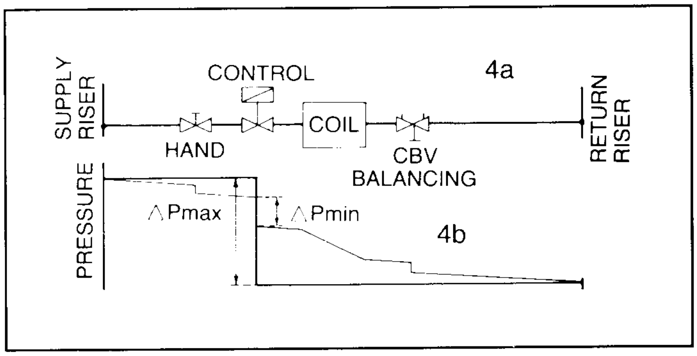

Fig. 4. Differential pressure across control valve

Other characteristics

Most three-way valves made in North America are symmetrical, which means that the two control ports have the same characteristics. Linear/Linear or Equal/Equal %. This makes it possible to pipe the valve so any one of the ports is either the control port or the by-pass port.

If the characteristic is Linear/Linear the resistance in the common port will be constant, but Equal/Equal % valves will have a resistance to the flow through the common port that varies with the position of the stem. At the 0% and 100% positions, the resistance will be "normal" and correspond to the Cv value, but in the mid position (50%) the resistance will be much higher. This is because the equal percent characteristic produces about a 13% flow in each port when 50% open. The total flow will be 26% instead of 100% which was expected.

Some of the European made three-way valves are asymmetrical, Linear/Equal%. This reduces the problem with a variable resistance in the common port. In the mid position the flow will be 50% + 13% = 63% instead of 26%. This also means that the control port and the by-pass port cannot be interchanged. In addition, the control port is the upper port and the bottom port is the by-pass port, which is the opposite to what is typical for most installations.

The domestic valves often, instead of having an Equal% characteristic, have a quadratic/quadratic characteristic, which means that the flow in the common port is 25% + 25% = 50% in the mid portion, (instead of 26%). This pragmatic approach is sacrificing a little of the flow characteristics to gain flexibility, and the possibility of mistakes is reduced. Three-way valves are used to provide a constant flow in the system. It may seem that the variable resistance in the common port will defeat this purpose, but by sizing a three-way valve with this problem in mind, the effect of the variations in resistance can be significantly reduced, and a relatively constant flow is accomplished.

2.6 Terminal heat emission

ASHRAE Systems and Application Handbook (1991) on basic water system design points out that the sensible heat transfer characteristic of heat exchangers is generally non-linear with changes in flow. (Ref: B-2 Chapter 34)

In Figure 5 shown here, a water to air coil characteristic is shown: % heat emission vs. % water flow. These curves were prepared from an actual coil design based on.

Entering Air Temperature = 60°F
Entering Water Temperature = 200°F
Water Temperature Drop = 20°F

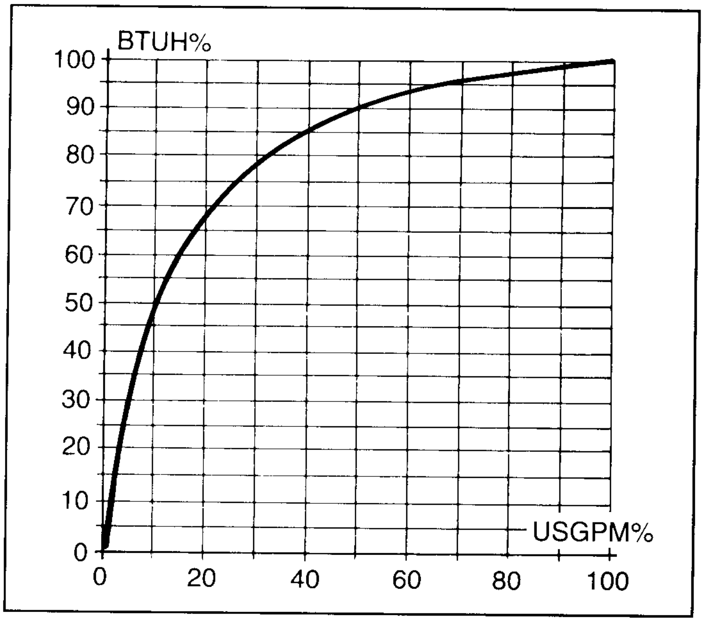

Fig. 5. Heat emission of typical heating coil vs. water flow

This curve is similar to a typical curve shown in the ASHRAE 1991 Systems and Applications Handbook Chapter 34.7 and this characteristic is used as an example.

When examining the curve characteristics in Figure 5, notice that a decrease in water flow by 50% creates only a 10% decrease in the coil heat emission. The flow must be decreased by 90% to obtain a 50% decrease. A flow reduction from 10% to 0% decreases heat emission from 50% to 0%.

In many cases a coil's performance cannot be calculated with an accuracy better than 10%. It can then be considered that the overflow variations are in fact without consequences and that hydronic balancing might appear as an unsound or nonexistent problem.

This conclusion can also be extended to other hot water heat transfer terminals. Figures 6 and 7 represent the emission as a function of the water flow for a convector-radiator and a radiant floor heating system, respectively, at various temperature drops.

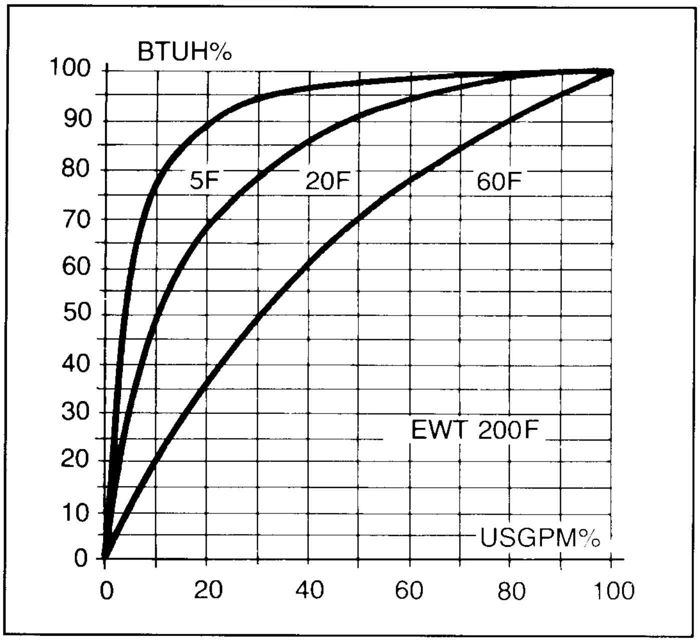

Fig. 6. Heat emission of typical convector vs. water flow for various temperature drops

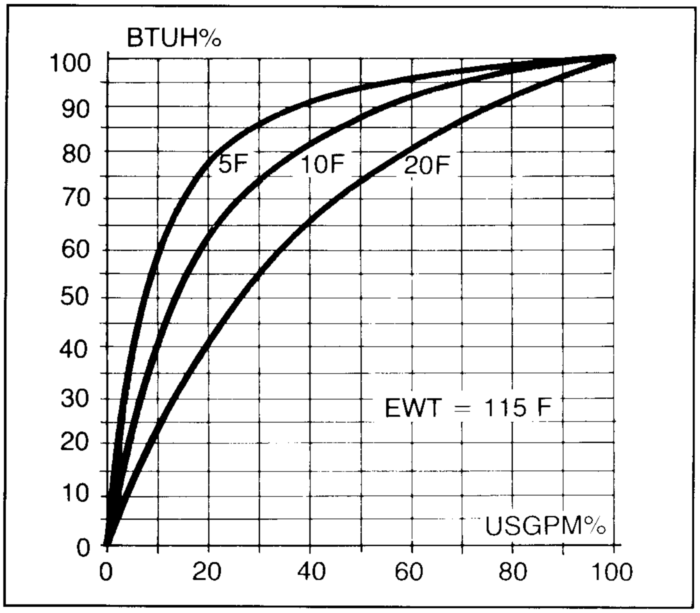

Fig. 7. Heat emission of typical radiant floor vs water flow for various water temperature drops

This lack of sensitivity between the terminal emission and its water flow is significant when the terminal emission is calculated for a small water temperature drop of 5°F as shown in Figure 8 where the coil emission shown in Figure 6 is examined for three water temperature drops.

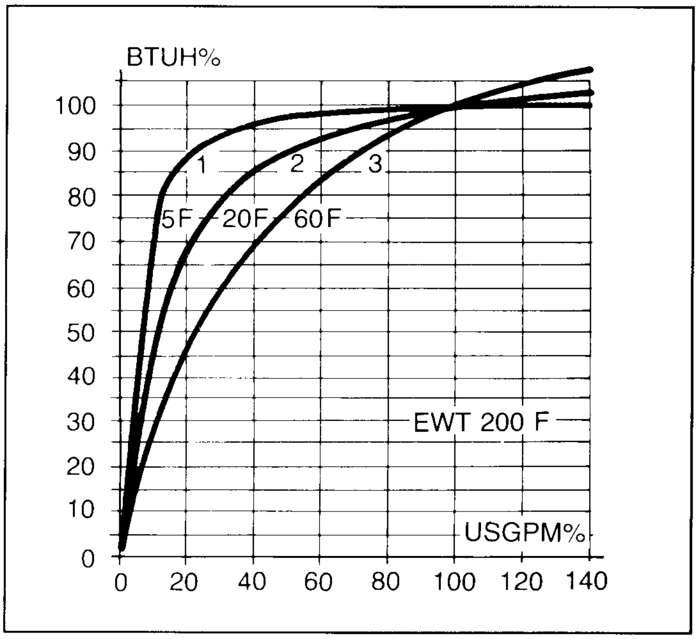

Fig. 8. Typical heating coil emission for various water temperature drops vs. flow

Working with small water temperature drops has some disadvantages:

1. Increased water flows are necessary, requiring
   - larger pipe sizes
   - larger pumps and higher pumping costs.
2. When the system is not balanced, some terminals work with such low and variable flows that emissions are hard to predict.
3. Proportional control of terminals may be impossible for small loads because minor increases in the water flow cause extremely large increases in the emission rate.

In fact, in most heating installations, the actual load is usually 50% below the nominal output, due to part load conditions, and the flow must be controlled below 10% of the nominal flow.

A cooling coil behaves similarly to a heating coil when studying its sensible heat emission versus flow. Figure 9 shows this case.

From a temperature-control point of view, the curve characteristic is the sensible heat curve which has the same general form as that for a heating coil. Total heat includes the addition the latent heat removal (moisture).

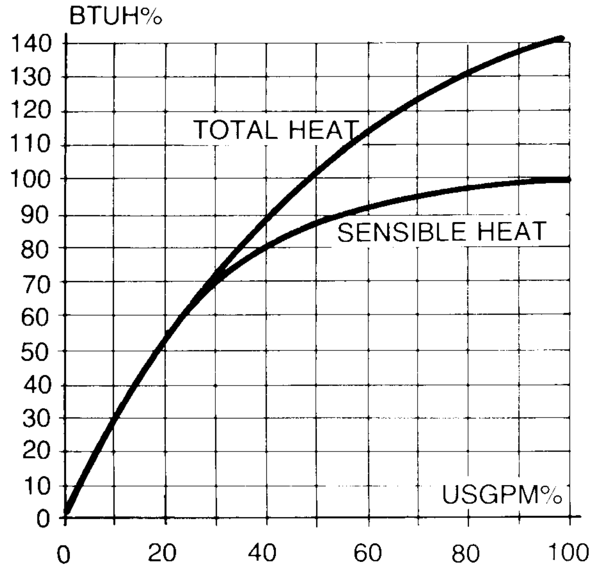

Fig. 9. Cooling coil emission for total and sensible loads vs. water flow

Characteristics shown in Figure 9 are obtained for a coil under certain conditions with the following design criteria.

(CHWS = 45°, WTR = 10°F; EAT = 80 DB/67 WB)

In most applications, a cooling coil is more sensitive to the water flow in comparison with a heating coil due to smaller temperature drops with chilled water. Hydronic balancing is therefore even more critical in cooling than in heating.

However, under the same conditions of entering water temperature and water temperature drop (for instance, EWT = 200°F and WTD = 20°F) certain heating coils are more sensitive to water flow and the above conclusion cannot be generalized.

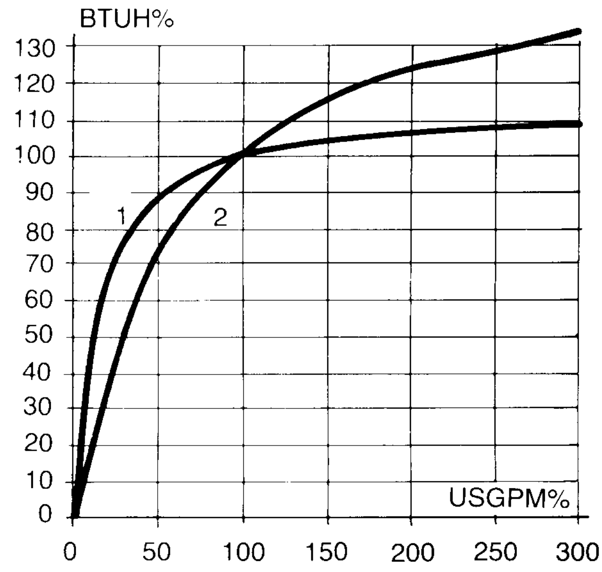

Fig.10. Typical heating coil emission vs. hot water overflows

Figure 10 shows heating coil No. 1 from Figure 5 subjected to flows in excess of the maximum (100%) to show the effect of heat emission of twice (200%) and three times (300%) of flow rate. Note that with 200% flow the emission increases by 8% (100 to 108%); at 300% flow the emission increased by 10% (100 to 110%). Coil No. 2 is a different heating coil design and shows an emission increase of 35% (100 to 135%) with a 300% overflow.

2.7 Control of terminal emission

A two-way or three-way control valve regulates the water flow in a coil to obtain the necessary emission. This emission depends on the coil's characteristics, which are the entering water temperature, EWT, the entering air temperature, and the water flow in USGPM. Figure 11a is a reproduction of the curve emission of Figure 5. The relation between the emission and the water is not linear, i.e. 10% of flow equals 50% of emission rate.

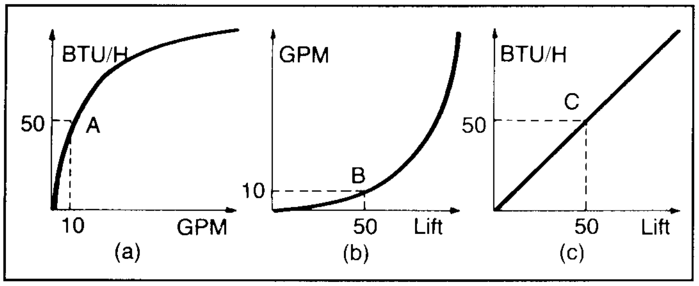

Fig. 11. Combination heating coil and control valve characteristics

With small loads, the emission is very sensitive to water flow, and control is very difficult in this region. To obtain stable control, we have to select a valve characteristic which compensates for the coil characteristic. The valve characteristic is the relation between the water flow and the valve opening for a constant differential pressure applied on the valve (Figure 11b).

In Figure 11a, we see that the emission is 50% when the water flow is only 10% (point A). The valve characteristic is chosen in such a way that the water flow will be 10% when the valve is open 50% (point B). In this case, we obtain 50% emission when the valve is open 50% (point C). This specific characteristic shown on Figure 11b is similar to a typical valve characteristic called equal percentage.

The emission of heating or cooling coils can be controlled by proper selection of each of the following:

1. Control valve size
2. Control valve characteristics
3. Balancing valve size
4. Pressure drop of control valve versus. balancing valve

Our objective is to provide a linear coil output change with each control valve lift position (Figure 11c), so there is a proportional emission change with each proportional signal change from the temperature controller to the valve. In these conditions the stability of the control loop does not depend on the load.

2.7.1 Control valve authority

When the control valve is closed, the differential pressure across the control valve ΔPMax is the same as the differential pressure between the supply and return. When the valve is fully open the differential pressure ΔPMin is reduced because of all the pressure drops across the elements in series with the control valve. This pressure variation is expressed as a ratio ß:

The ratio $ß = \dfrac{\Delta PMin}{\Delta PMax}$ is called the valve authority.

Figure12 shows how a linear flow characteristic is distorted by the other elements in series with the valve ß

Example : The control valve in figure 12 has a linear characteristic and a Cv value of 31.6. The differential pressure between supply and return is 16 psi. When the control valve is fully open the total resistance of all the elements (have been adjusted to) limit the flow to 40 USGPM.

All of the differential pressure is applied across the control valve when it is fully closed. This means that the flow begins to increase at a very high rate and if it could continue at the same rate would reach

$$31.6 \text{ x } \sqrt{16} = 126.4 \text{ USGPM or}$$

$$Cv \text{ x } \sqrt{\Delta P(PSI)} = \text{USGPM}$$

when the valve is fully open. However, as the flow is increased, the influence of the coil, balancing valve, etc., gradually increases so the curve is deflected more and more, resulting in a flow of only 40 USGPM when the valve is fully open. The differential pressure across the fully open control valve is:

$$\Delta P \text{ min} = \frac{(Q)^2}{Cv^2}$$

$$\Delta P \text{ min} = \frac{40^2}{31.6^2} = 1.6 \text{ psi}$$

$$\text{Valve authority } \text{ß} = \frac{\Delta P \text{ min}}{\Delta P \text{ max}} = \frac{1.6}{16} = .1$$

There is another quite interesting way of calculating the valve authority.

The maximum flow, 40 USGPM, and the differential pressure, 16 psi, is known, so it is easy to calculate the total flow coefficient Cv is $\frac{40}{\sqrt{16}} = 10$ for the control valve, coil, balancing valve etc. The control valve has a flow coefficient Cv value of 31.6.

$$\text{Valve authority } \text{ß} = \frac{Cv^2 \text{ total}}{Cv^2 \text{ valve}} = \frac{10^2}{31.6^2} = .1$$

The interesting aspect about this formula is that there is a quadratic relationship between the Cv value of the valve and the valve authority. This means that an oversized control valve will seriously affect the valve authority, and thereby distort the valve characteristics.

It may seem that a valve authority of .1 is not a realistic example, but consider this; if a control valve with a Cv = 10 would have produced a valve authority ß=.5, but instead a valve with a Cv = 22.4 is installed, the valve authority will drop all the way down to ß =.1, which is unacceptable. This is partly due to the fact that if a larger valve is used, in order to prevent overflow, the balancing valve has to be adjusted for a higher resistance, so the maximum flow is not exceeded. The result is that the pressure drop is shifted away from the control valve to the balancing valve. Unfortunately, it is not uncommon to find installations with control valves that are twice as large as they should be.

Figure 12 and 13 show how a linear and an equal percent valve characteristic is distorted as a function of different values of the valve authority. The primary objective with valve sizing is to accomplish the best valve authority possible. It is usually very difficult to accomplish a valve authority ß = 1 but as can be seen from the figures, an authority ß =.5 is quite acceptable, because it produces only a modest distortion. *A control valve should therefore be sized so it produces a pressure drop, when it is fully open, which is at least 50% of the total differential pressure between supply and return.* In other words, the pressure drop across the control valve should be at least as large as the pressure drop across the coil plus fittings and balancing valve.

Figures 12 and 13 show the distortion of the characteristics as a function of the authority and for linear and equal-percentage valves respectively.

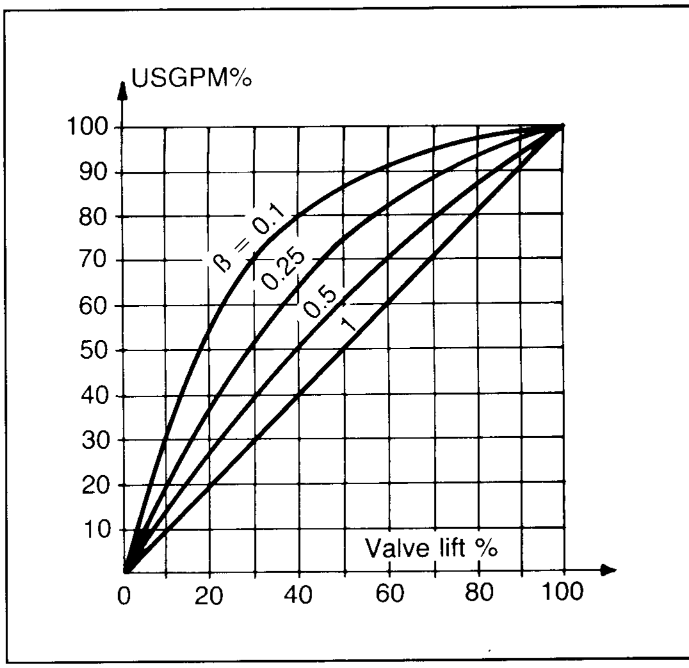

Fig. 12. Distortion of a linear characteristic

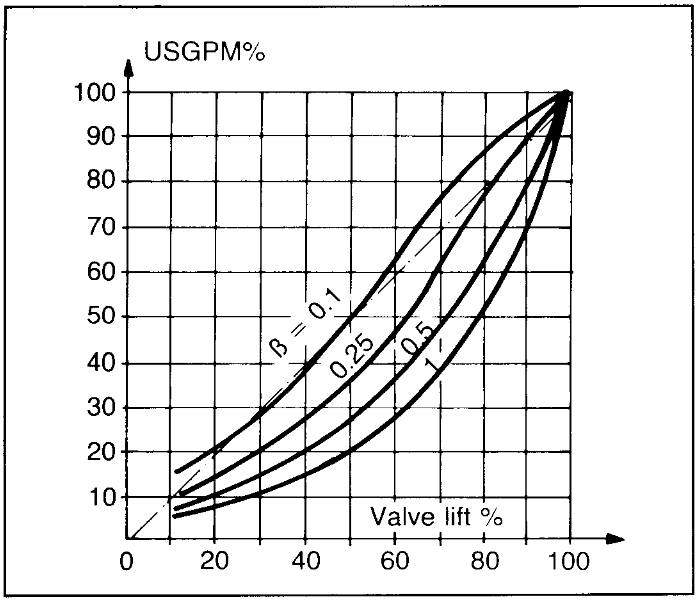

Fig. 13. Distortion of an equal percentage characteristic

We generally adopt values of ß greater than or equal to 0.5. It means the pressure drop across a fully open control valve should at least equal the pressure drop of the variable flow circuit.

The total pressure drop in the circuit should not be more than twice the pressure drop across the control valve. If too large a valve is chosen, control is likely to be difficult because the valve works over a limited range of its travel.

It is not advisable to reduce the maximum flow by fitting a balancing valve in series unless the authority ß remains within reasonable limits.

The best solution is to replace the control valve with another of more suitable size. If it is not possible, despite the authority factor, the maximum water flow has to be reduced with a balancing valve in series.

For "On/Off" control valves, the authority factor is not significant since only the open and closed positions are considered.

An oversized proportional motorized valve may be compensated at the limit by a balancing valve fitted in parallel with the coil. The total water flow increases to meet the real sizing of the control valve. This curious solution does have the dual advantage of limiting the water flow through the coil to the desired value and of improving the authority factor ß of the control valve

2.7.2 Rangeability

The plug has two parts that perform different functions. One part is specially contoured to control the flow in accordance with the characteristics of the valve. The other part provides the shut-off against the seat.

The flow is controlled in accordance with the flow characteristic of a valve, by the variation of the free area between the contoured parts of the plug and seat. However, near the closed position a sudden transition takes place in the control of the flow, from the contoured part to the shut-off part of the plug. The result is that the flow can only be controlled in accordance with the characteristics, down to a certain minimum flow, which is referred to as the "Minimum controllable flow".

The rangeability factor (RF) of a control valve is the ratio between the maximum flow for a fully open valve, and the minimum controllable flow, under a constant differential pressure.

$$RF = \frac{\text{Maximum flow}}{\text{Minimum controllable flow}}$$

The rangeability is an inherent factor of the valve itself, (regardless of the rest of the control system) and it must not be confused with the resolution of the actuator or the performance of the controller. The factor that determines the rangeability is the minimum clearance between the contoured plug and the seat. Not only is there a question of how closely the plug and seat can be machined, but consideration must be given to the risk of the plug sticking in the seat. Therefore normal control valves have a rangeability factor RF of about 30. This means that the minimum controllable flow is about 3.3%.

OG = overflow factor

GM% = Minimum control water flow as a % of the maximum flow

QM = BTUH %.

Moreover, if the circuit is not balanced, the maximum water flow is, for instance, OG (overflow factor) times greater than design and the minimum flow calculated by the same factor OG. If, for example, OG = 2, the real rangeability of the valve becomes

$$RF3 = \frac{RF2}{OG} = \frac{21.2}{2} = 10.6$$

When we experience small WTD, a low valve authority and improper hydronic balancing, it is practically impossible to control the coil's heat output in a proportional way. If, in addition, the coil is oversized, you can imagine what will occur!

To obtain good control several actions are required, i.e.:
1. Calculate the control valve to obtain a minimum authority of 0.5
2. Balance the system to avoid water overflows
   - correct choice of the Cv value of the two way valves
   - use three-way valves to allow balancing valves in series without changing their authority
   - stabilize the differential pressure on the branches
   - avoid hydronic interferences with secondary pumps
3. Avoid oversizing coils
4. In heating, reset the inlet water temperature according to the outdoor temperature to prevent the control valve operating near its closed position
5. Select a control valve with adequate rangeability and with actuator having small resolutions to take advantage of the valve characteristic

A way of improving the total rangeability is to pipe two valves in parallel, and to operate them in sequence. The smallest valve opens first, and the second valve will not begin to open until the first is fully open. The Cv values of the valves are 1/3 and 2/3 respectively of the total required Cv value. This solution is sometimes needed where very large load changes are encountered, such as the main air handler in VAV-systems.

2.8 Actuators

Pneumatic, electric and self-contained (thermostatic) are the most common actuators. A linear or rotary operation is provided depending upon the type of valve. The actuator must be powerful enough to provide a sufficient force or torque to operate the

Authority	Overflow factor OG = 1			Overflow factor OG = 2		
ß	Gm%	Qm ($\Delta T=20$)	Qm ($\Delta T=60$)	Gm%	Qm ($\Delta T=20$)	Qm ($\Delta T=60$)
1	3.33	21.6	8.4	6.66	36.4	16.0
0.5	4.71	28.3	11.6	9.43	45.4	21.7
0.1	10.54	48.5	23.9	21.08	68.1	41.6

Fig. 14. Rangeability

needs to be considered. There are some electrical actuators that provide only on/off action, but most actuators can modulate the opening of a valve.

2.8.1 Resolution

A modulating actuator operates the valve in fine increments, in response to an in-signal coming from a controller. The smaller the increments are, the higher the resolution. A high resolution is desirable, specially when the valve is operated near the closed position, where each increment can cause a relatively large change in the heat emission.

The operation of pneumatic actuators can be improved significantly by using "position relays," which control the positioning of the valve so it exactly corresponds to the in-signal, thereby eliminating the influence of variations in the differential pressure across the valve and the friction in the box packing. The additional cost of a "position relay" is fully justified on all valves, except the smallest.

2.8.2 Automatic control valves

Automatic control valves provide dynamic balancing of a hydronic system so it actively responds to changing conditions. The valves are self-contained, and depending upon the type, automatically control either the flow or the differential pressure between the supply and return.

The pressure control valve has an actuator consisting of a chamber, divided in two halves by a diaphragm. One half has a thin pipe connected to the supply riser, and the other half is connected to the return riser. The differential pressure acts upon the surface area of the diaphragm and creates a force which acts against a spring.

The stem of a valve is connected to the diaphragm, so when the differential pressure increases, the spring is compressed and the valve closes. This reduces the differential pressure until the forces from the diaphragm and spring are in balance. The desired differential pressure can be adjusted by varying the spring tension.

The flow controlling valve consists of a hollow plunger, which has slots through the sides. The plunger is spring loaded, and it operates in and out through an orifice. When the flow is low, the spring pushes the plunger out through the orifice, against the flow. All the slots are exposed and the resistance to the flow is low. When the flow increases the differential pressure increases, as does the force acting upon the plug. As the plug is pushed inside the orifice, fewer slots are exposed, and the resistance to the flow increases. The variation of the resistance and the variation of the spring force is matched so a constant flow is accomplished over a wide operating range.

A conventional balancing valve can adjust to whatever value is needed, but the automatic valve lacks flexibility. If, in the future, a new control strategy is implemented, – for example, resetting the speed of the main circulating pump versus the outdoor temperature, – the automatic valves will be wide open as soon as the speed begins to be reduced. This means that the system will be without any balancing.

True balancing valves, on the other hand, continue to distribute the flow in the same proportion as before, even when the flow is reduced.

The automatic valve cannot shut off the flow, so a separate shut-off valve must be installed. This is an additional expense that is avoided with a conventional balancing valve which also doubles as shut-off valve.