Pressure-Independent Control Valves vs. High-Performance Control Valves

Balancing and Control Valve Sizing for Direct-Return, Variable-Flow Hydronic Systems

This technology report is primarily directed to the consulting engineering community and other technical advisors representing building owners. It is intended to be an objective summary of the findings from research conducted by Siemens Building Technologies on balancing and valve sizing for variable-flow hydronic systems. This report examines the balancing and valve sizing issues of pressure-independent control valves versus high-performance, pressure-dependent control valves.

Self-Balancing Hydronic Systems

There is a growing consensus in the HVAC systems design community that if a hydronic system is properly designed, and control valves and actuators are properly sized and selected, then the system is self-balancing and does not require mechanical balancing valves. Elimination of mechanical balancing valves enhances controllability of the heat transfer coil and reduces pumping energy costs. Properly sized and controlled two-way, pressure-dependent, high-performance control valves will adequately balance the system such that at least 80% of the valve stroke is available for coil control. Further, if only the maximum flow and pump head are known, a high-performance, pressure-dependent control valve can be effectively sized such that it will properly balance the system and have sufficient stroke available for precise coil control. Pressure-independent valves are relatively easy to size properly, and if installed on every circuit of a water distribution system, will eliminate the need to know the distribution system branch design flows and pressure losses needed to most accurately size a high-performance control valve. However, the first cost of these valves is as much as four to five times greater than that of a comparably-sized pressure-dependent control valve.

Balancing Variable-flow Hydronic Systems

How, and indeed whether, to balance variable-flow hydronic systems with two-way valves has been debated in the industry for many years (Avery, et. al. 1990, Avery 1993, Hegberg 1997, Taylor 2002).1,2,3,4 The ASHRAE 2003 HVAC Applications Handbook5 continues to recommend two basic methods for balancing hydronic systems, including those that are variable-flow:

- Balancing by temperature difference between supply and return water at the terminals.
- Balancing the circuits in the system proportionately according to the actual flow rate divided by the design flow rate through the circuit.

However, after a careful review of the literature, there appears to be a growing consensus among HVAC system designers that if a hydronic system is properly designed, and control valves and actuators are properly selected, and the sensor controlling the coil valve is always in control of the waterflow through the coil, then the system is self-balancing and does not require additional balancing valves (Avery 1990, 1991, Bell & Gossett 1965, Haines 1991, Hansen 1991, Stethem 1990, Taylor 2002, 2003).

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5 See Chapter 37, pp. 37.8 through 37.10.
Utesch 1991). Avery (1990) goes so far as to say that balancing a variable-flow hydronic system will “ruin the control system.”

One of the great concerns with the use of balance valves for balancing water circuits is the pressure drop across these valves, and the resultant additional pump energy consumption. This extra pump energy consumption can be calculated approximately as:

\[ \Delta P = \frac{(\text{pump GPM})(\Delta p_{\text{bal valv}})}{3960 \eta_{\text{wwe}}} \]  

(1)

\( \Delta P \) is the additional pump horsepower.

\( \Delta p_{\text{bal valv}} \) is the pressure drop across the balance valve in feet of head.

\( \eta_{\text{wwe}} \) is the wire-to-water efficiency of the pump. In addition to the pumping energy saved, chilled water \( \Delta T \) and effective plant cooling capacity increases when balancing valves are eliminated. Anecdotal evidence (Rishel 1997) indicates that these savings can be substantial.

System waterflow balance is affected by the following system components:

- Piping circuit
- Pump or pumps
- Control valves
- Coil pressure drops and design \( \Delta T_s \)
- Chiller or boiler pressure drops and design \( \Delta T_s \)
- Balance valves (if used)

The hydronic system designer should select these components carefully so that when assembled into a working system, they work together to provide an adequately balanced system. Evaluations should be made for pressure drops of all of the above components.

### Balance Valves in Hydronic Systems

Rishel (1997) states that certain types of chilled water systems should use balance valves, while other types do not require them, and that the pressure gradient diagram demonstrates which chilled water systems should use balance valves. The pressure gradient diagram is constructed by plotting the pump head in feet on the vertical scale and the distance from the pump on the horizontal scale. The horizontal scale on the pressure gradient diagram is used to separate the various pressure loss elements in the circuit. Figure 1 illustrates a pressure gradient diagram for a variable-primary-flow chilled water system. If the pressure drop designated as “chiller loss” in the figure is removed, this figure could also apply to a secondary chilled water loop of a primary-secondary chilled water system.

Balance valves were used extensively in the past on hydronic systems with constant-speed distribution pumping and three-way control valves on the coils. This was a proper use of balance valves for these systems. However, due to their energy waste, three-way control valves should not be used in today’s chilled water systems; they should be replaced with two-way control valves and the distribution pumping should be variable speed whenever economically possible.

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8 Pump wire-to-water efficiency, \( \eta_{wwe} \), can be written as the following equation: \( \eta_{wwe} = \eta_p \times \eta_m \times \eta_v \), where \( \eta_p \) is the pump efficiency, \( \eta_m \) is the pump motor efficiency, and \( \eta_v \) is the pump speed drive efficiency.


Rishel (1997)\textsuperscript{11} demonstrates that it is impossible to adjust manual balance valves on variable-volume, direct-return chilled water circuits with modulating-type coil control valves. Therefore, manual balancing valves should not be used on these circuits. Rishel recommends manual balance valves be used on reverse-return systems with two-position control valves while automatic balance valves be used on direct-return systems with two-position control valves.

Using balancing valves on variable-flow hydronic systems is detrimental to the control system performance and energy use because it reduces the authority\textsuperscript{12} of the control valve and adds a permanent restriction on every branch circuit (Avery 1993).\textsuperscript{13} Therefore, balancing valves should not be used on direct-return variable-flow hydronic systems with two-way control valves.

**Balancing Options**

Most recently, Taylor (2002\textsuperscript{14} and 2003\textsuperscript{15}) analyzed eight of the most commonly used methods for balancing hydronic systems. Two hydronic distribution systems were analyzed, one chilled water system and one hot water system, both of which were based on a real building. For each of the balancing options, flow through the system was analyzed using a commercially available pipe network analysis program that modeled how the flow and pressure vary throughout the system. Both first costs and balancing costs of each option were estimated.

The eight balancing options studied were:

1. No balancing. The rationale behind this option is that if the coils can achieve their discharge air setpoint at or below the design flow of the coil, then the control valves themselves will automatically and dynamically balance the system. Under this option, neither balancing valves nor balancing labor is required.

2. Manual balance using calibrated balancing valves. Calibrated balancing valves have flow measurement capabilities integrated into the design of the valve. Flow is measured and valves are adjusted by the test and balance contractor to achieve design coil flow rates.

3. Automatic flow limiting valves. Automatic flow limiting valves (also called automatic flow balance valves) are self-powered devices that limit flow to a preset value when the differential across the valve is within a certain range. Typically, the valves include a cartridge with specially shaped orifices controlled by a spring. As the differential pressure across the valve depresses the spring, a varying amount of orifice area is opened. The area and shape of the orifices are designed to deliver a constant flow rate, within the limits of the spring.

4. Reverse-return piping. The effect of reverse-return piping causes the differential pressure across each terminal unit (coil and control valve combination) to remain constant.

5. Oversized main piping. This option attempts to equalize the differential pressure across each sub-circuit by reducing the pressure drop of the piping mains by keeping the mains the same size for the entire length of the system.

6. Undersized branch piping. This option is similar to option 2, except that pipe size is reduced to increase pressure losses in branches instead of adjusting a valve.

7. Undersized control valves. This option is similar to option 6, except control valves are undersized.

8. Pressure-independent control valves.

\textsuperscript{11} Ibid.

\textsuperscript{12} Valve authority will be defined and discussed later in this report.


Option 1, no balancing, requires further explanation.

No Balancing (Option 1)

Two questions arise with this design:

1. What happens when control valves cannot meet setpoint and are wide open, as occurs during transients such as warm-up (heating systems) or cool-down (cooling systems) when setpoints are beyond attainable levels, or if the coils are simply undersized?

2. What is the impact on controllability caused by control valves having to be partly closed at design flow?

The computer simulation of this option showed that when all the control valves are wide open, the coils hydraulically closest to the pump had a flow rate substantially above design (143% for chilled water and 212% for hot water), while those farthest from the pump had flow rates approximately 25% below design.

For equal percentage valves selected with an authority\(^16\) of 0.5, in the case for chilled water systems, for those coils closest to the pump, the valve would have to close to about 85% of full open to reduce the flow to design flow, and for hot water systems, the valve would have to reduce to 75% of full open to reach design flow. It is unlikely that reducing the effective control range of the valve by 15% to 25% will cause problems if the valves have been properly selected, particularly with today’s controllers and actuators. However, for systems larger than the ones studied, reducing excess flow for coils near the pump may require further reductions in valve stroke. The more pressure the control valve has to absorb to achieve design flow rates, or the larger the difference between the pressure drop through the hydraulically farthest circuit and that through the closest circuit, the more likely control problems will result from direct-return systems using control valves for balancing. Therefore, for large systems, or those zones (control valves) that experience a high degree of pressure variation, other balancing options described later should be considered.

For those coils farthest from the pump, it is also unlikely that the reduced flow will cause problems. This is due to the inherently non-linear nature of coil heat transfer characteristic (Figure 2 and Figure 3). In the case of cooling coils, a reduction in flow to 75% of design flow still results in a coil heat transfer rate that is 89% of design, while for heating coils, a reduction in flow to 75% of design results in a coil heat transfer rate that is 96% of design.

Summary and Conclusions of Balancing Options

Table 1 lists the advantages and disadvantages of each balancing option. Taylor (2002)\(^17\) concludes that the “no balancing” option is the best, except for very large distribution systems, because it combines low first costs with minimal operational problems.

In Taylor’s 2003 presentation on this topic,\(^18\) he concludes with the following recommendations:

- Automatic flow-limiting valves and calibrated manual balancing valves are not recommended on any variable-flow system with modulating, two-way control valves. There are few advantages and high first costs.
- Reverse-return and oversize mains may have reasonable pump energy savings payback on 24/7 chilled water systems.
- Undersized piping and valves near pumps improves balance and costs are reduced, but significant added engineering time is required.
- Pressure-independent control valves should be considered on very large systems for coils near pumps. The valve first costs are high but are coming down now due to competition.
- For other than very large distribution systems, no balancing (option 1) appears to be the best option.

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\(^{16}\) Valve authority will be defined and discussed later in this report.


<table>
<thead>
<tr>
<th>Balancing Option</th>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 No balancing</td>
<td>• No balancing labor &lt;br&gt; • Low first cost and energy use &lt;br&gt; • Coils may be added/subtracted without rebalance</td>
<td>• Imbalance during transients or if setpoints are improper &lt;br&gt; • Control valves near pumps can be over-pressurized, reducing controllability</td>
</tr>
<tr>
<td>2 Manual balance using calibrated balancing valves</td>
<td>• Valves can be used for future diagnosis (flow can be easily be measured) &lt;br&gt; • Reduced over-pressurization of control valves close to pumps</td>
<td>• Added cost of calibrated balancing valve &lt;br&gt; • High balancing labor cost &lt;br&gt; • Complete rebalance may be required if coils added/subtracted &lt;br&gt; • Slightly higher pump head due to balancing valve &lt;br&gt; • Coils may be starved if variable-speed drives are used without ΔP reset &lt;br&gt; • Slightly higher pump energy depending on flow variations and pump controls</td>
</tr>
<tr>
<td>3 Automatic flow limiting valves</td>
<td>• No balancing labor &lt;br&gt; • Coils may be added/subtracted without rebalance</td>
<td>• Added cost of strainer and flow limiting valve &lt;br&gt; • Cost of labor to clean strainer at start-up &lt;br&gt; • Higher pump head and energy due to strainer and flow limiting valve &lt;br&gt; • Valves have custom flow rates and must be installed in correct location &lt;br&gt; • Valves can clog and springs can fail over time &lt;br&gt; • Control valves near pumps can be over-pressurized, reducing controllability</td>
</tr>
<tr>
<td>4 Reverse-return piping</td>
<td>• No balancing labor &lt;br&gt; • Coils may be added/subtracted without rebalance &lt;br&gt; • No significant over-pressurization of control valves close to pumps &lt;br&gt; • Usually lower pump head due to reverse-return piping having lower pressure drop than mains (due to larger pipe)</td>
<td>• Added cost of reverse-return piping &lt;br&gt; • Not always practical, depending on physical layout of system</td>
</tr>
<tr>
<td>5 Oversized main piping</td>
<td>• No balance labor &lt;br&gt; • Coils may be added/subtracted without rebalance &lt;br&gt; • Reduced over-pressurization of control valves close to pumps &lt;br&gt; • Lowest pump head/energy due to oversized piping &lt;br&gt; • Increased flexibility to add loads due to oversized piping</td>
<td>• Added cost of oversized piping</td>
</tr>
<tr>
<td>6 Undersized branch piping</td>
<td>• No balancing labor &lt;br&gt; • Coils may be added/subtracted without rebalance &lt;br&gt; • Reduced cost of smaller piping &lt;br&gt; • Reduced over-pressurization of control valves close to pumps where piping has been undersized</td>
<td>• Limited effectiveness and applicability due to limited available pipe sizes &lt;br&gt; • High design and analysis cost to determine correct pipe sizing &lt;br&gt; • Reduced flexibility to add coils where piping has been undersized &lt;br&gt; • Coils may be starved if variable-speed drives are used without ΔP reset &lt;br&gt; • Slightly higher pump energy depending on flow variations and pump controls</td>
</tr>
</tbody>
</table>

19 “Over-pressurized” means that the pressure differential across the control valve is greater than its rated differential pressure due to its proximity to the pump(s) (see Figure 1). Control valves that are over-pressurized will require a greater percentage of the valve stroke for balancing the circuit, thereby reducing the valve stroke available for control.

20 Taylor (2002) states this as a disadvantage in his ASHRAE Journal article. If the control valve is properly sized with or without the balancing valve the branch pressure drop will be the same at design flow.

21 Ibid.

22 Ibid.
Designing Self-Balancing Hydronic Systems

A self-balancing hydronic system is one that does not need to be mechanically balanced to maintain adequate flow through the coil for all load conditions. Bell & Gossett\(^{23}\) states that the great majority of hydronic systems are free from balance problems. Why is this so? Most hydronic systems operate without the need for mechanical balance because adequate waterflow is maintained through the coil. Inadequate flow through the coil results in the common balancing problem—the inability to meet the heating or cooling demand at high loads.

Coil Heat Transfer Characteristic Curves

Why is adequate flow through the coil maintained in most circumstances? Figure 2 shows a typical characteristic curve of the heat transfer versus flow for a hot water coil. Note that a decrease in water flow rate to 50% of design decreases the heat transfer ability of the coil only about 10%, and that flow must be reduced all the way down to 10% before the coil will exhibit a 50% drop in heat transfer ability. For a cooling coil, the characteristic cooling transfer curve as shown in Figure 3 is not quite as steeply sloped and there are sensible and latent heat components to the characteristic curve. The more gradual curvature is due to the lower design water ΔT’s for cooling coils. Nevertheless, for a cooling coil, 50% design flow will result in a sensible cooling transfer of about 85%, and a drop in flow to 20% of design will still result in about a 50% sensible cooling transfer at the coil. These curves represent the fundamental reason why most systems do not need to be mechanically balanced—substantial flow reduction must take place before heat transfer ability is affected to any extent. The balance problem occurs when flow rates through the coils have been reduced to the point where minor flow variations result in significant heat transfer changes, as shown by the “steep” part of the curve in Figure 2.

If it can be agreed that a coil flow volume that will result in at least 90% of the design heat transfer of the coil will meet the heating or cooling load, Bell & Gossett\(^{24}\) has developed design procedures that will eliminate all waterflow balance problems. We will call the coil flow rate that results in at least 90% of the design heat transfer of the coil the critical flow rate. HVAC distribution system designers should employ the following procedures to ensure that coil flow rates remain above the critical flow rate for all load variations.

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\(^{24}\) Ibid.
Transfer of Heat as a Function of Supply Temperature and Temperature Differential

Table 2 and Table 3 define the percentage of design coil flow rate that allows 90% heat transfer for various hot water system and chilled water system designs, respectively. Note that for either heating or cooling, the higher the design ΔT, the more the control valve must use more of its stroke for mechanically balancing the system. For heating systems, the higher the supply water temperature, the less stroke the control valve must use in mechanically balancing the system, whereas for cooling systems, the higher the supply water temperature, the more stroke the control valve has use to mechanically balance the system.

Table 2. Percent of Design Flow to Provide 90% of Design Heating Load.

<table>
<thead>
<tr>
<th>Design ΔT, °F</th>
<th>140</th>
<th>180</th>
<th>220</th>
<th>260</th>
<th>300</th>
<th>340</th>
</tr>
</thead>
<tbody>
<tr>
<td>Percent of Design Flow (%)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>61.5</td>
<td>52.3</td>
<td>46</td>
<td>41.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>40</td>
<td>76</td>
<td>68.9</td>
<td>63.7</td>
<td>58</td>
<td>52.5</td>
<td>47.6</td>
</tr>
<tr>
<td>60</td>
<td></td>
<td>74.7</td>
<td>70.5</td>
<td>66.5</td>
<td>62.3</td>
<td>58.4</td>
</tr>
<tr>
<td>80</td>
<td></td>
<td></td>
<td>76.5</td>
<td>71.5</td>
<td>67.6</td>
<td>63.5</td>
</tr>
<tr>
<td>100</td>
<td></td>
<td></td>
<td></td>
<td>78</td>
<td>73.4</td>
<td>70.7</td>
</tr>
</tbody>
</table>

Courtesy: Bell & Gossett.

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Designing Coils to be Self-Balancing

Since all coils in a distribution system are piped in parallel, the general circuit balancing problem can be described by considering the arrangement shown in Figure 4, where one of the two parallel circuits has been designed for a higher pressure drop than the other. Since the pressure drop across the headers joining the two parallel circuits must be the same, the problem is that the flow in the lower pressure drop circuit will be greater than design flow, and the flow in the higher pressure drop circuit will be less than design flow. We will show that deviations in the flow rates of each circuit are dependent on the ratio of design pressure drops of one circuit to the other and the ratio of design flow rates.

Figure 5 plots the ratio of the design flow rates against the ratio of the pressure drops for the two circuits shown in Figure 4. Note in Figure 5 that the 90% curve levels off at about 0.75 on the vertical axis. This shows that at least 90% of design flow will occur in the high pressure-drop circuit, as long as the design pressure drop of the low pressure-drop circuit is at least 75% of that for the high pressure-drop design. Since from Figure 2 or Figure 3, for a flow of 90% or more of design, one can safely assume that the coil heat transfer rate will be adequate to satisfy the heating or cooling load, one can conclude that if the design pressure-drops for the various parallel circuits can be kept within 25% of one another, flow distribution problems will be kept to a minimum. Since design pressure drops are determined by the coils, piping, and valves the distribution system design engineer selects for the sub-circuits in the system, the design engineer can essentially make the distribution system self-balancing. In fact, Avery (1990) states, "if the control valves are sized properly and the actuators are large enough to position the valve plugs properly, the system is self-balancing."27

Table 3. Percent of Design Flow to Provide 90% of Design Cooling Load.

<table>
<thead>
<tr>
<th>Chilled Water Supply Temperature, °F</th>
<th>Design ΔT, °F</th>
<th>40</th>
<th>45</th>
<th>50</th>
</tr>
</thead>
<tbody>
<tr>
<td>Percent of Design Flow</td>
<td>6</td>
<td>56.7</td>
<td>63.5</td>
<td>68.6</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>64</td>
<td>68.4</td>
<td>74/3</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>69</td>
<td>73</td>
<td>78.5</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>73</td>
<td>77</td>
<td>82.1</td>
</tr>
<tr>
<td></td>
<td>14</td>
<td>76</td>
<td>79.8</td>
<td>84.8</td>
</tr>
<tr>
<td></td>
<td>16</td>
<td>78.7</td>
<td>82.3</td>
<td>87.2</td>
</tr>
</tbody>
</table>

Figure 4. Two Water Circuits Piped in Parallel.

Selecting Control Valves and Coils to Assure a Self-balancing System

This section will define various valve performance parameters and state how each is important in control valve sizing.

Combined Valve and Coil Characteristic Curves

Figure 2 shows a typical characteristic curve for heating transfer for a 20°F temperature-drop heating coil. Figure 3 shows a typical characteristic curve for cooling transfer.

For good heat transfer control throughout the control range, it is desirable to have a linear heat transfer output with valve stem travel. This is due to the fact that the PID (proportional, integral, derivative) algorithm is a linear control algorithm and performs best on linear systems. This is accomplished by choosing a control valve with an equal percentage (plug) characteristic (Figure 6). When the heat transfer characteristic curve is combined with the equal percentage control valve characteristic curve, the heat output of the coil shows a theoretically linear relationship with the stem travel, as shown in Figure 7.28

Avery (1993)29 states that a direct-return variable-flow hydronic system must be designed to be self-balancing, and that for this system to be self-balancing, the sensor controlling the coil valve must always be in control of the waterflow through the coil.30 Selecting the control valve characteristic to produce as linear a curve of the coil heat output versus stem travel as possible will advance this goal.

Figure 8 shows the distortion of an equal percentage control valve characteristic versus control valve authority. A control valve authority of 1.0, as shown in Figure 8, most closely matches the ideal equal percentage characteristic curve shown in Figure 6.

Control valves for Variable Flow Hydronic Systems must have equal percentage valve flow characteristics to compensate for the non-linear relationship between heat output and flow of the coil. The goal is to achieve a linear process relationship

References:
30 If the valve controller is a room thermostat, then it should be locked at design temperature. If the public can adjust the setpoint so that the coil valve always stays open, then the system is no longer self-balancing.
between the lift of the valve defined by the controller output, \( Y \), and the heat output of the coil. This is because the Proportional plus Integral plus derivative (PID) control algorithm is linear and designed to control linear processes. The combined coil and valve characteristic exhibits a relationship that is approximately linear between stem travel and the related heat output.

**Control Valve Authority**

The designer should select a high control valve authority to ensure the least distorted flow characteristic.

The control valve authority is defined differently by different sources. Hegberg (1998)\(^{32}\) defines authority as:

\[
P_{v} = \text{Valve Authority} = \frac{\text{Pressure drop across control valve}}{\text{Pressure drop across system}} \tag{2}
\]

Where Pressure drop across control valve is determined from the equation:

\[
\frac{GPM}{C_v} = \sqrt{\Delta p} \tag{3}
\]

Where:

- \( GPM \) = coil design flow in gallons per minute.
- \( \Delta p \) = pressure drop across the valve at design flow in psi.
- \( C_v \) = control valve flow coefficient. The valve \( C_v \) is defined as the flow in gallons per minute through the wide open valve with a one psi pressure drop across it.\(^{33}\)

The Pressure drop across the system is determined by totaling the pressure drop path from the pump discharge through the supply main, branch supply, coil, branch return, and return main to the pump suction.

ASHRAE\(^{34}\) defines valve authority as the ratio of the control valve pressure drop when it is controlling to its pressure drop when it is wide open.

Siemens Building Technologies, Inc. defines control valve authority as the ratio of the pressure drop across the wide open valve to the branch circuit pressure drop when the valve is fully closed.

The distortion of an equal percentage control valve characteristic versus control valve authority is shown in Figure 8. A control valve authority of 1.0, as shown in Figure 8, most closely matches the ideal equal percentage characteristic curve shown in Figure 6.

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Relationship Between Valve Authority, Valve Stroke Needed for Control, Pressure Rise, and Pressure Drop Ratio

The valve authority simply relates the valve pressure drop at nominal design flow, \( V_{100} \), defined as \( \Delta p_{V100} \), divided by the branch circuit pressure drop \( \Delta p_{VR} \). Figure 9 shows the relationship between the coil pressure drop, \( \Delta p_{MV} \), the valve pressure drop at nominal design flow, \( \Delta p_{V100} \), and the circuit pressure drop, \( \Delta p_{VR} \). If the pressure drop across the valve and coil are the same, the valve authority of 0.5 results and the valve has its full stroke available for control. For example, in Figure 10, if the branch circuit pressure drop \( \Delta p_{VR} = 200 \text{ kPa (29.1 psi)} \) and the valve pressure drop at design flow \( \Delta p_{V100} = 50 \text{ kPa (7.3 psi)} \), the valve authority, \( P_v = 0.25 \), and 82% of the valve stroke is available for control. The valve authority \( P_v \) could also be called Pressure Drop Ratio (PDR). In the literature, Avery (1993)\(^{36}\) refers to the Pressure Rise Ratio (PRR), which is identical to the inverse of the Pressure Drop Ratio or valve authority \( P_v \).

\(^{35}\) Ibid.

Figure 9. Pressure Loss in a Typical Branch of a Variable Flow System.

The valve constantly changes to adjust flow to the branch depending on the actual load in the building.

\[ \Delta p_{V100} = 50 \text{ kPa} \]
\[ \Delta p_{PR} = 200 \text{ kPa} \]

Example:

\[ P_v = \frac{\Delta p_{\text{valve fully open at } V_{100}}}{\Delta p_{\text{valve fully closed}}} = \frac{\Delta p_{V100}}{\Delta p_{PR}} \]

\[ \Rightarrow P_v = \frac{\Delta p_{V100}}{\Delta p_{PR}} = 0.25 \]

There is \( Y = 82\% \) available for control.

Note: Siemens equal percentage control valves with slope factor \( n_{gl} = 3.0 \).

Figure 10. Relationship of the Valve Authority and Available Stroke.

The relationship of the valve authority \( P_v \) and the available stroke \( Y \) for control for an equal percentage valve. To achieve more than \( Y = 82\% \) the valve authority \( P_v \) should be larger than 0.25.
Here the valve authority at design conditions is simply defined as the valve pressure drop at nominal flow divided by the branch pressure drop at the location of the coil. The further away the coil and valve are from the pump(s), the lower the available branch pressure drop. The critical coil should be sized for a valve authority of 0.5. Therefore, the branch pressure drop at this location should be twice the pressure drop across the coil.

Figure 11 shows the relationship of the valve flow $C_v$ as a function of the stroke of the control valve. This is the typical control valve characteristics for an equal percentage control valve of Siemens. If the stroke $Y$ is plotted against the valve authority $P_v$ it can be shown how much of the stroke $Y$ is required to achieve nominal flow and how much stroke is available for control of the load of the coil.

### Pressure Rise Ratio

The pressure rise ratio of a hydronic system is the ratio of the design pump head to the valve pressure drop at design flow, with the valve wide open (Avery 1993). In a properly controlled variable-flow hydronic system, there can only be one valve fully open for any given system load.

When the two-way modulating control valve is seating and unseating, the entire branch pressure drop is across this valve. Therefore, in variable-flow hydronic systems the balancing is done by the last segment of the plug lift (Avery 1993). This segment is not usable to modulate the flow of water to the coil under design operating conditions. It is desirable to minimize this loss in lift so that more of the valve stroke can be used for flow modulation. This can be accomplished by minimizing the pressure drops in the mains and all components in the branch circuits, except the control valve. As a rough rule-of-thumb that appears to be derived from Avery’s experience, Avery (1993) recommends that the circuit be designed so that only the last 20% of the plug lift or less is used to balance the system.

### Valve Rangeability

The designer should also select a control valve with as high a valve rangeability as feasible to ensure a predictable minimum flow through the control valve at low loads. Valve rangeability is defined as:

$$Valve\ Rangeability = \frac{Maximum\ controllable\ flow}{Minimum\ controllable\ flow}$$

Typically, since a great number of operating hours are spent at low-load conditions, it is important that the valve has the rangeability as possible.

The two-way coil modulating valve will always control better if the balancing valve is not used and the pressure drop across the control valve is increased by the drop that would be allocated to the balancing valve. The control valve will then be smaller and the rangeability and close-off will be greater (Avery 1993).

The minimum controllable load is defined by the rangeability and the valve authority (Petitjean 1994). The minimum controllable load can be estimated by the following relationship.

$$Q_{min} \approx \frac{100}{Rangeability \times \sqrt{P_v}}$$

Where:

- $Q_{min}$ is the minimum percentage controllable load.
- $P_v$ is the valve authority.

A high rangeability results in a good control performance at low load conditions. As shown in Equation (5), the influence of the valve authority on the minimum controllable load is minor, since $Q_{min}$ varies as the reciprocal of the square root of $P_v$.

Figure 11 shows the relationship between valve rangeability, installed valve rangeability, and minimum controllable flow. The valve rangeability is based on the actual maximum flow, which is derived from the calculated $C_v$ of the valve, whereas the

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37 The “critical coil” is the coil that will typically drive the flow requirements of the distribution pumps. The critical coil is usually the coil farthest from the distribution pumps in a direct-return distribution piping system.

38 Ibid.

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installed valve rangeability is based on the installed maximum flow, which is based on the $C_v$ of the valve actually installed in the system. It is important to distinguish between these two definitions of rangeability because the $C_v$ of the valve actually installed is often rounded down from the calculated (ideal) valve $C_v$ because valves are manufactured only in discrete sizes.

The recommended basic sizing rule is to take half of the design pressure drop across the branch at design flow and use this value to size the valve. For example, if the branch pressure drop is 30 psi, the valve is sized with 15 psi. The sizing takes this pressure loss and calculates the required $C_v$ value with the required nominal GPM flow $V_{100}$ as shown in Figure 12.

The ASHRAE 2004 Systems and Equipment Handbook recommends that the control valve pressure drop should be at least 25% to 50% of the system loop pressure drop, which is equal to the pressure drop across the pump discharge flange, supply main, supply riser (if applicable), coil, control valve, balancing valve (if used), and return main to the pump suction flange. Valve pressure drops more than this enhance controllability, but at the expense of pump energy consumption.

Design specifications for variable-flow hydronic systems must stress the importance of the valve actuator and the need for high quality valve bodies to withstand the additional dynamic forces and static pressures that are prevalent in these systems.

All valves, in particular the globe-type control valve commonly used in the hydronic circuit for heating and cooling coils, has four pressure ratings: (1) the valve body static pressure rating, generally 250 psig (1724 kPa); (2) a close-off rating based on the closing force of the actuator; (3) valve rangeability (pressure-rise ratio); and (4) a valve dynamic pressure rating, which is defined as the maximum differential pressure that the wetted parts will withstand. This last rating is often inappropriately ignored when selecting valves for variable-flow systems.

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44 See pages 42.8 – 42.9
The basic rule for sizing the valve is to

... assume the flow \( V_{100} \)

and size the valve for

\[
\Delta p_{V_{100}} \geq \frac{\Delta p_{VR}}{2}
\]

Figure 12. Basic Valve Sizing Rule.

In Figure 12 \( V_{100} \) is the full-load flow. \( \Delta p_{V_{100}} \) is the pressure drop across the control valve at full-load flow, and \( \Delta p_{VR} \) is the pressure drop across the branch circuit.

**Required Parameters for Good Valve Sizing**
- Design pump flow
- Total system pressure drop at design flow
- If the total system pressure drop at design flow is unknown, use the design pump head

This information can easily be obtained from the pump manufacturer. *This information will provide the least accurate valve sizing, but it can be shown that the control valve will still have sufficient stroke (nearly 80%) available for control.*

**Required Parameters for Better Valve Sizing**
- Coil design pressure loss
- Total system pressure drop at design flow
- If the total system pressure drop at design flow is unknown, use the design pump head

If the valve authority is less than 0.25, then the \( C_v \) of the valve should be reduced by one (valve) size. This information can easily be obtained from the coil and pump schedule blueprints for the job, or from the pump manufacturer.

**Required Parameters for Best Valve Sizing**
- Branch circuit design flow.
- Branch circuit pressure loss for each branch circuit.

This information can be obtained from the coil and pump schedule blueprints for the job, plus information on pipe size and length of pipe to each branch circuit. The branch circuit pressure loss for each branch circuit that is required for best valve sizing can also be estimated using a model of branch pressure loss similar to the pressure gradient diagram illustrated in Figure 13.
Figure 13 shows the pressure gradient model used in Siemens valve sizing tool to estimate branch circuit pressure loss for a typical variable flow hydronic system. The table in Figure 13 shows the calculated $C_v_{\text{calc}}$ and the selected $C_v$ of the control valve. The table beneath the layouts also shows the effective pressure loss $\Delta p_{V100}$ and the valve authority $P_v$.

The required branch pressure loss information is taken from the pipe sizing calculations. The branch pressure loss can be estimated when the critical coil
pressure loss and the system pressure loss, which the system is designed for, are known. A linear estimation for each branch can be made.

For example, if the critical coil has a pressure drop of 45 kPa (6.55 psi) and the system pressure loss is 225 kPa (32.6 psi), the following assumptions allow simple sizing.

The table beneath the layouts in Figure 13 shows the sizing assumptions. The sizing rule is to take half of the pressure drop across the branches to size the valve. The result is to always obtain a valve authority $P_v > 0.25$ and with an equal percentage valve characteristics it is assured to attain more than 82% of the stroke for control.

The best valve size is achieved with this sizing approach. However, if less information is available, a valve can be sized with less information. However, it must have a valve authority $P_v > 0.25$ to assure good control performance with a high performance control valve.

Other Valve Parameters Needed for Proper Control Valve Selection

Valve Body Static Rating
Commercial HVAC valve bodies are generally rated at 250 psig, which is adequate for most installations (Avery 1993). On variable-flow hydronic systems, however, the valve bodies must withstand not only the static hydronic head, plus the imposed expansion tank pressure, but also the full-speed pump head when the valve is closed. This also applies to all pipes and fittings in each variable-flow hydronic system circuit.

Valve Close-off Rating
The valve close-off rating is the highest pressure differential the control valve will close off against to a high performance leakage. For example, if the sub-circuit closest to the pump experiences a no load condition, while the other sub-circuits in the system are at design load, this valve close-off rating must equal the design head of the pump.

Valve Dynamic Pressure Rating
The valve dynamic pressure rating is defined as “the maximum flow differential pressure (psid) that the wetted parts are designed for” (Avery 1993). This last rating is often inappropriately ignored when selecting valves for variable-flow hydronic systems. In certain valve manufacturer’s literature, this may be referred to as the water modulating differential pressure, the maximum allowable throttling differential pressure, or the maximum recommended differential pressure for modulating service. Avery (1993) recommends that the valve dynamic pressure rating for any valve on a variable-flow hydronic system should be at least 1.5 times the design pump head. It is easy to understand that the valve closest to the pump must withstand a differential pressure across it equal to the design pump head when this valve closes when system flow demand is at design. The 50% safety factor should provide enough reserve rating to handle the higher pressure differentials when the pump is operated at full speed or is accidentally deadheaded, though this factor is redundant as the valve manufacturer has chosen the valve wetted surface materials, including the manufacturer’s own safety factor.

Pump cut-off head, shown in Figure 14, can occur across any valve during emergency conditions; therefore, all coil valves in the system must be selected for this differential. On large installations, it may be necessary to use industrial valves to meet this high performance because of the higher pump heads required for the longer pipe runs on these systems.

![Figure 14. Pump Design Head and Cut-Off Head.](image)

Because the maximum throttling differential pressure occurs when the valve is near close-off, this

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parameter is based on the valve seat “trim”. The harder the trim material, the higher the differential pressure it can withstand. Therefore, the valve dynamic pressure rating is effectively limited by the trim material of the valve plug and seat. Operational problems will occur when valves with dynamic pressure ratings, too low for the application, are selected because the valve plugs and seats deteriorate.

**CAUTION:**

The valve body dynamic pressure rating is not necessarily identical to the valve’s close-off rating.

The present method in the industry of rating two-way valves may be confusing. Many times, the close-off rating is the dynamic pressure rating. For example, a manufacturer’s valve selection chart may list a valve with a large actuator having a close-off rating of 120 psi (827 kPa), but a footnote may specify a “maximum flow differential of 25 psi (172 kPa) for normal seat and disc life”. The 25 psi rating is the dynamic pressure rating of the valve. If this valve were installed in a real system, where the dynamic pressure requirements typically approach the close-off rating of the valve, this valve would have a short service life.

An alternative criterion for selecting valve dynamic pressure rating is to select the valve for the expected worst case dynamic pressure at the valve location. This allows the pipe friction losses between the pump and the valve to be taken into account when determining the differential pressure the valve will experience when modulating.

Figure 15, adapted from Figure 2 of Avery et al. (1990), shows a typical example of a primary-secondary chilled water system where the speed of the secondary chilled water pump is controlled from an end-of-line differential pressure sensor. In this example, the following conditions were assumed:

- All six coils are of the same size.
- The coil pressure drop is 10 feet of head.
- The differential pressure sensor (DP) will maintain 20 feet of head across the ends of the supply and return mains by operating the pump variable-speed drive.
- The mains are sized for a 4-foot head pressure drop between adjacent sub-circuits.
- Pump P-5 is large enough to handle the design flow when all control valves are wide open.

The differential pressures that are shown must exist across the control valves and coils at design flow conditions. In most cases, these are the normal worst case dynamic pressures expected at each of the valve locations.

Of course, sizing the control valves according to these conditions does not take into account emergency conditions, such as a partial plant power failure that might cause all control valves to suddenly shut. However, if this situation is expected to occur periodically, one can select normally open or fail-in-place valves instead of normally closed valves for the application. Also, the time that this emergency condition might occur must be viewed within the context of how likely is this to occur and when this condition does occur, as well as, how long will it endure. In most cases, emergency conditions of this nature seldom occur and when they do occur, they are likely to be a tiny fraction of the total operating hours of the system.

*All else being equal, the control valves closest to the pump(s) are most likely to experience the highest dynamic pressures.* This is due to the fact that these zones experience the least head loss between the pump and the control valve, and therefore dynamic pressure fluctuations due to control valves’ modulation are superimposed on the higher operating pressures across the valves (see Figure 1). However, since modulating control valves have a slow-acting response on the order of 30 seconds to two minutes, both the frequency and the magnitude of dynamic pressure fluctuations are relatively small compared to the differential pressures across the control valves.

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49 Ibid.

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Valve and Actuator Capability

When the distribution system ΔT is low, the differential pressures across control valves close to the pump(s) are higher than what has been designed for. This is due to a higher chilled water supply temperature; therefore, more flow is needed to satisfy a given load, creating the higher differential pressures across terminals close to the pump(s). This is illustrated in the pressure gradient diagram of Figure 16. Improperly-sized control valve actuators may lack the strength to close off the valve completely at the higher differential pressures experienced for circuits close to the pump when the load on that circuit drops to zero, or even to sufficiently close for low loads. In these cases, the control valves may begin to operate like two-position (on/off) valves.

Valve Actuator Sizing

Due to the widely variable pressure differentials across hydronic branches, it is recommended (Avery 1993) that large valve actuators be installed that can close tightly and can precisely position the valve plugs. Actuators should be sized to close against at least 1.5 times the design pump head. This will ensure good valve plug positioning, and for pneumatic valves, minimize any spring range shift caused by high differential pressure across the valve.51

50 Ibid.
51 The spring range shift on pneumatic valves with small actuators can be so severe that both the heating and cooling valves can be forced wide open. For example, many four-pipe chilled-hot water systems have heating valve operators with a 3 to 8 lb spring range and cooling valve operators with an 8 to 13 lb spring range. If undersized actuators are installed and high valve pressure differentials occur such that the spring range is shifted 5 to 10 lbs on the heating valve, simultaneous heating and cooling will occur.
discrete flows which the valve can produce at a given differential pressure.

- High valve authority. As we have seen, valve authority is defined differently by different sources. By the Siemens Building Technologies definition, the valve authority should be between 0.25 and 0.5 for typical heating and cooling coil heat transfer characteristics.

- Good repeatability (3% hysteresis or less). The Siemens 599 Series Flowrite™ and Powermite globe control valves are specially designed for campus-level applications, such as modulation of chilled water (down to 32°F), hot water (up to 250°F), or steam (up to 15 psig) coil modulation. Valves are available from ½ inch to 6 inch. The valve can be either pneumatically or electronically actuated, with either actuator having a 100:1 resolution and a less than 2% hysteresis. The valve body static rating is either 125 psig or 250 psig, and the pneumatic, electro-hydraulic, and electro-mechanical actuators are capable of producing thrusts of up to 1000 pounds. Thus, this valve conforms to the valve body static rating and valve actuator sizing (close-off) criteria recommended by Avery (1993).

If a chilled water control valve is selected with a rangeability of 20:1, the minimum flow predictable in the valve is \((100\%/20) = 5\%\) of maximum flow. Specifying a valve rangeability of 100:1 will bring the minimum predictable flow in the valve to \((100\%/100) = 1\%\) of maximum flow. Thus, the Siemens 599 Series Flowrite™ and Powermite globe control valve can be classified as a high-performance control valve.

Criteria for Superior Valve Performance

From the preceding discussion, there are a number of requirements that must be met to assure a superior valve performance, both from a control point of view and from an energy point of view:

1. The system must be designed to be self-balancing. For this to occur:
   - The design pressure drops from the various parallel sub-circuits must be kept within 25% of one another.

2. The control valve for each circuit must be sized based on the maximum differential pressure expected across that valve.

3. The pump control loop must be well-tuned. The water valve is controlled to maintain a coil discharge air setpoint, while the pump speed is controlled to maintain a setpoint differential pressure across the farthest control valve from the pump(s) (for a direct-return piping system) or at the middle terminal unit (for a reverse-return piping system). This requirement also arises from the pressure fluctuations that exist across the control valves for any real chilled or hot water plant and for the need to have the pumps respond immediately to any changes in load. Note that this requirement becomes relatively easy to satisfy with today's modern DDC controllers.

Pressure-Independent Control Valves

Pressure-independent control valves have been touted since the early 1990's (mainly by their manufacturers) as a means of assuring the proper balancing of hydronic (chilled or hot water) systems and eliminating the need for separate balancing valves, bypass valves, three-way valves, pressure-reducing valves, and reversed-return piping.

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52 See product brochure 153-277P10 Rev. 4 (05/02), or visit the Web site: www.sbt.siemens.com/hvp/components.

53 This is generally the case when the supply air temperature controls the coil valve on a VAV air handling unit.
designs. The main impediment to wider use of these valves has been their initial cost. However, since 2003, at least one pressure-independent valve manufacturer has been promoting the idea that these valves can also maximize the plant and distribution system supply/return water $\Delta T$, and that this will lead to substantial reduction in boiler or chiller, pump, and air handler fan power consumption over the course of a heating or cooling season. The valve manufacturer claims the plant and distribution equipment power savings from the use of pressure-independent control valves, will more than offset their higher initial cost. Does a pressure-independent control valve offer an inherent advantage in attaining the highest possible water $\Delta T$ in a hydronic circuit if all of the other factors that affect $\Delta T$ are eliminated?

From testing that has been conducted, one of the conclusions is that if a high-performance, pressure-dependent control valve is sized and installed properly, it can actually attain a higher $\Delta T$ performance compared to that for a pressure-independent control valve. Another conclusion is that for the vast majority of applications, control valves are not sized perfectly due to the finite number of valve selections available from the manufacturer, and less than optimal $\Delta T$ performance is attained. However, if the valve control logic of a high-performance control valve is enhanced to control for maximum coil water $\Delta T$ within a set tolerance of the discharge air temperature setpoint, a high-performance control valve can at least attain the same $\Delta T$ as that for a pressure-independent control valve. The water $\Delta T$ performance between these two types of valves will be examined in a separate Technology Report that will be published in the future.

Theory of Operation

Figure 17 shows a comparison between a high-performance control valve and a pressure-independent control valve. Note that a pressure-independent control valve is really two valves in series—the upstream valve is the control valve, while the downstream valve functions as a pressure-regulating valve. The first valve is the $C_v$ section. This section changes the flow rate through the valve as the $C_v$ is changed by rotating the valve stem manually or with electronic or pneumatic actuation. The second valve is the differential pressure section. The spring and piston in this section function to maintain a constant differential pressure ($P_1$ through $P_2$) across the control surface using only the internal fluid pressure in the valve. Therefore, flow is unaffected by pressure changes ($P_1$ through $P_3$) across the valve. When a pressure-independent valve is installed in a circuit that experiences a high differential pressure across the valve, only a small part of the differential pressure exists across the control surface; the remaining pressure is absorbed by pressure regulating part of the valve.

As load varies on a chilled water system, the differential pressure across any given coil control valve changes continuously. Pressure-independent control valves maintain relatively constant differential pressure across the control surface within the valve so that the flow rate remains steady until a change in load for that zone (coil) signals the valve actuator to move. Dynamic pressure fluctuations are absorbed by the diaphragm and spring arrangement as shown in Figure 17. Pressure-independent control valves automatically balance the system regardless of the amount or location of the system load because the pressure regulating part of the valve absorbs any excess pressure across the valve. As long as the differential pressures across the pressure-independent valve are within their specified range, high-load zones being near or far from the pump(s) does not affect the flow rate through the valve.

Figure 18 shows the pressure-independent valve pressure-flow characteristics. The typical operating range of the valve is in the flat (vertical) part of the characteristic curves. Note that because the pressure-independent valve has different pressure-flow characteristics compared to a high-performance, pressure-dependent valve, in general a larger valve size must be selected for the pressure-independent valve compared to a pressure-dependent valve for the same design (operating) point.

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54 Currently, there are three major manufacturer’s of pressure-independent control valves: (1) Flow Control Industries, Inc. who introduced their “Delta P” valve in 1992, (2) Belimo, who introduced their “PICCV” valve in 2003, and (3) Griswold, who introduced their “PIC” valve in 2003.
**Figure 17. Globe Valve Compared to Pressure-independent Control Valve.**

Pressure independent control valve

\[ GPM = \text{Cv} \times \sqrt{\Delta p_W} = 40 \times 2.6 = 105 \text{ gpm} \]

High performance control valves

\[ \Delta p_{V100} = \left(\frac{GPM}{\text{Cv}}\right)^2 = \left(\frac{100}{25}\right)^2 = 16 \text{ psi} \]

*Figure 18. Pressure-Independent Valve vs. High Performance Control Valve Performance Characteristic Curves.*

In Figure 18, the smaller valve with a \( \text{Cv} = 25 \) is selected. The valve authority is defined as \( \Delta P_v = 16/30 = 0.53 \).

For example, if a pressure dependent control valve has a \( \text{Cv} = 40 \) and a pressure differential regulator with a \( \Delta p_W = 7 \) psi, the control valve pressure drop is...
controlled at this value. This results with a flow of 105 gpm, as the example shows in Figure 18. If the required flow is 100 gpm, there is an overflow of 5 gpm when the valve is fully open. The required minimum pressure differential to operate the pressure-independent control valve is about 10 psi. This pressure is required to operate the pressure regulator part.

The high performance control valves closer to the pump can be sized one size smaller. The smaller size allows better control. The valve should not be too small; otherwise, the coil may be starved. This is assured by applying the sizing rule that the valve pressure drop is half of the branch pressure drop at any location in the system.

Advantages

If pressure-independent control valves are installed throughout a chilled or hot water system, the system will definitely be balanced. Therefore, one of the chief advantages of installing these valves is that no balancing labor is ever required. This means that coils can be added or removed from the system without rebalancing the system. However, as we have seen, if pressure dependent valves are sized correctly, the water circuits are self-balancing, and no balancing labor is required on these systems either.

Another advantage of pressure-independent control valves is that as long as the differential pressures across the valve are within their specified range (the valve dynamic pressure rating), no overpressurization of control valves close to the pumps will occur. As Table 1 shows, failure to accurately control the flow to zones close to the pumps on large systems is one of the disadvantages of the no balancing option, which Taylor (2002) recommends in most cases. For this reason, he recommends that these valves be considered on very large systems for coils near the pumps. In addition, a high control valve authority is maintained with a pressure-independent control valve. This makes valve selection easier. Ease of valve selection is a major advantage of pressure-independent valves, as most pressure-dependent valves are not sized properly for the application.

Note that if a coil near the end of the distribution system were to experience a lower than design $\Delta T$, as shown in Figure 16, the differential pressure across control valves closer to the pump would shift upwards and dynamic pressures would fluctuate about this higher differential pressure, increasing the chances of $\Delta T$ degradation for those circuits closer to the pump. This implies that pressure-independent valves must be installed on every circuit in order to gain the increased water $\Delta T$ performance for the entire water distribution system.

Disadvantages

A major disadvantage of the pressure-independent valve is its expense. It is estimated that the first cost of pressure-independent valves can be as much as four to five times greater than that of a comparably sized high-performance pressure-dependent valve. Also, Flow Control Industries states that pressure-independent valves have to be installed on every chilled or hot water circuit in order to gain the increased water $\Delta T$ performance for the entire water distribution system. However, the claim of higher water $\Delta T$ performance compared to an

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57 Ibid.
58 This cost difference is estimated based on purchasing a Flow Control Industries pressure-independent (DeltaP™) valve for a head-to-head valve performance test with a comparably sized Siemens 599 Series Flowrite valve.
59 Per e-mail correspondence with Flow Control Industries, Inc. on November 3, 2004.
“industrial-quality” control valve\textsuperscript{60} has not been proven. If a high-performance control valve is sized and piped properly, and enhanced $\Delta T$ valve controls have been installed, then the water $\Delta T$ attained equals and exceeds that of a pressure-independent control valve.

There are several other disadvantages to installing pressure-independent control valves, which are listed in Taylor (2003).\textsuperscript{61}

- Valves have custom flow rates and must be installed in the correct location.
- Valves can clog or springs can fail over time.

Taylor (2003) also says that strainers are needed for each individual terminal unit, upstream of the valve. Flow Control Industries disputes this, saying that strainers are only needed on the “system level” (headers) to minimize fouling, pump impeller, or chiller tube erosion.\textsuperscript{62}

Taylor (2003) also says that higher pump head and energy due will result when these valves are installed due to the strainer and the valve itself. However, this last point can be logically disputed since the pressure drop across a high-performance, two-way control valve (assuming no balancing valves are installed) would be identical to that of a pressure independent control valve if it were installed in the same circuit and controlled to identical water $\Delta T$s across the coil.

\textsuperscript{60} An “industrial-quality” control valve is comparable in performance to a high-performance control valve.


\textsuperscript{62} Per correspondence with Flow Control Industries, Inc., March 2004.

Conclusions

Maximizing control valve performance for variable-flow hydronic systems means maximizing the water distribution system supply and return $\Delta Ts$, as well as responding quickly to changes in load and dynamic pressure across the valve. Unfortunately, this is a complicated subject, involving issues of hydronic system design, hydronic system balancing, valve and actuator sizing, and the inherent characteristics of pressure-dependent valves versus pressure-independent valves. Citing credible sources throughout this report and starting from a careful review of basic control valve and coil theory, this report leads the reader through procedures to assure a self-balancing hydronic system, proper valve and actuator selection, how pressure-independent valves work, and how high-performance, pressure-dependent control valves can be optimally sized. Assimilating this material leads one to the following conclusions:

1. For a system designed to be self-balancing (see sections \textit{Balance Valves in Hydronic Systems} and \textit{Selecting Control Valves and Coils to Assure a Self-balancing System}) balance valves are unnecessary. This is the most cost-effective approach (taking into account first costs and energy costs) for balancing a variable-flow hydronic system.

2. Pressure-independent valves are not necessary for optimal control valve sizing. Optimal valve sizing can be obtained with high-performance, two-way pressure-dependent control valves. Optimal valve sizing means that for all loads on a given hydronic branch circuit, at least 80\% of the valve stroke is available for control, while less than 20\% of the valve stroke is used for balancing.

3. The Siemens Valve Sizing Tool can be used to obtain good valve sizing such that nearly 80\% of the valve stroke is available for control, even if only the pump design flow and the design pump...
head pressure are known. The more information that is known about the branch coils, flows, and branch pressure drops, the more accurately the tool will calculate the valve size.

4. It can be shown that to achieve a valve stroke of more than 80% for controlling the valve, the valve authority (based on the Siemens Building Technologies definition) must be between 0.25 and 0.5.

5. Pressure-independent control valves are relatively easy to size properly, and if installed on every circuit of a water distribution system, will eliminate the need to know the distribution system branch pressure losses that are needed to ideally size a high performance, pressure-dependent control valve. However, the initial cost of these valves can four to five times higher than the cost of a comparably sized, high-performance pressure-dependent valve.

6. Pressure-independent control valves will definitely balance a variable-flow hydronic system, provided the valve is selected for the static and dynamic pressures expected in the circuit. Properly sized high-performance control valves will also balance the system.

7. Sizing all control valves the same results in a progressive oversizing of the control valve in the circuits as they get closer to the distribution pump(s).

8. Instead of sizing all control valves in a distribution system the same as is commonly done, the valve at each circuit should be sized individually according to the maximum pressure differential expected across that valve.