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Doubling Down on *Not* Balancing Variable Flow Hydronic Systems

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In October 2002, I co-wrote a *Journal* article[1] on balancing variable flow hydronic systems where we concluded that there was, in general, no need to balance these systems—the systems were self-balancing via the two-way valve controls. In this month's column, I am doubling down: not only do variable flow systems not need to be balanced, they should not be balanced.

To be clear, this discussion relates to hydronic distribution systems with modulating* two-way valves on all or most† coils controlled by closed control loops to maintain, for example, space temperature or supply air temperature. This discussion does not apply to variable flow condenser water systems serving water-cooled AC units because the flow through condenser is not controlled by a feedback loop; those systems must be balanced as if they were constant flow systems. The discussion also assumes systems have variable speed drives on pumps, which help reduce over-pressurization of control valves, although it also applies to simply riding the pump curve on low head systems (e.g., design heads less than about 40 to 60 ft [120 to 180 kPa]).

Methods to Balance Variable Flow Systems

Table 1 is a summary of the most common balancing methods for variable flow hydronic systems. It is drawn from our 2002 article with two changes:

- Two methods (undersizing pipe sizes and undersizing control valves) have been deleted since they are difficult to engineer, they can increase pumping energy

* This discussion probably applies to systems with two-position two-way valves, such as those commonly used with residential and hotel room fan-coils. Over a period of time, these valves should behave like modulating valves, sort of like a slow pulse-width-modulation. But I have never tested this notion, choosing instead to always use modulating control valves.

† The self-balancing nature still works if there is a three-way valve at the end of each branch, a common practice. These valves are in general not needed to provide instantaneous supply water to a valve that just opens; most systems are close-coupled enough that water will be delivered to the coil from the source in minutes if not seconds, fast enough for most applications. But they are useful in engaging the mass of water in the piping system to reduce short-cycling of chillers and boilers (particularly the very light-mass boilers in common use today). However, they also increase piping heat gains/losses and reduce system ΔT.

when peak loads shift to coils closer to pumps, and they are inflexible to future load changes. They are also not commonly used.

- One method was added: pressure independent control valves (PICVs). There was only one manufacturer of PICVs in 2002, and they were expensive and not commonly used. While PICVs were not analyzed with the rigor that the other methods were in our 2002 analysis, their relative performance is included based on recent experience as well as articles such as Kent Peterson's recent Engineer's Notebook column.[2] Note that the PICV referenced here is the type composed of two valves, the first to regulate and maintain a fixed pressure across the second modulating valve, which regulates flow. The other type of PICV that uses a flow meter and modulating valve (analogous to a pressure independent VAV box) discussed in Peterson's column does not offer the same perfect control authority and ease of sizing.

TABLE 1 Ranking balancing methods for variable flow hydronic systems.

OPTION	BALANCING METHOD	RELATIVE RANK (1 = BEST, 5 = WORST)			RECOMMENDED USE
		CONTROLLABILITY	PUMP ENERGY COSTS	FIRST COSTS	
1	No Balancing	5	3	1	Best option for all except high head systems (>90 ft).
2	Calibrated Balancing Valves (CBVs)	4	4	2	Nice for future diagnostics on small (≤1 in. pipe size) low pressure drop coils–but leave them wide open.
3	Automatic Flow Limiting Valves (AFLVs)	5	5	3	Not recommended on any variable flow system
4	Reverse Return	2	2	6	Use if "free," such as floor piping loops
5	Oversized Main Piping	3	1	4	Use on 24/7 chilled water systems and campus distribution systems
6	Pressure Independent Control Valves (PICVs)	1	5	5	Use on high head systems (≥90 ft) for valves near pumps

The rankings in *Table 1* are, of course, subject to application variability and somewhat subjective. But here are my conclusions and recommendations, numbered per the options listed in *Table 1*:

1. Except for systems with high pump heads, no balancing (i.e., letting the control valves automatically self-balance the system) is usually the best choice. The first costs are lowest (no balancing devices, no balancing labor) and, as has been demonstrated with decades of experience, it works well. There is no strict definition of "high" head; *Table 1* lists 90 ft (270 kPa) as the threshold but the range is probably 80 to 100 ft (240 to 300 kPa). When the differential pressure across the valve is high, it must be partially closed just to provide the design flow rate and small changes in valve position result in large changes in flow, so control can be unstable.

2. Manual balancing with calibrated balancing valves (CBVs) offers few benefits other than a small improvement in controllability for valves closest to the pump and the CBVs are useful on small, low pressure drop coils (such as reheat coils) for diagnosing problems that may arise in the future. Without them, it is difficult to deduce flow from coil pressure drop because the pressure drop is so low. For this reason, our company standard is to use CBVs as a combination flow measurement and shut-off valve for small coils with 1 in. (25 mm) pipe size and below. The type of CBV we use is a ball valve with handle so it is just as reliable and convenient as a regular ball valve for isolation duty and not much more expensive. But we don't use it for manual balancing; the valve is specified to be left wide open.[‡] Balancing assumes we accurately know what flow is required (more on this below) and the system can only be manually balanced for one flow condition, the design load condition, and if the pumping system is sized assuming load diversity, as most chilled water systems are, it is not even clear how to balance for that (which coils have reduced flow at the time the system load peaks?). In real systems, the coil that requires the most flow and pressure varies by hour of the day and from season to season, yet we can only manually balance for one condition. In some cases, manual balancing can lead to starved coils at part-load conditions. This is demonstrated in *Figure 1* and *Figure 2*; when the coil close to the pump is balanced

‡ For balancers who don't read specifications, this usually requires two trips to the jobsite. They manually balance the system the first time, then we reject the TAB report and they un-balance the system the second time.

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at full load, it will not be able to achieve design flow when flow through remote coil falls. Flow could be attained by resetting the DP setpoint controlling the pump variable speed drive, but with higher pump energy than needed had the system simply not been balanced. Finally, with manual balancing, if any new coils are added to the system, the entire system must be rebalanced. This problem was what drove me to do the 2002 study: I was the engineer for the initial tenant work on a few floors of a new 20-story high-rise office building and the building engineer insisted we manually balance the hot water reheat coils. I explained that they would need to be rebalanced each time future tenant work was done and coils were added to the system; otherwise the coils on these initial floors would be starved. The 2002 study was done to convince the building engineer no balancing was needed and in fact balancing would be detrimental to future system performance.[§]

3. Automatic flow limiting valves (AFLVs) are not recommended in any variable flow system. This may seem counterintuitive because they were largely invented for balancing variable flow systems. But they offer just one benefit: they provide balanced flow during transients like warm-up or cool-down where all control valves may be wide open. However, as the circuit analysis in our 2002 article showed, this has almost zero value because coil capacity is nonlinear with respect to flow so coil capacity does not change substantially as coils close to the pump have excessive flow while remote coils have reduced flow. Once control valves reduce flow below the AFLV flow limit, the AFLVs do nothing to reduce the differential pressure across the control valve and thus provide no better controllability than having no balancing devices at all. Worse still, they can harm comfort control by limiting flow below what is actually needed to meet the load—more on this issue below.

4. Reverse-return usually reduces pump head and works very to equalize differential pressure across all control valves so it improves controllability. See Steve Duda's Engineer's Notebook comprehensive column[3] on this design. Reverse return usually adds significant first costs due to the need for a dedicated return line, as in the cooling system on left side of *Figure 3*, but there are times when it costs almost the same as direct return, such as the heating system on the right side of *Figure 3*. In the former case, our analysis found that other options were less expensive with equal or adequate performance. In the latter case, reverse return is highly recommended.

FIGURE 1 Manually balanced system–design condition.

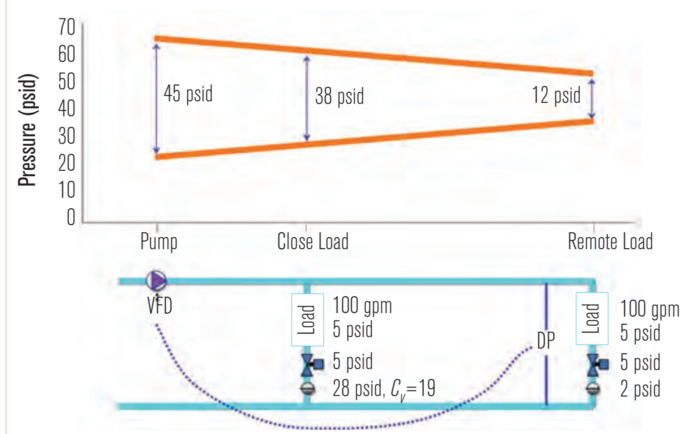
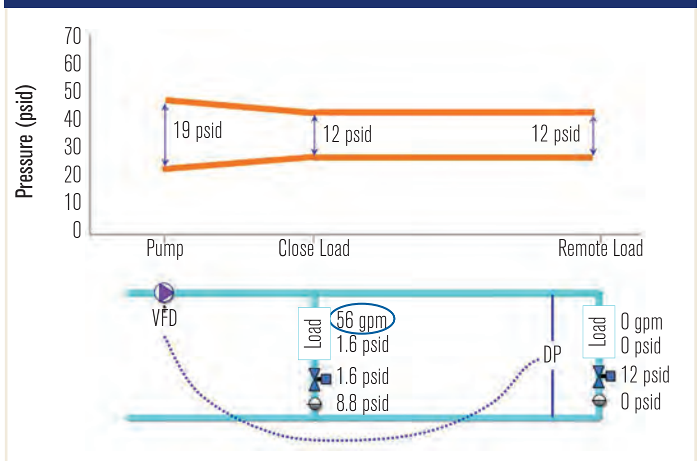


FIGURE 2 Manually balanced system–zero flow at remote coil.



5. Oversizing mains simply means not reducing the size of piping mains as loads drop off. Because of the reduction in pump energy, this option can be cost effective and is recommended for chilled water plants that run 24/7, such as those serving data centers. It also provides greater flexibility for adding or relocating loads in the future, a great benefit to campus distribution systems since campuses often grow in unexpected ways.

6. Pressure independent control valves offer im-

§ The building engineers were unconvinced on that project, but the 2002 paper has been used successfully to convince others who had similar misconceptions.

proved controllability, ease of selection (no valve authority analysis), and reduced tuning and commissioning time. They can also reduce average flow rate compared to systems with very unstable control, as might occur with conventional valves operating against high differential pressure. But they still have a high cost premium, especially on large valves. Peterson's Engineer's Notebook article argues that these benefits outweigh the costs for chilled water systems. I think that will eventually be true as PICV costs continue to fall, and their long-term reliability is proven over time, but I have had too much success with pressure-dependent ball valves to fully endorse using PICVs for all applications. But they are certainly recommended for valves that will experience high differential pressure such as those near pumps with high design head (>80 to 100 ft [240 to 300 kPa]).

FIGURE 3 Reverse return systems.

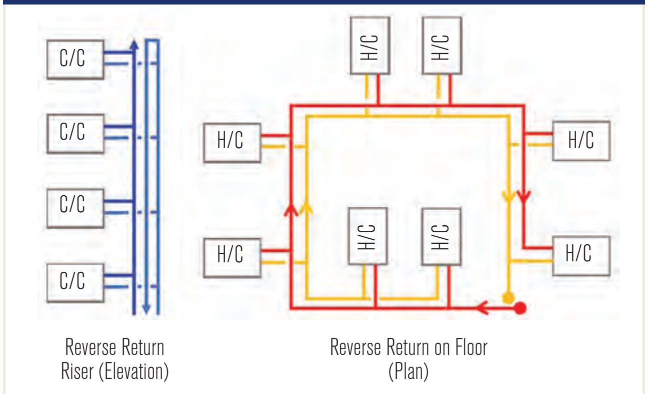


TABLE 2 Partial reheat coil schedule vs. submittal data.

HEATING COIL								SUBMITTAL		RATIO
ROW	AIRFLOW (CFM)	EAT (°F)	LAT (°F)	HEATING CAPACITY (MBH)	EWT (°F)	LWT (°F)	WATER FLOW (GPM)	LWT (°F)	WATER FLOW (GPM)	
2	540	57	95	22.2	140	110	1.5	128.0	3.7	251%
2	780	57	95	32.0	140	110	2.1	118.7	3.0	142%
2	375	57	95	15.4	140	110	1.0	128.1	2.6	254%
2	500	57	95	20.5	140	110	1.4	126.2	3.0	219%

Doubling Down

So, with this background, let me explain my "doubling down" comment above that variable flow systems should not be balanced. By "balanced" I am referring to Options 2 (CBVs) and 3 (AFLVs) above, and to Option 6 (PICVs) for PICVs that have flow limiting capability. Using these options can actually cause more harm than good. Why? Because they assume the specifier of the valve actually knows what the required maximum flow rate is. This is seldom true with any certainty.

First, engineers all know there is significant uncertainty in load calculations. There are numerous load calculation programs and they all give different results even with the exact same inputs. Add to that, thousands of input variables are required, some of which are not known with any certainty, and many change over time. For example, the u-value of that metal stud wall or aluminum framed window may vary substantially depending on construction techniques. Do we really know the minimum outdoor air rate that will come through that damper given the difficulty balancing and controlling outdoor air intakes? Do we really know the server load in that computer room, and will it be the same three years from now when every server has been replaced? Load calculations from 10 competent engineers on the same building will have 10 different results that are very unlikely to be within 10% and can vary as much as 100% depending on assumptions and level of conservatism. Yet we choose an AFLV that provides, for example, 2.67 gpm [0.606 m^3/h], one of many possible flow rate selections with three significant digits from a major AFLV manufacturer. Three significant digits when the actual required flow might be half or double what is scheduled!

Added to this uncertainty is the incorrect way coils are sometimes selected. Here are three real-life examples:

Case 1: Incorrect Assumed ΔT

It is very common to assume a constant temperature difference (ΔT) when scheduling coils. This makes no sense from a design standpoint and I have been pushing against it for years (e.g., my eight-row, 10 to 12 fins per inch [2.0-2.5 mm fin spacing] chilled water coil mantra explained in a December 2011 *Journal* article).[4] But this can also lead to errors on small coils (e.g., VAV box reheat coils) where there are no or few coil options that might be tweaked to deliver the specified ΔT. *Table 2* shows a partial VAV box reheat coil schedule. The designers specified two-row coils but also assumed a ΔT of 30°F [17°C], which over-specifies the coil selection; it also was a poor assumption with the low hot water supply temperature (140°F [60°C]).

FIGURE 4 Screenshot of a generic VAV box schedule from an actual job that leaves it to the contractor to make VAV box selections based on the sum of diffuser airflow rates on drawings. This results in highly inaccurate hot water flow rates.

SINGLE DUCT VAV TERMINALS															
				COOLING		HEATING		REHEAT COIL							
TYPE	INLET SIZE (")	CASING DIMENSION (L"xW"xH")	MFR / MODEL NO.	MAXIMUM AIRFLOW	MINIMUM AIRFLOW	MAXIMUM AIRFLOW	MINIMUM AIRFLOW	MAXIMUM AIRFLOW	MAXIMUM COIL CAPACITY (MBH)	WATER FLOW RATE (GPM)	EWT (°F)	LAT (°F)	COIL MAX APD (IN WG)	COIL MAX WPD (FT WG)	COIL ROWS
B	6	20X12X8		400	60	160	60	160	6.6	0.5	180	91.8	0.60	0.2	1
C	8	20x12X10		700	105	280	105	280	10.1	0.7	180	87.4	0.31	0.1	1
D	10	20X14X12.5		1100	165	440	165	440	16.0	1.1	180	87.5	0.31	0.3	1
E	12	20X16X15		1600	240	640	240	640	24.5	1.7	180	89.2	0.40	0.8	1
F	14	24X20X17.5		2200	330	880	330	880	32.5	2.2	180	88.0	0.50	1.4	1
G	16	24X20X18		3000	450	1200	450	1200	40.3	3.2	180	85.1	0.50	1.9	1

The actual performance from the manufacturer's submittal is shown on the right side of *Table 2*. The actual required flow rates are 142% to 254% higher than what was scheduled. This project's specifications called for AFLVs. Unfortunately, the contractor selected the AFLVs off the scheduled flow rates, not those in the submittal data, a common error. Heating problems resulted. The fix was two-fold: increase the hot water supply temperature (very fortunately, the 140°F [60°C] hot water supply temperature was a designer choice; the condensing boilers could generate as high as 180°F) and remove the AFLV flow restricting cartridges to unbalance the system. The latter would have been unnecessary if the hot water supply temperature had been increased high enough, but the AFLV flow rates were all wrong so removing the cartridges allowed lower hot water temperatures, which reduces piping losses and improved boiler efficiency.

Case 2: Generic VAV Box Schedules

Sadly, some designers reduce engineering costs by pushing VAV box selection onto the contractor; instead of scheduling each VAV box separately with the required airflow rates, coil conditions, etc., the designer provides only the generic schedule in *Figure 4*, leaving it to the contractor to make VAV box selections based on the sum of diffuser airflow rates on drawings. The VAV box selection is entirely based on the maximum cooling airflow rate. There are dozens of flaws with this concept, but in the context of this discussion, it incorrectly assumes that all VAV boxes of a certain inlet size have the same heating airflow, the same supply air temperature, and the same hot water flow rate. This is, of course, not true. The required rates can vary from one-third to three times the flow rate scheduled. This same project included specifications that required that coils be balanced to ±5%!

Case 3: Simple Schedule Error

In this example, the engineer made what was likely a cut-and-paste error that was not noticed during the design phase or during the construction phase. In the underfloor terminal box schedule in *Figure 5*, all coils are scheduled to have the same 0.3 gpm (0.07 m^3/h) hot water flow rate. This flow rate was even low for the small coils (55°F [30°C] ΔT) but it was clearly way too low for the larger coils (over 100°F [55°C] ΔT). Unfortunately, the contractor installed AFLVs (accepted by the engineer who actually specified CBVs), guaranteeing that every coil would get too little flow. When the building could not be properly heated, many fixes were tried including replacing pumps and adding another boiler, all of which were both ineffective and unnecessary. The simple fix was to remove the flow limiting cartridges from the AFLVs.

Conclusions

There are some applications where reverse return (Option 4) and oversized mains (Option 5) make sense for variable flow hydronic systems. Calibrated balance valves (Option 2) are convenient for small coils for future diagnostics but they should not be used for

FIGURE 5 Screenshot of an underfloor terminal box schedule from an actual job. The 0.3 gpm hot water flow rate was low for the small coils, but way too low for the larger coils.

UNDERFLOOR FAN POWER BOX SCHEDULE

SYMBOL	MANUFACTURER MODEL & SIZE	COOLING MAXIMUM CFM	COOLING MINIMUM CFM	HEATING CFM	MAX APD (IN.WG.)	MOTOR HP	POWER	FULL LOAD AMPS	REHEAT COIL 1 ROW OR 2 ROW	REHEAT COIL FLOW (GPM)	REHEAT COIL HEATING CAPACITY (MBH)	REHEAT COIL FLUID PD (FT.WG.)
UFT 1-1		380	300	380	0.08	1/2	120V / 1ø	2.6	1	0.30	8.2	0.07
UFT 1-2		1010	300	380	0.12	1/2	120V / 1ø	2.6	1	0.30	8.2	0.07
UFT 1-3		490	300	380	0.13	1/2	120V / 1ø	2.6	1	0.30	8.2	0.07
UFT 1-4		380	300	380	0.08	1/2	120V / 1ø	2.6	1	0.30	8.2	0.07
UFT 2-1		850	300	520	0.13	1/2	120V / 1ø	5.4	1	0.30	11.0	0.09
UFT 2-2		1730	645	720	0.40	2ø1/2	120V / 1ø	7.8	1	0.30	15.6	0.24
UFT 3-1		870	300	520	0.13	1/2	120V / 1ø	5.4	1	0.30	11.0	0.09
UFT 3-2		380	300	380	0.14	1/2	120V / 1ø	2.6	1	0.30	8.2	0.07
UFT 3-3		250	150	175	0.0.4	1/2	120V / 1ø	2.6	1	0.30	8.2	0.07

manual balancing. But there are no variable flow system applications where automatic flow limiting valves (Option 3) are recommended; moreover, I strongly recommend they never be used because they are inflexible in cases where design flow rates are incorrect, which is likely. For low head pumping systems, I recommend direct return with no balancing (either Option 1 or Option 2 with balance valves wide open), a design I have used successfully for over 30 years. But for high head systems with direct return, pressure independent valves should be used for valves close to the pump where differential pressure can be expected to be high (>80 ft [240 kPa]).

As costs fall for PIVCs, and they prove to be reliable over time, Peterson's conclusion that they should be used in all chilled water systems may well prove to be true, perhaps for all hot water systems as well. But PICVs that have flow limiting capability can suffer the same problems as AFLVs; accordingly, the flow limit setpoints should be cranked up to the maximum possible. Doing so will have no impact on the pressure independence and controllability of the valve but it allows the flexibility of providing higher flow rates should the design flow rate be incorrect.

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