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Balancing Variable Flow Hydronic Systems

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How, and indeed whether, to balance variable flow (two-way valve) hydronic systems has been debated for many years.^{1,2,3} In most of the literature, either the analysis has been limited to only a few of the many balancing options, or analyses of flow control problems and their energy impacts were based on simplified or atypical hydronic systems. Few references compare in detail the first costs and energy costs of the various balancing options. This article attempts to fill these gaps by analyzing the balancing of variable flow systems in a detailed and comprehensive manner.

Features of the analysis include:

- Seven of the most commonly used methods for balancing hydronic systems are addressed.
- Two HVAC system applications are analyzed, one chilled water system and one hot water system, both of which are based on a real building.
- Flow through the system for each of the balancing options is analyzed using a commercially available pipe network analysis program that accurately models flow through all design elements as flow and pressure varies through the system.
- First costs and energy costs of each balancing option are estimated.

Example Systems

Two hydronic systems, a chilled water system and a hot water system, were analyzed. Both systems serve a 20-story 470,000 ft² (47 000 m²) office building.

The chilled water system is a primary-only variable flow system serving floor-by-floor air handler units (AHUs) with two-way modulating control valves (*Figure 1*). The design flow rate



High-rise building in California serves as a real-world base case for seven variable flow systems.

of the system is 1,200 gpm (75 L/s) (60 gpm [3.8 L/s] per floor). The chilled water pump has a variable speed drive (VSD) controlled by a differential pressure sensor located across the riser taps at the most remote AHU at the 20th floor. A bypass valve at the plant is provided to maintain the minimum flow rate through the chillers (only one of which is shown in the schematic).

The hot water system also is a primary-only variable flow system with a design flow rate of 540 gpm (34 L/s) (27 gpm [1.7 L/s] per floor) serving 400 VAV box reheat coils (20 per floor) (*Figure 2*). The design is similar to the chilled water system except there is no minimum flow bypass (the boilers in this case can operate at zero flow) and the hot water pumps do not have VSDs.

Figure 3 shows the layout of the hot water system on a typical floor. Because there is one main HW riser in this ex-

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ample, the piping on the floor can be routed using a reverse-return approach (first coil supplied is the last returned and vice versa) at little or no cost above direct-return. The data in *Table 1*, therefore, are all based on a reverse-return floor loop design.

Piping was sized using a pipe-sizing chart developed from life-cycle cost analyses.⁴ Chilled water control valves were sized to achieve approximately the same pressure drop as the branch circuit from the main risers, including the cooling coil, to achieve a “valve authority” of approximately 0.5.⁵ Hot water control valves were selected for approximately 2 psi [14 kPa] pressure drop. All control valves are globe-type with equal percentage plugs.

Balancing Options

Seven of the most common balancing options were analyzed:

1. No balancing,
2. Manual balance, using calibrated balancing valves (CBVs),
3. Automatic flow limiting valves (AFLVs),
4. Reverse-return,
5. Oversized main piping,
6. Undersized branch piping, and
7. Undersized control valves.

Figures 1 and 2 show schematics of balancing Options 1 through 5. Options

6 and 7 are schematically similar to Option 1 but pipe and control valve sizes are changed, as discussed in more detail later.

Figure 4 shows the coil piping details for each option. Table 1 lists performance data based on the pipe flow program and coil selection software along with first costs, energy demand, and annual energy costs of the two sample systems for each of the seven balancing options. Key assumptions are listed in notes at the bottom of the table. Table 2 summarizes the advantages and disadvantages of each balancing option. These and other issues are discussed in more detail in the following sections.

Option 1: No Balancing

The rationale behind this option is that if the coils are able to achieve their control setpoints at the coil design flow rate or less, then the control valves themselves will dynamically and automatically balance the system. Neither balancing devices nor balancing labor is required. The balancing contractor is asked only to ensure that coils are receiving flow (e.g., by measuring coil entering and leaving water and air tempera-

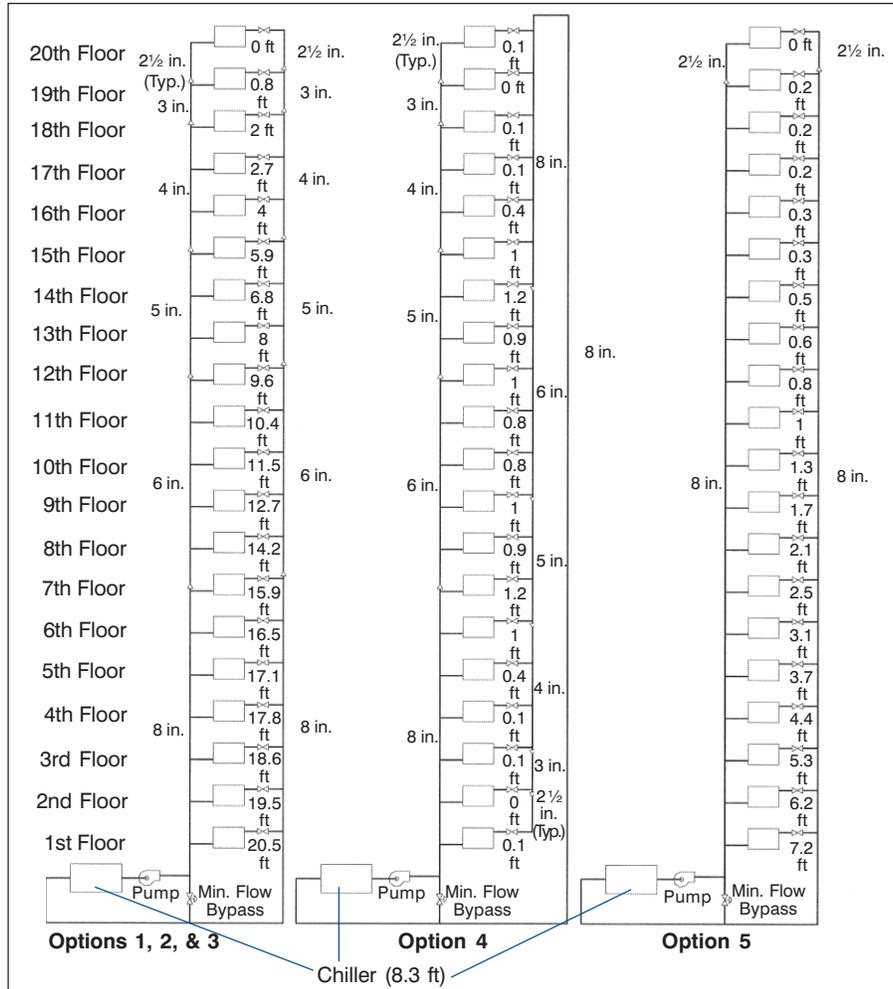


Figure 1: Chilled water riser diagram. Options 6 and 7 are not shown but are similar to Options 1 to 3. The pressure drops listed at each floor are the pressure drops across the balancing device at design flow rates. For all options except 2 and 3, the balancing device is the control valve.

tures), and to determine the design VSD control setpoint for systems with VSDs.

However, two questions arise with this design:

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

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

The computer simulation of this option, summarized in Table 1, shows that when all control valves are wide open, the coils that are hydraulically closest to the pump will have a flow rate above design (143% for chilled water and 212% for hot water). Those furthest from the pump will have flow rates below design (73% for chilled water and 75% for hot water). The flow variations in the hot water system are more extreme because the pipe size to all of the reheat coils was fixed at 3/4 in. [19 mm] regardless of flow rate (for construction simplicity). This causes heat-

ing coils with low flow rates to receive a larger percentage of their design flow during transients, and vice versa. The more customized the pipe sizes are to the design flow rate at each coil, the smaller these flow variations will be. Of course, piping is limited to certain standard sizes.

At first glance, one might expect that the low flow rates at the extremes of the system could cause problems. For instance, during cool-down or warm-up, the reduced flow might significantly increase the time it takes for the system to cool or heat the space to comfort levels. However, coil performance is inherently non-linear so a reduction in flow is not matched by an equivalent reduction in coil capacity. In this case, the reduced chilled water flow (73%) results in a coil sensible capacity that is 89% of the design capacity (based on the coil manufacturer's rating program). The reduced hot water flow (75%) results in a coil sensible capacity that is 96% of the design capacity (based on the VAV box manufacturer's reheat coil rating program), assuming supply water temperatures are at design conditions. These capacity shortages are most likely within the error (or safety factor) of the engineer's design load calculations and are unlikely to result in any significant increase in warm-up or cool-down times.

The controllability issue may be more problematic depending on control valve type and sizing. To reduce the excess flow at coils near the pump, the control valve must partially close. In the case of the chilled water system, an equal percentage valve selected for a valve authority of about 0.5 would need to close the valve to about 85% of full open to reduce the flow from the 143% excess flow to the design flow rate (see flow versus stroke curves in Hegberg 2000⁵). For the hot water system, the valve would have to close to about 75%. Will reducing the effective control range of the valve by 15% to 25% cause controllability problems? Not likely, particularly with modern controllers and actuators, but it does emphasize the need to correctly select valve size and to select an equal percentage flow characteristic for two-way valve applications.

The hot water system in this case uses direct-return on the risers but reverse-return on the floor piping, as previously noted. While not included in *Table 1*, a direct-return system on each floor also was simulated. With this design, the maximum differential pressure control valves had to create to deliver design flow increased to 66 ft (197 kPa) and the maximum and

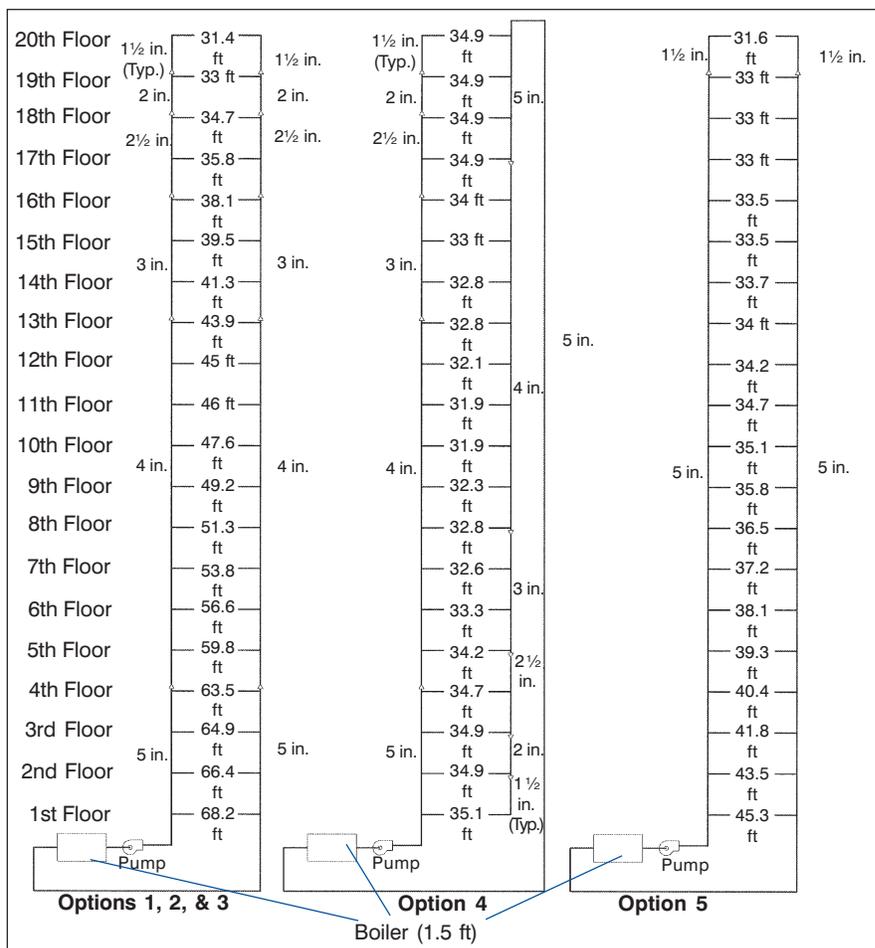


Figure 2: Hot water riser diagram. Options 6 and 7 are not shown but are similar to Options 1 to 3. The pressure differentials listed at each floor are the pressure differentials across the supply and return taps at each floor at design flow rates.

minimum flow with valves wide open broadened to 293% and 46%, respectively. To reduce the excess flow at the coil closest to the pump, its control valve would have to close to approximately 60% of full stroke. This may reduce the effective valve control range enough to reduce controllability. The more pressure the control valve has to absorb to achieve design flow rates (the larger the difference between the pressure drop through the hydraulically most remote circuit and that through the closest circuit), the more likely for control problems to result from direct-return systems using control valves for balancing. For these large, "hydraulically diverse" systems, other balancing options described later may provide better performance and should be considered.

Option 2: Manual Balance using Calibrated Balancing Valves

Manual balancing requires throttling valves and a means to measure flow at each coil. Flow is measured and valves are adjusted by the test and balance contractor to achieve design coil flow rates. Flow can be measured at coils indirectly by correlating flow to differential pressure measured across the coil using the coil manufacturer's pressure drop data. However,

the more common approach, which is the approach analyzed here, is to use valves with integral flow measurement capability, so-called calibrated balancing valves (CBVs). Modern CBVs have many forms, including globe valves with stroke and differential pressure correlated to flow, and standard ball or globe valves combined in the same body with a small venturi-type flow meter.

Balanced CBVs allow control valves to be wide open at design flow, increasing the effective valve control range, and they reduce the pressure that the control valves have to absorb at part load. Another advantage of CBVs is that they allow flow through the coil to be easily measured, not only for initial balance but also in the future to verify system operation and to diagnose flow problems that may arise.

On the other hand, the system has several disadvantages:

- Added cost of calibrated balancing valve compared to standard ball and butterfly valves. In this example, we have assumed that the CBV also can provide shut-off duty (*Figure 4*). If not, or if the designer does not want the CBV to double as the shut-off valve (a common practice), the first costs (and pressure drop) of this option will be even greater than indicated in *Table 1*.
- Higher balancing cost. The CBVs must be manually adjusted and, because throttling flow at one coil increases the flow through others, balancing typically requires more than one iteration to achieve the 5% to 10% accuracy generally required in specifications. The cost can be significant on a project with many coils, like the heating system in this example.
- Complete rebalance may be required if coils are added or subtracted. Again, changes in flow in one part of the system affect the balance in other areas.
- Slightly higher pump head due to balancing valve. The pressure drop characteristics of CBVs vary with the specific type, but they typically will have a somewhat larger pressure drop than standard ball valves and butterfly valves, as reflected in the design pump heads in *Table 1*.
- Coils may be starved if variable speed drives are used on pumps, as in our chilled water system example. The pump VSD control setpoint is determined by calculating or field measuring the differential pressure at the DP sensor location required to provide design flow through the most remote AHU coil. In this case, the setpoint was calculated to be 5.6 psi (39 kPa). If the system is manually balanced, the CBV at the first floor AHU would be adjusted to create a pressure drop of 8.8 psi (61 kPa). Now assume that only the first floor AHU is on, for instance, to serve a tenant during off-hours. The pump will slow

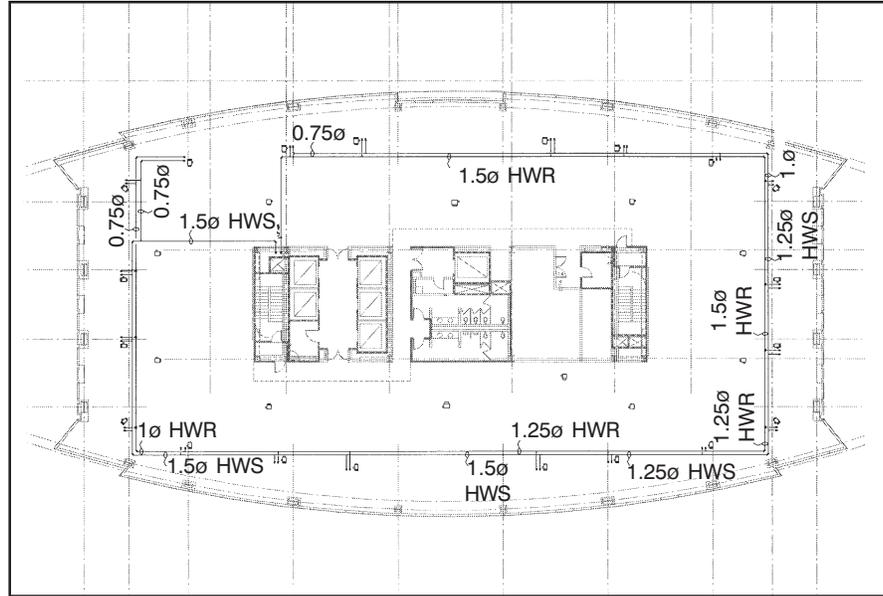


Figure 3: Floor plan showing HW piping. Piping is in a reverse-return arrangement on each floor for all balancing options since there is little or no cost premium compared to direct-return.

down and maintain 5.6 psi (39 kPa) at the 20th floor. Since there is no flow above the first floor, the differential pressure across the 20th floor and first floor is the same.

With the CBV balanced to 8.8 psi [61 kPa] pressure drop at design flow, the AHU will not be able to operate at full capacity with only 5.6 psi [39 kPa] available differential pressure. The maximum flow possible in this case would be about 37 gpm [2.3 L/s], 61% of design flow rate and sufficient to deliver about 81% of design sensible coil capacity. Will this cause comfort problems? Not likely, given that loads during off-hour operation tend to be well below design. If it were a problem and the control valve positions are known, as they generally would be with a direct digital control system, the DP setpoint can simply be adjusted upward until no valves and coils are starved.

- Slightly higher pump energy depending on flow variations and pump controls. The energy usage of this option may be higher at part load than Option 1 depending on how and where flow rates vary and if pump DP setpoint is reset by valve position. As noted previously, the partly closed CBVs at coils near the pump can result in higher DP setpoints if these coils have higher loads than more remote coils, and if DP setpoints are reset.

Option 3: Automatic Flow Limiting Valves

Automatic flow limiting valves (also called automatic flow control valves) are self-powered devices that limit flow to a preset value when the differential pressure across the valve is within a certain range (e.g., 2 psi to 32 psi [14 kPa to 220 kPa]). 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. AFLVs eliminate the need for balancing labor and allow coils to be added and subtracted from the system without rebalance. On the other hand, this option has several disadvantages:

- Strainers required at coils. AFLV manufacturers recommend installing strainers upstream of AFLVs so that construction debris in the piping system does not clog the small orifices in the valve. (Some designers install strainers at coils anyway to protect the control valve. It has been our experience that they are not required in closed systems except, perhaps, on valves with very small flow coefficients.)

- Cost of labor to clean strainers at start-up to remove construction debris. For coils mounted above ceilings and other locations with limited access, the cost to clean strainers can be very high.

- Added cost of strainers and AFLVs. Unlike CBVs, AFLVs cannot provide shut-off duty so service valves still must be provided. For smaller sizes, some manufacturers offer a combination AFLV, strainer, and service ball valve in a single assembly to reduce installation costs.

- Higher head and pump energy due to strainers and AFLVs. AFLVs have a significantly higher pressure drop than CBVs or standard ball and butterfly valves. The higher the AFLV control range (2 to 32 psi [14 kPa to 220 kPa] in this example), the higher the full-open pressure drop. This increase can be compounded if control valves are sized to retain the same valve authority as they were without the AFLVs. In this analysis, however, the control valve size was left the same, reducing valve authority to about 0.3 from 0.5.

- Valves have custom flow rates and must be installed in the correct location. Valves of the same physical size may be preset to one of perhaps 10 to 20 available flow rates, custom picked to match the coil. Typically, valves are clearly tagged at the factory, but it is not uncommon for them to be mixed up during installation. The resulting flow problems can be difficult to diagnose.

- Valves can clog or springs can fail over time. Any small orifice in a piping system can be subjected to fouling, particularly in systems that are frequently modified, as would be the case in our example hot water system where each tenant remodel could introduce construction debris.

- Control valves near pumps can be over-pressurized, reducing controllability. AFLVs are effectively only in the circuit when the control valve attempts to allow flow in excess of the design flow. When the flow drops below design (the usual condition when the valve is under control), the valve has essentially no impact on flow or differential pressure in the circuit. Hence, the control valve must operate against whatever differential pressure is available, just as it would with Option 1. The effective stroke range of the valve also is the same since as the valve begins to close, the flow through the coil will not change until the pressure drop across the AFLV is reduced below its control range (2 psi [14 kPa] in this example). Hence, in a large

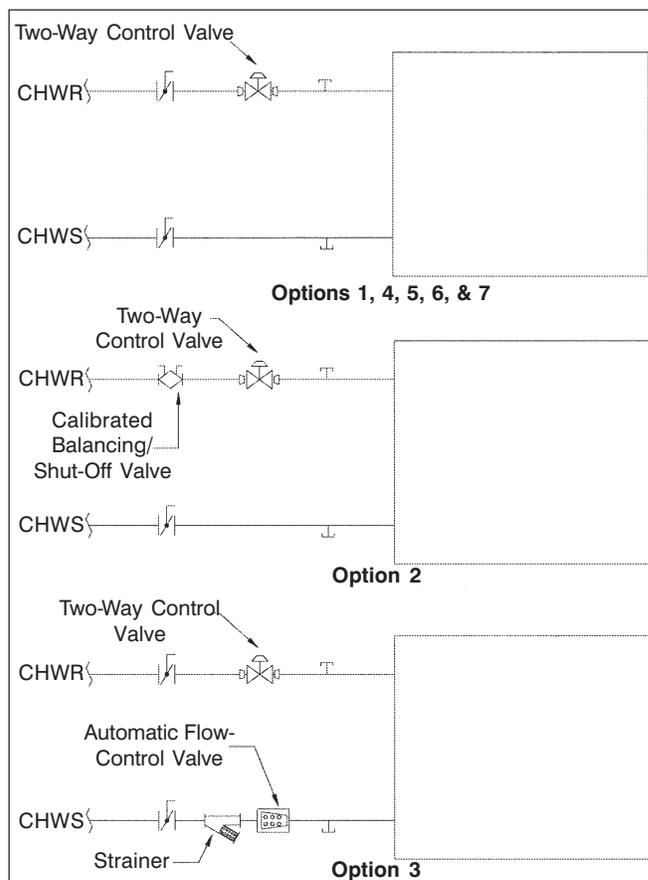


Figure 4: Coil piping diagrams.

system, the same controllability questions arise as with Option 1, and they may be worse since the overall system differential pressure will be higher due to the added pressure drop of the AFLV and strainer.

Option 4: Reverse-Return

The middle schematics in *Figures 1* and *2* show the reverse-return piping arrangement where the first floor supplied is the last floor returned. The effect is to cause differential pressure across each coil to remain fairly constant. This DP consistency will remain over all operating conditions provided coil flow rates change similarly.

Advantages of this option (*Table 2*) include:

- No balancing labor. The system is close to self-balancing even without the help of control valves. As shown in *Table 1*, flow variations for the chilled water system during transients are insignificant. On the example hot water system, flow variations were larger in percentage terms (from 150% to 85%) but very small in absolute terms (less than 1 gpm [0.06 L/s]). The impact on coil capacity in either case is negligible.

- No significant over-pressurization of control valves close to pumps. The data in *Table 1* indicate that differential pressure differences across valves to achieve design flow are almost zero for the chilled water system and reduced to 25% of the differential for Option 1 for the hot water system.

Balancing Option		Maximum Pressure Drop of Control Valve Required for Design Flow, ft [kPa]		Percent of Design Flow (% of Design Coil Sensible Capacity) with All Control Valves 100% Open				Pump Head, ft [kPa]		Annual Pump Energy, \$/yr		Incremental First Costs vs. Option 1			
				Maximum Flow Through Closest Coil		Minimum Flow Through Most Remote Coil						\$		\$ per Design gpm [\$ / L/s]	
		CHW	HW	CHW	HW	CHW	HW	CHW	HW	CHW	HW	CHW	HW	CHW	HW
1	No Balancing	20.5 [61.2]	44.4 [133]	143% (106%)	212% (119%)	73% (89%)	75% (96%)	58.5 [175]	82.7 [247]	\$1,910	\$3,930	—	—	—	—
2	Manual Balance Using Calibrated Balancing Valves	0	0	100% (100%)	100% (100%)	100% (100%)	100% (100%)	60.3 [180]	83.6 [250]	\$1,970	\$3,970	\$7,960	\$47,530	\$6.60 [\$105]	\$88.00 [\$1,395]
3	Automatic Flow Limiting Valves	Note 6	Note 6	100% (100%)	100% (100%)	100% (100%)	100% (100%)	66.6 [199]	90.8 [271]	\$2,170	\$4,310	\$11,420	\$50,750	\$9.50 [\$151]	\$94.00 [\$1,490]
4	Reverse-Return	1.2 [3.6]	10.4 [31.0]	103% (100%)	150% (109%)	99% (100%)	85% (97%)	55.3 [165]	80.0 [239]	\$1,810	\$3,800	\$28,460	\$17,290	\$23.70 [\$376]	\$32.00 [\$507]
5	Oversized Main Piping	7.0 [20.9]	20.9 [62.4]	122% (103%)	173% (112%)	94% (99%)	82% (97%)	45.0 [134]	59.3 [177]	\$1,470	\$2,820	\$12,900	\$7,040	\$10.80 [\$171]	\$13.00 [\$206]
6	Undersized Branch Piping	19.5 [58.2]	NA	142% (106%)	NA	73% (89%)	NA	58.5 [175]	NA	\$1,910	NA	(\$250)	NA	(\$0.20) [(\$3.2)]	NA
7	Undersized Control Valves	8.0 [23.9]	NA	120% (103%)	NA	86% (95%)	NA	58.5 [175]	NA	\$1,910	NA	(\$2,340)	NA	(\$2.00) [(\$32)]	NA
Notes:		<ol style="list-style-type: none"> Annual pump energy based on 90% motor efficiency, 75% pump efficiency, \$0.15 per kWh average, and 650 equivalent full load hours for CHW (with VSD) and 2,100 for HW (without VSD), which are typical of an office building in a climate such as Sacramento, Calif. Energy impact of off-hour operation (e.g., only one AHU on) and transients such as warm-up are not included since the number of hours operating in these modes is relatively small. First costs for balancing valves are based on pricing from a single vendor to ensure pricing consistency. Piping costs based on standard estimating manual.⁶ Balancing costs and costs to clean strainers were obtained from local contractors. Variations in costs of pumps, motors, and VSDs are small and not included. Cooling coil design data, typical for each floor: 79°F (26°C) entering dry bulb, 63°F (17°C) entering wet bulb, 54°F (12°C) leaving dry bulb, 548.8 MBH sensible capacity, 60 gpm (4 L/s) chilled water flow entering at 42°F (5.5°C) and leaving at 60.9°F (16°C), 3.65 ft (1.1 kPa) pressure drop, 6 rows with fins at 8.9 fins per inch (3 fins per mm). Reheat coil design data: Based on 8 in. and 10 in. (200 mm to 250 mm) variable air volume boxes with 1 row coils, 180°F (82°C) entering water temperature, 90°F (32°C) supply air temperature, and flow varying from 0.75 to 1.5 gpm (0.05 to 0.09 L/s). NA = system not analyzed because of the complexity of determining which pipes/valves could be reduced without affecting pump head. The valve differential pressure is zero when the flow is at design, but for the valve to drop flow below design during normal operation, the DP will be the same as Option 1. See body of article for additional discussion. 													

Table 1: Balancing system performance data.

- Usually lower pump head. This is because the reverse-return piping is larger than the normal mains (Figures 1 and 2) and large piping usually will have lower friction rates than smaller piping, which is stepped down as flow reduces keeping friction rates near design limits.

Disadvantages of this option include:

- Added cost of reverse-return piping. The additional cost is significant for systems with large pipe sizes but the difference decreases relative to other options as pipe size decreases, as the cost data in Table 1 indicate.
- Not always practical depending on physical layout of system. The routing of a reverse-return line can be convoluted if coils are scattered about in a non-uniform pattern.

Option 5: Oversized Main Piping

The reverse-return design attempts to equalize the differential pressure in all coil circuits by making flow path lengths similar. This option attempts to equalize DP by reducing the pressure drop of the piping mains by keeping the mains the same size for the entire length of the system, as shown in the

right-hand schematics of Figures 1 and 2.

Advantages of this option (Table 2) include:

- Reduced over-pressurization of control valves close to pumps. The valve differential pressure range is not as uniform as reverse-return but it is significantly improved compared to Option 1. Controllability should not be an issue.
- Lowest pump head/energy due to oversized piping. Pump head for this option in the two example systems is about one-third less than the option with the highest head (Option 3).
- Increased flexibility to add loads since the piping is oversized. If the system is designed with future expansion in mind, this option may be particularly attractive since the designer need not guess where future loads may be added.

The only disadvantage of this design is the added cost of larger main piping. However, the reduced pump power and energy cost can pay for the added cost over time for chilled water systems. In our example California office building, which has relatively few run hours, the payback on energy savings is poor (about 25 years including the impact on chiller energy). In a system with more run-hours, such as a data center that runs 24/7,

the payback can be attractive, particularly when the design's other benefits are considered.

Option 6: Undersized Branch Piping

This option is similar to Option 2 but instead of adjusting a valve to create a pressure drop, pipe size is reduced to increase piping pressure losses in branches with excess pressure. Advantages of this option (Table 2) include:

- Reduced cost of smaller branch piping.
- Reduced over-pressurization and better controllability of control valves close to pumps where piping has been undersized. As discussed later, because of the availability of piping only in standard, incremental sizes, not all coils will be able to benefit from this design option.

This option also has some significant disadvantages:

- Limited effectiveness and applicability due to limited available pipe sizes. This limitation is illustrated in our chilled water system example. Only the piping to the first floor AHU could be reduced (from 2½ in. to 2 in. [64 mm to 51 mm]) without affecting the design pump head. If any other floor piping was similarly reduced in size, pump head would have had to increase.

- High design and analysis cost to determine correct pipe sizing. This limitation is illustrated in our hot water system example. The system is complex, with a reverse-return loop on each floor. To determine which piping could be undersized, a network flow computer program, similar to the one used in this analysis, must be used. Few designers are willing (or paid) to invest in this type of analysis. (It was because of this complexity that this option was not analyzed for the heating system in Table 1.)

Balancing Option		Advantages	Disadvantages
1	No Balancing	<ul style="list-style-type: none"> • No balancing labor • Low first cost and energy use • Coils may be added/subtracted without rebalance 	<ul style="list-style-type: none"> • Imbalance during transients or if setpoints are improper • Control valves near pumps can be over-pressurized, reducing controllability
2	Manual Balance Using Calibrated Balancing Valves	<ul style="list-style-type: none"> • Valves can be used for future diagnosis (flow can be easily measured) • Reduced over-pressurization of control valves close to pumps 	<ul style="list-style-type: none"> • Added cost of calibrated balancing valve • High balancing labor cost • Complete rebalance may be required if coils added/subtracted • Slightly higher pump head due to balancing valve • Coils may be starved if variable speed drives are used without ΔP reset • Slightly higher pump energy depending on flow variations and pump controls
3	Automatic Flow Limiting Valves	<ul style="list-style-type: none"> • No balancing labor • Coils may be added/subtracted without rebalance 	<ul style="list-style-type: none"> • Added cost of strainer and flow limiting valve • Cost of labor to clean strainer at start-up • Higher pump head and energy due to strainer and flow limiting valve • Valves have custom flow rates and must be installed in correct location • Valves can clog or springs can fail over time • Control valves near pumps can be over-pressurized, reducing controllability
4	Reverse-Return	<ul style="list-style-type: none"> • No balancing labor • Coils may be added/subtracted without rebalance • No significant over-pressurization of control valves close to pumps • Usually lower pump head due to reverse-return piping having lower pressure drop than mains (due to larger pipe) 	<ul style="list-style-type: none"> • Added cost of reverse-return piping • Not always practical depending on physical layout of system
5	Oversized Main Piping	<ul style="list-style-type: none"> • No balancing labor • Coils may be added/subtracted without rebalance • Reduced over-pressurization of control valves close to pumps • Lowest pump head/energy due to oversized piping • Increased flexibility to add loads due to oversized piping 	<ul style="list-style-type: none"> • Added cost of larger piping
6	Undersized Branch Piping	<ul style="list-style-type: none"> • No balancing labor • Coils may be added/subtracted without rebalance • Reduced cost of smaller piping • Reduced over-pressurization of control valves close to pumps where piping has been undersized 	<ul style="list-style-type: none"> • Limited effectiveness and applicability due to limited available pipe sizes • High design and analysis cost to determine correct pipe sizing • Reduced flexibility to add coils where piping has been undersized • Coils may be starved if variable speed drives are used without ΔP reset • Slightly higher pump energy depending on flow variations and pump controls
7	Undersized Control Valves	<ul style="list-style-type: none"> • No balancing labor • Coils may be added/subtracted without rebalance • Reduced cost of smaller control valves • Reduced over-pressurization of control valves close to pumps where control valves have been undersized • Improved valve authority which could improve controllability where control valves have been undersized 	<ul style="list-style-type: none"> • Limited effectiveness and applicability due to limited available control valve sizes (C_v) • High design and analysis cost to determine correct control valve sizing • Coils may be starved if variable speed drives are used without ΔP reset • Slightly higher pump energy depending on flow variations and pump controls

Table 2: Summary of advantages and disadvantages.

- Reduced flexibility to add coils where piping has been undersized.
- Coils may be starved if variable speed drives are used on pumps, and this option will have slightly higher pump energy depending on flow variations and pump controls. See the discussion of these issues under Option 2. Whether CBVs or undersized piping are used to increase pressure drop at coils close to the pump, the impact is the same.

Option 7: Undersized Control Valves

This option is similar to Option 6 but instead of undersizing piping, control valves are undersized. Advantages of this option (Table 2) include:

- Reduced cost of smaller control valves. In the chilled water system example, control valves could be reduced from 2 in. (51 mm) (valve flow coefficient, $C_v = 40$) to 1½ in. (38 mm) ($C_v = 25$) on the lower 12 AHUs while maintaining the same pump head.
- Reduced over-pressurization of control valves close to pumps where control valves have been undersized. However, usually it is not possible to apply this approach to all valves, as discussed later.
- Higher valve authority where control valves have been undersized. However, this probably has only a minor impact on controllability since increasing valve authority above 0.5 will have only a minor impact on valve stroke versus flow curves.⁵

Disadvantages of this option include:

- Limited effectiveness and applicability due to limited available control valve sizes. It is theoretically possible to machine customized throttling plugs for each control valve to provide the desired pressure drop, but the costs would be too high. In the chilled water system example, control valve size could be reduced on the lower 12 AHUs but not on the upper floors without increasing pump head.
- High design and analysis cost to determine correct control valve sizing. As with the previous option, a detailed pressure drop analysis is required to determine which control valves may be undersized. The time required for this analysis was the reason this option was not evaluated on the hot water system.
- Coils may be starved if variable speed drives are used on pumps, and this option will have slightly higher pump energy depending on flow variations and pump controls. See the discussion of these issues under Option 2. Whether CBVs or undersized valves are used to increase pressure drop at coils close to the pump, the impact is the same.

Conclusions and Recommendations

While care must be taken in generalizing the results of this study to other applications, we suggest the following guidelines in applying each of the seven balancing options to typical building variable flow hydronic HVAC applications:

1. No balancing. For other than very large distribution systems, this option appears to be the best, combining low first

costs with minimal or insignificant operational problems. The flow imbalances during transients seldom impact coil capacity significantly, and controllability issues due to excess differential pressure are not likely to be a problem on systems of the size analyzed here. Ease of design and commissioning are particularly attractive features of this option.

2. Manual balancing using calibrated balancing valves. This option adds significant first costs for the CBVs and for valve balancing labor. Starving coils with systems that have VFDs is a potential problem, but most likely minor. Still, this option has few operational advantages relative to the cost compared to Option 1.

3. Automatic flow limiting valves. This option results in high first costs and energy costs. The only apparent benefit is a minor one: reducing flow excursions during transients, which the analysis of Option 1 showed to be only a minor problem for the systems analyzed here.

4. Reverse-return. While providing almost true self-balancing and the best controllability, this option has high first costs that are hard to justify, particularly in comparison to Option 5.

5. Oversized main piping. This option should be considered on very large distribution systems where controllability issues with Option 1 are a concern. It should prove to be cost effective from pump energy savings for chilled water systems that run many hours per year. The added flexibility for future additions to the system should also be attractive to large campus-type central plant applications.

6. Undersized branch piping. This option offers low first costs but it has few applications because piping is available only in standard sizes. It reduces system flexibility for future expansion and requires a detailed piping system analysis.

7. Undersized control valves. This is a better approach than Option 6 since it both reduces costs and increases valve authority, but like Option 6 its applicability is limited due to standard equipment sizes. It also requires more piping system analysis than most designers and control contractors are typically willing to invest.

On our own variable flow hydronic system projects, we plan to use Option 1 primarily. However, we will analyze Options 5 through 7 where they appear to be appropriate.

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