What They Found Along the Way

Columnists Explore Lessons Learned

ATLANTA—A few years ago, Steven T. Taylor, P.E., Fellow ASHRAE, had just finished writing a six-part series on chilled water plants for *ASHRAE Journal*. The series proved popular, so then-editor Fred Turner asked Taylor if he would be willing to write a column similar to Fellow ASHRAE Joe Lstiburek's Building Sciences column. Taylor responded that he didn't have enough time, but he suggested forming a group of authors to write it.

That idea evolved into the Engineer's Notebook column written by consultants who were also experienced authors: Taylor, Stephen W. Duda, P.E., BEAP, HBDP, HFDP, Fellow ASHRAE; Daniel Nall, P.E., BEMP, HBDP, Fellow/Life Member ASHRAE; and Kent Peterson, P.E., BEAP, Presidential Member/Fellow ASHRAE. The multi-author format has worked well.

"I like that we compiled a team with different perspectives from different parts of the country where the conventions of engineering practice may vary a little," says Duda.

The column's mission was to cover practical technical topics that might not be covered in full-length articles: what the authors had learned along the way, basic engineering issues, and things that people should know but might not.

The four columnists presented their favorite columns at the ASHRAE Winter Conference in Las Vegas on Feb. 1. The seminar, which was initiated by Duda, was partially motivated by a desire to "give back to others what I have learned along the way."

The four columns follow with author comments.

"I wanted to popularize one of the results of our analysis: using oversized reheat coils in VAV boxes. Doing so saves significant energy and is very cost-effective, but it would be even more cost-effective if this was a standard option from more VAV box manufacturers, which would reduce costs."
~Steven T. Taylor, P.E. Originally published in July 2015

Steven T. Taylor

VAV Box Duct Design

BY STEVEN T. TAYLOR, P.E., FELLOW ASHRAE

VAV systems are the most common HVAC system for commercial buildings, but design practices vary widely around the country and even among design firms in a given area. Some of the variation is due to local construction practices and labor costs, but most of the variation, in the author's experience, is due simply to how engineers are taught by their mentors in their early years of practice; design techniques and rules-of-thumb are passed down through the generations like family cooking recipes with little or no hard analysis of whether they are optimum from a life-cycle cost perspective.

This month's column compares various VAV box inlet and outlet duct design options including their impact on first costs and pressure drop. It focuses on single duct VAV reheat systems, but most of the principles apply to other VAV system variations, such as dual duct and fan-powered box systems. First cost data are based on San Francisco Bay Area contractor sell prices, which are higher than most other areas due to high labor costs.

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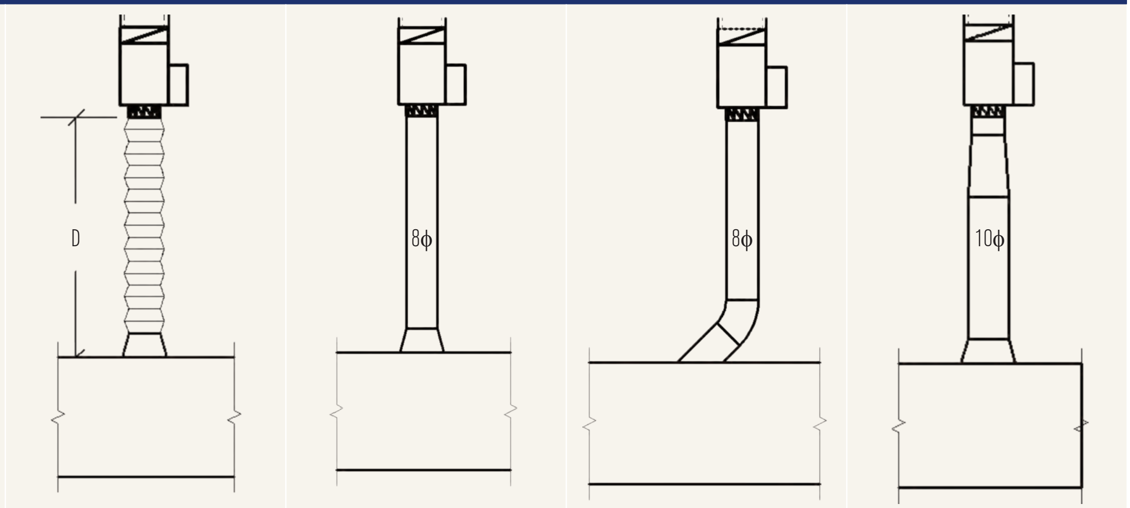
TABLE 1 VAV box inlet ducts off rectangular main (based on 8 in. inlet box, 630 cfm, 1,500 fpm duct main velocity).

Option



		A. Conical, Flex			B. Conical, Hard			C. 45°, Hard			D. Oversized Conical, Hard		
Dimension D (ft)		5	10	15	5	10	15	5	10	15	5	10	15
Relative First Cost		Base	Base	Base	$55	$75	$90	$160	$180	$200	$210	$235	$260
Total Pressure Drop (in. w.g.)	Tap	0.25	0.25	0.25	0.25	0.25	0.25	0.18*	0.18*	0.18*	0.16	0.16	0.16
	Duct	0.06	0.13	0.20	0.03	0.06	0.10	0.03	0.06	0.10	0.01	0.02	0.03
	Taper	–	–	–	–	–	–	–	–	–	0.01	0.01	0.01
	Total	0.31	0.38	0.45	0.28	0.31	0.35	0.21	0.24	0.28	0.18	0.19	0.20
Application Note		1	1	1	2	2	2	3, 4	3, 4	3	4, 5	4, 5	5, 6

1. Not recommended
2. Recommended for most VAV boxes but not at low velocity main ducts or for "obviously critical" VAV boxes
3. Recommended when VAV box is at a 45° angle to main (not shown in option figure)
4. Recommended for "obviously critical" VAV boxes
5. Recommended at low velocity main ducts (see *Figure 3*)
6. Recommended for VAV boxes that are greater than about 15 ft from main

*Neither the ASHRAE Duct Fitting Database nor the SMACNA HVAC Systems Duct Design Manual includes this tap type. Pressure drop is estimated by author based on comparison of other similar fittings.

Pressure drop data were calculated using ASHRAE's "Duct Fitting Database"[1] or SMACNA's *HVAC Systems Duct Design*.[2]

VAV Box Inlet Duct Design

Table 1 shows typical VAV box connections to the duct main with first cost premiums, estimated pressure drop for the listed example, and recommended applications.

Option A (conical tap with flexible duct) is the least expensive option, but it is not recommended for any applications for the following reasons:

- It results in the highest pressure drop, usually even higher than that shown in *Table 1*. The pressure drop shown in the table is for perfectly straight flex duct, which has a roughness factor of about 2.1 relative to hard sheet metal duct.[2] But most real applications will have some drooping at a minimum and often will have bends or offsets due to boxes being misaligned with the main duct tap.
- Even when straight, the roughness of the flexible duct can cause errors in velocity pressure (VP) sensor readings by the boxes flow sensor, as shown in *Figure 1*. When flex duct is kinked, the impact is even worse.
- Flexible duct is largely transparent to breakout noise so any noise generated by partially closed VAV box dampers can be readily radiated to the space. Conversely, hard round duct is highly resistant to breakout noise.

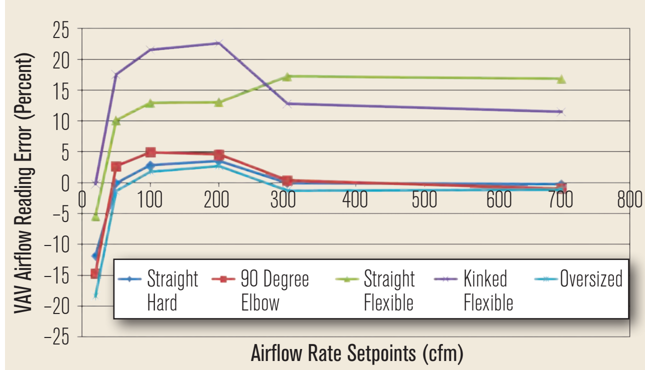
Option B is also a low cost option. It has a higher pressure drop than Options C and D but much lower first costs. The added costs of Options C and D would only be cost effective if they were applied to only the "critical zones," which are the zones that require the highest fan speed and pressure. All other zones will have excess pressure available and thus any pressure drop savings from using a more efficient inlet duct design will be throttled by the VAV box damper. But the critical zone will vary due to variations in internal loads, weather, sun angle, etc. It is possible for most systems that 50% or more of the zones can be the most critical at any given time (see Figures 6 through 8 in Taylor & Stein[4]). This would require that the first cost penalty of Options C or D would apply to many zones, not just one.

For example, simulations of a 60,000 cfm (28 000 L/s) VAV system serving an Oakland office building showed that adding 0.15 in. w.g. (38 Pa) to the fan design pressure for Option B versus D increased energy costs only a few hundred dollars per year. That would result in an excellent payback if one particular zone was always the critical zone and Option D were only applied to it. But if all 70 zones in the system were designed using Option D, the payback would be 75 years. To get a 15-year payback, no more than 20% of the potentially critical zones could be ducted using Option D, but the designer would have to figure out in advance which zones are potentially critical. Option C has similar economics: it is less expensive than Option D but not as efficient.

So instead of using Options C or D at all zones, they should be used in special cases only:

- Use Option C for VAV boxes that are at a 45° angle to the duct main. This eliminates the cost and pressure drop of the 45° elbow shown in *Table 1.*
- Use either Option C (a bit less expensive) or D (a bit more efficient) for "obviously critical" zones. This will require some engineering judgment on the part of the designer. Examples include zones that are a long distance from the main or zones that are expected to be at high loads for many hours per year, such as those serving an equipment room.
- Use Option D for zones tapping into low velocity mains. One technique for sizing duct mains is to "start fast and end slow," as shown in the top half of *Figure 2.* Rather than using conservative duct design sizing techniques, such as a constant 0.1 in. w.g. per 100 ft friction rate (80 Pa per 100 m) for duct mains, this technique uses a higher starting velocity and friction rate and then keeps the duct main the same size for long distances, e.g., up to 60 ft (18 m). This results in lower first costs due to eliminated fittings but results in similar overall pressure drop. The pressure drop of the taps to VAV boxes also benefits from the lower velocities at the end of the duct main, but only to a point. As shown in *Figure 3,* when the duct main velocity is much lower than the velocity in the tap (less than about 60%), the pressure drop through the tap starts to increase. So VAV boxes at the end of the main should use Option D.

FIGURE 1 VAV sensor error under different inlet conditions for 8 in. inlet VAV box (Figure 7 from RP-1353 Final Report[3]).



Note that none of the options includes a manual volume (balancing) damper upstream of the VAV box. They are never necessary in VAV systems with pressure independent controls; the VAV box controls provide continuous, dynamic self-balancing.

Note also that Option D shows a tapered reducer at the inlet to the box. Many engineers will include two or three duct diameters of inlet-sized duct between the reducer and the box to ensure that the velocity profile at the velocity pressure sensor is uniform. This is unnecessary. As shown in *Figure 1,* the "oversized" inlet resulted in the same VP accuracy as the straight "hard" inlet. Furthermore, in the research project upon which *Figure 1* is based, the 10 × 8 reducer was only 8 in. (200 mm) long, much more abrupt than the taper shown in Option D of *Table 1.*

Figure 1 also shows that even having a 90° elbow directly in front of the VAV box has little impact on VP sensor accuracy. VAV box manufacturer's installation instructions encourage using SMACNA's recommended three duct diameters of straight duct at the inlet but also note

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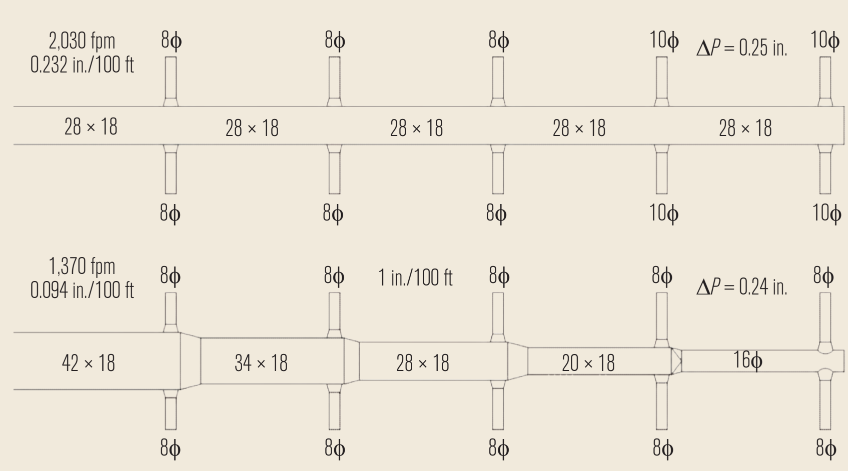
that their VP sensors are in fact designed to allow for poor inlet conditions that frequently occur due to space constraints.

FIGURE 2 VAV duct main design: "Start fast and end slow."



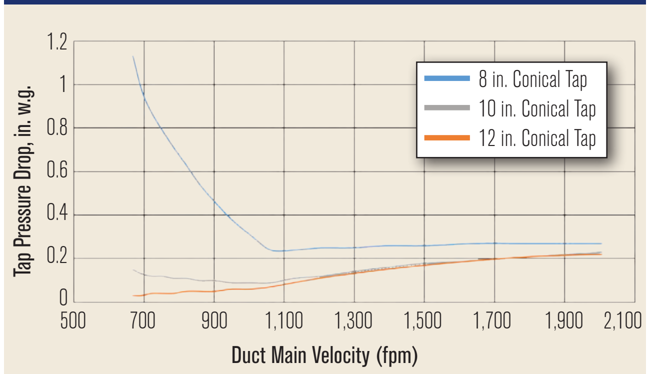
VAV Box Discharge Duct Design

Table 2 shows options for discharge plenums from VAV boxes. Both applications with and without 1 in. (25 mm) duct liner are shown. Duct liner is not allowed for some occupancies (e.g., hospitals) and is discouraged due to indoor air quality concerns in consistently humid climates, but it is still standard practice in many areas of the country. The cost of liner is generally close to being net first cost neutral with the same duct outside dimensions (OD) since the unlined duct must be externally insulated in the field.

Option B is the least expensive lined duct option. The OD of the discharge plenum matches the dimension of the box outlet so that a simple "S and drive" duct connection can be made without any fittings. This has the disadvantage of increasing plenum velocity and the liner also creates an abrupt reduction in free area right after the coil. To avoid those losses, Option C includes a 1 in. (25 mm) flange around the VAV box discharge so that the inside dimensions (ID) of the plenum matches the coil dimensions. (This could also be a standard duct transition, but the flange is usually a bit less expensive and takes up less space.) Unfortunately, the flange is expensive when shop fabricated and it is not available as an option from most VAV box manufacturers. Its costs can be offset, however, if it avoids the need for shop fabricated square-to-round taps to diffusers; the larger plenum height allows for larger standard diffuser taps.

But a better option in any case is to oversize the heating coil by using the next-size-up box and coil instead of the box and coil that comes standard with the inlet size. In this case, the box and coil are for a standard 10 in. (250 mm) VAV box but the damper and velocity pressure sensor are still 8 in. (200 mm). This is a "special" order from most VAV box manufacturers but the cost is usually the same price as the larger box; in other words, the box in this example with an 8 in. (200 mm) inlet but the box/coil of a standard 10 in. (250 mm) box costs the same as a standard 10 in. (250 mm) box. Care must be taken to make VAV box equipment schedules very clear of the design intent since this is non-standard construction. For instance, include coil size in the schedule and include a note in the "Remarks" column noting the non-standard construction.

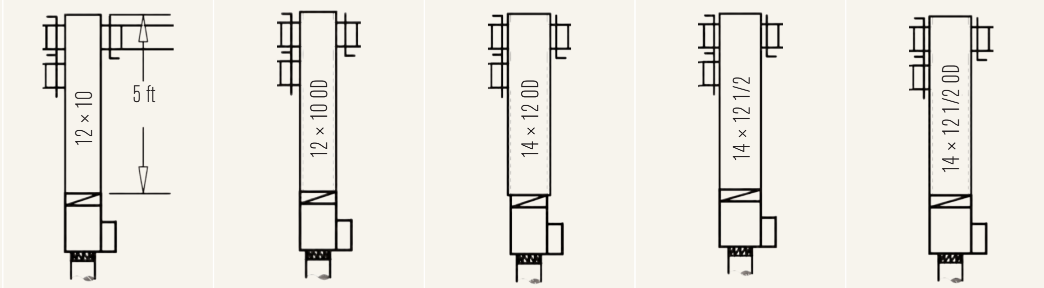
FIGURE 3 Tap pressure drop vs. duct main velocity (from ASHRAE Duct Fitting Database).



This oversized box/coil option is recommended with and without duct liner. An option with a discharge flange like Option C is also possible but it is not likely to be cost effective because the pressure drop of the oversized plenum is already low.

One valuable side benefit of Options D and E is the improved waterside performance of the coil resulting from the increased heat transfer area: the coil leaving water temperature with the oversized coil is about 10°F

TABLE 2 VAV box discharge ducts. Based on 8 in. inlet box, three 210 cfm diffuser taps.



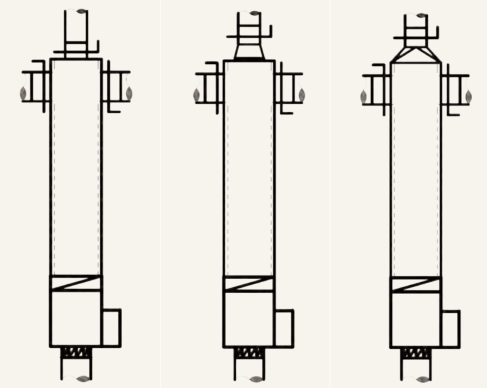
	Option	A. Unlined Plenum	B. Lined Plenum, Constant OD	C. Lined Plenum, Constant ID	D. Unlined Plenum, Oversized HW Coil	E. Lined Plenum, Constant OD, Oversized HW Coil
	Relative First Cost	Base	$55	$285	$90	$145
Total Pressure Drop (in. w.g.)	HW Coil	0.30	0.30	0.30	0.15	0.15
	Liner Edge	0.00	0.02	0.00	0.00	0.01
	Plenum	0.00	0.02	0.01	0.00	0.01
	Diff. Tap	0.05	0.09	0.05	0.03	0.05
	Total	0.35	0.43	0.36	0.18	0.22
	Application Note	1	1	1	2	3

1. Not recommended
2. Recommended where acoustic considerations are met without liner or liner is not allowed/desired
3. Recommended where liner is required for acoustics and allowed by code and local practice

to 15°F (5.5°C to 8°C) lower than for the standard coil. This reduces flow rates, pump size, and pipe sizes, and can improve the efficiency of condensing boilers. It can also allow low temperature water systems, such as those using condenser heat recovery, to work effectively with a two-row coil.

Table 3 shows three options for tapping the end of the discharge plenum to serve a diffuser. Many engineers forbid end taps because of perceived high pressure drops. In fact, according to the "Duct Fitting Database," the pressure drop even for a straight tap out the end (Option A) is very low due to the low velocities in the plenum and duct to the diffuser. The straight end tap also will have a lower pressure drop than the side taps, 0.01 in. w.g. (2.5 Pa) versus 0.05 in. w.g. (12.5 Pa) in this example, so the volume damper in the end tap will have to be throttled. Regardless,

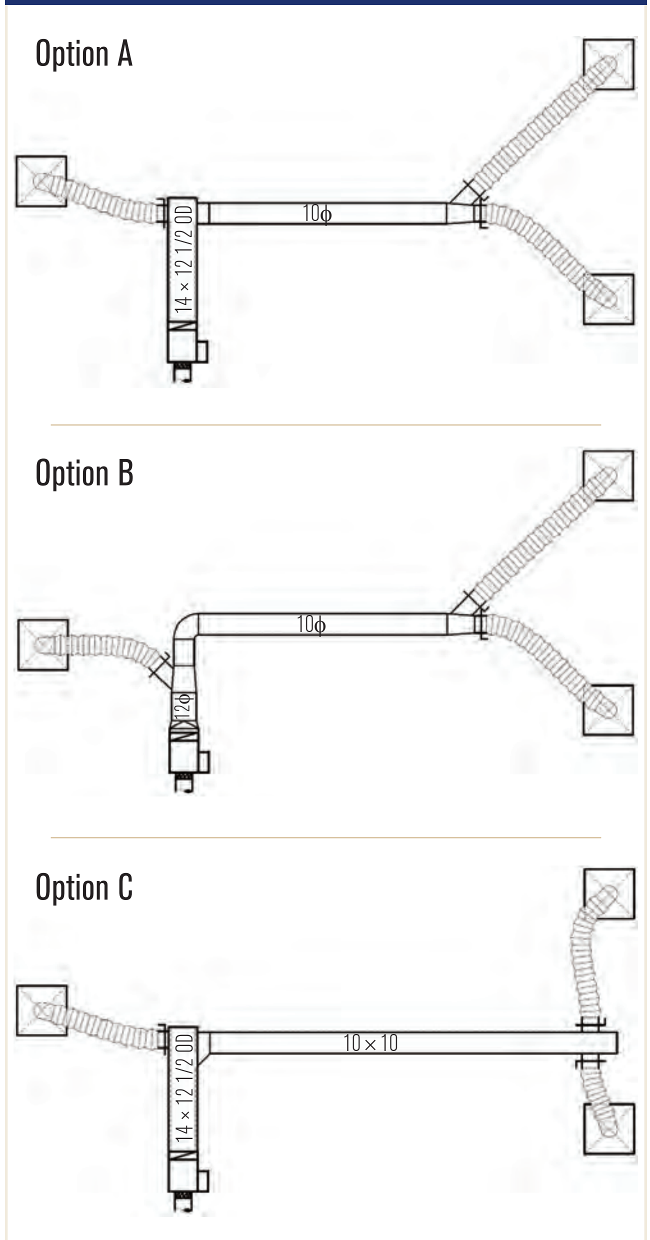
TABLE 3 VAV box diffuser end taps.



Option	A. Straight Tap	B. Conical Tap	C. Square-to-Round
Relative First Cost	Base	$20	$80
Pressure Drop (in. w.g.)	0.01	0.00	0.00
Application Note	1	2	2

1. Recommended where end-taps must be used due to space constraints
2. Not recommended

FIGURE 4 Duct design options from VAV boxes. Option A (Top): Plenum plus round duct. Option B (Center): All round duct. Option C (Bottom): Plenum plus rectangular duct.



end taps should be avoided unless mandated by space constraints for two reasons:

- One of the acoustical benefits of the plenum (end reflection) is at least partially lost.
- Airflow balance among the diffusers tapped out of the sides and that tapped out the end is not accurately maintained over the full range of VAV box airflow rates. This is because the pressure drop behavior of the side taps is not linear with airflow. So at low airflow rates, proportionally more air will go through the end tap than through the side taps. But the effect is very small so unlikely to cause any comfort problems.

Figure 4 shows three examples of duct design from VAV boxes, described as follows:

- Option A has a lined (or unlined) discharge plenum per *Table 2*. The plenum should always be 5 ft (1.5 m) long, or multiples of 5 ft (1.5 m) if added length is needed for acoustics, so that standard coil-line straight ductwork can be used to reduce costs. Taps to outlets should be near the end of the plenum to gain its full acoustical benefits and to avoid “cushion head” losses. Straight taps should be used; conical taps have negligible pressure drop benefit but add to first costs and may not always fit into the side of the plenum whose height is generally determined by the VAV box dimensions. For diffusers close to the plenum, the tap should include a volume damper; a straight tap with damper is a standard off-the-shelf item. For diffusers that are more remote from the plenum, a round branch duct is used with reducing wyes with volume dampers at each

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diffuser. Some contractors will find it more cost effective to duct all diffusers independently from the plenum since it eliminates fittings and gangs volume dampers in a central location for ease of balancing. With this option, all ductwork is round except for the discharge plenum. This lowers costs not only because round duct costs less than rectangular duct, but also because it is easier to make coordination offsets in the field. For instance, if the workers find a sprinkler line or cable tray in the way of a hard round duct run, adjustable elbows (with sealed joints) can be easily inserted in the field.

- Option B eliminates all rectangular ductwork.

This design is often favored by contractors that do not have coil-lines for fabricating rectangular plenums. It increases the number of joints and fittings, but reducing wyes and adjustable elbows are easily obtained off-the-shelf. The one big disadvantage of this design is that it loses the acoustical benefit of the discharge plenum. The plenum is beneficial acoustically even if unlined.

- Option C is almost the opposite of Option B: it is composed of all rectangular duct except for flexible duct to diffusers. This is usually the most expensive design because rectangular duct costs more than round duct and it is less flexible to making field changes, e.g., offsetting to miss a sprinkler pipe or cable tray requires one or two shop fabricated fittings. In addition to the shop and material cost, there is usually an added labor cost to deliver the materials and possibly a time delay while it is being fabricated. Option A is recommended for almost all applications.

Conclusions

This column summarizes various duct design options for VAV boxes, both upstream and downstream and makes recommendations based on lowest estimated life-cycle costs. The recommendations are generally practical and easy to implement.

Acknowledgments

The author would like to thank Eddie Patterson and Todd Gottshall of Western Allied Mechanical for providing the cost estimates presented in this article.

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"I continue to see [reverse-return] dismissed out-of-hand or overlooked in situations where it would be beneficial, and misapplied where it isn't beneficial. Due to the popularity of commissioning in the past few years, I have the opportunity to see drawings and specifications produced by a large number of other engineering firms besides my own, and this is an item I often see overlooked."
~Stephen W. Duda, P.E. Originally published in August 2015

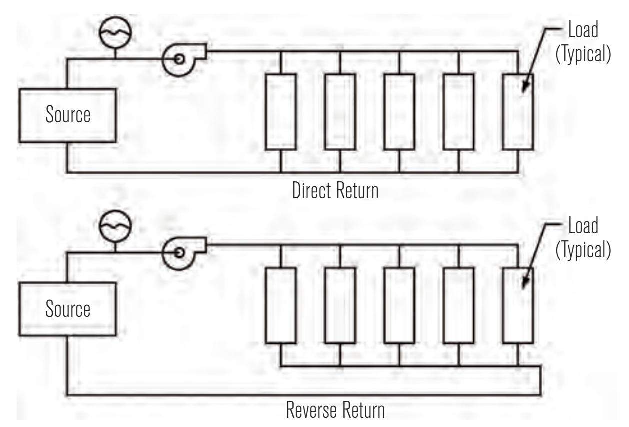
Stephen W. Duda

Reverse-Return Reexamined

BY STEPHEN W. DUDA, P.E., BEAP, HBDP, HFDP, FELLOW ASHRAE

A common perception is that a reverse-return hydronic piping configuration uses more piping and, therefore, is more expensive than its direct-return counterpart. For example, a hydronics primer recently published in *ASHRAE Journal*[1] briefly discusses direct versus reverse-return piping arrangements and quickly reaches that conclusion. While the cost disadvantage of reverse-return is true in some instances, this column presents a case that reverse-return doesn't always add piping length and system cost, depending on system configuration. In addition, this author has found reverse-return is sometimes overlooked or dismissed out-of-hand when it offers tangible benefits and could easily have been implemented at no net cost to the project. So, a goal of this column is to encourage pipe system designers to explore and consider reverse-return in further detail.

FIGURE 1 Simplified direct- and reverse-return two-pipe systems.[3]



To review, a direct-return system (*Figure 1*) is one in which the terminal nearest the supply source has both the shortest supply water path and shortest return water path to and from the source, while the terminal most remote from the supply source has both the longest supply water path and longest return water path. This can result in significantly different network piping losses from one terminal unit to another, requiring some type of correction (e.g., balance valves, active flow-limiting devices, or perhaps pressure-independent control valves) to keep the system balanced and piping losses equal.

By contrast, a reverse-return system is one in which the terminal nearest the supply source has the shortest supply water path but the longest return water path, while the terminal most remote from the supply source has the longest supply water path but shortest return water path. A reverse-return system means that the sum total of supply and return piping losses are approximately the same throughout the water system, making for a more even water flow to all terminals without additional correction. The reverse-return system is nearly self-balancing because the hydraulic distance traveled by the fluid is close to the same regardless of which terminal coil a given volume of fluid flows through; or said another way, reverse-return creates approximately equal hydraulic resistance through each flow path.

Benefits of Reverse-Return

The primary advantage of reverse-return is elimination of balance valves at each terminal coil (VAV reheat box, finned-tube convector, fan-coil unit, chilled beam, etc.) and the testing, adjusting and balancing labor cost

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FIGURE 2 Single-story office building using a direct-return piping arrangement.

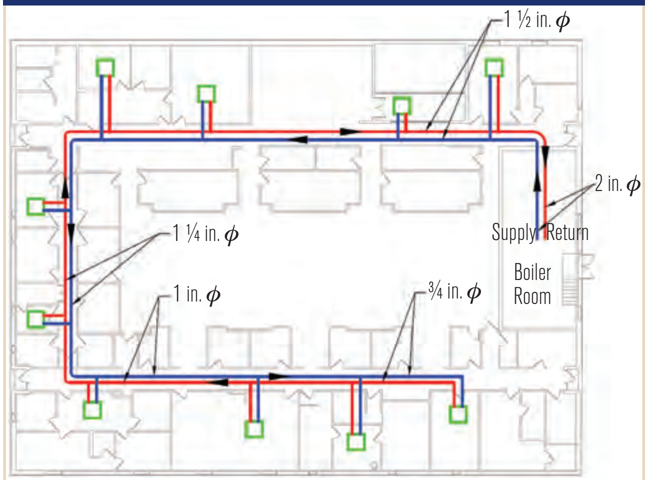
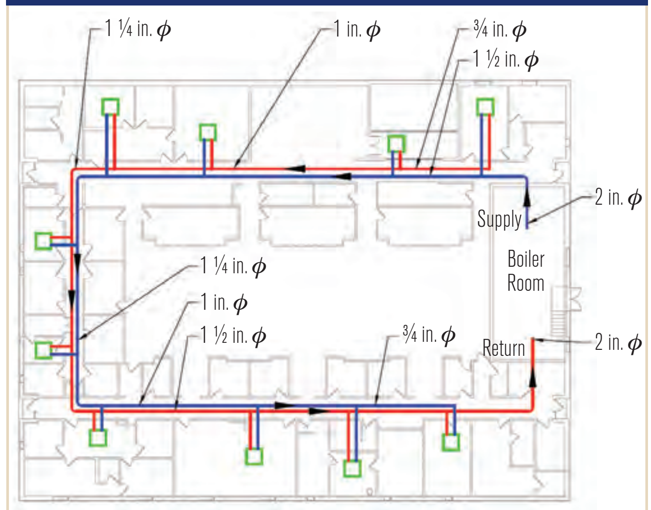


FIGURE 3 Single-story office building using a reverse-return piping arrangement.



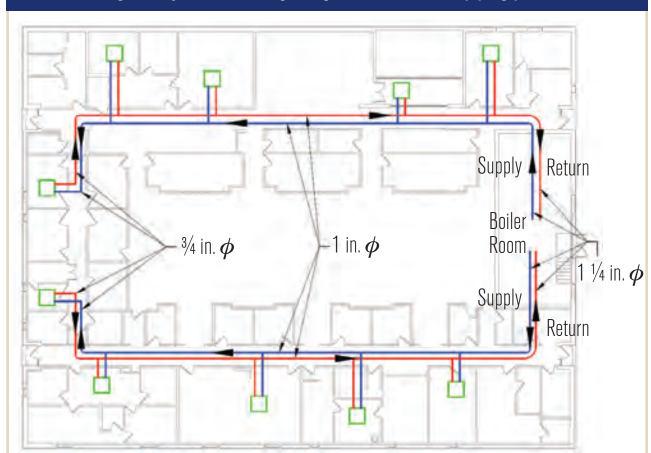
associated with that. Since the material and labor cost of many balance valves and the labor cost of manual balancing is eliminated, a small cost add for the reverse-return piping can be offset by the reduction in balance valves and manual balancing, or a reduction in active flow-limiting devices, or allow the use of standard control valves in lieu of the more expensive pressure-independent control valves. (The remainder of this column will assume balance valves and manual balancing for direct-return systems; but the reader can interpret these and other forms of system balance correction as well.)

While reverse-return systems are close to self-balancing, they are not exactly. Anything that creates a difference in the resistance of the supply or return piping from one flow path to another will change the resistance of that path relative to the others and affect the balance slightly, as would a different type of coil or differences in the coil runout piping itself (length, type, number of bends, size, etc.).

But, these differences are minor and are easily corrected by a standard coil control valve. If one coil path receives a little more than its fair share of fluid flow, the control system will eventually correct for it by repositioning the coil control valve. It is only those systems with large hydraulic differences from leg to leg, such as those found in direct-return systems with some coils very near the source and other coils very remote, that may exceed the pressure range ability of the control valve to perform adequately.

Not convinced that reverse-return allows the elimination of balance valves? See Taylor/Stein.[2] In fact, that article argues that in many cases balancing is not necessary in

FIGURE 4 Single-story office building using two direct-return piping paths.



direct-return systems either, and some engineering firms routinely don't require balancing of direct-return systems. However, that concept makes other engineers and designers uncomfortable, and in this author's experience, the majority of direct-return systems in design practice today still feature coil balance valves.

In any case, the Taylor/Stein article clearly demonstrates that minor differences in balance from leg to leg, such as that found in reverse-return designs, do not warrant balance valves; so this column assumes balance valves in direct-return, but none in reverse-return piping.

In a building that is likely to be remodeled or reconfigured several times over the life of the piping system, such as a speculative office building with tenant turnover, elimination of balance devices can be a significant advantage. In this case, if the primary building HVAC system is conventional VAV with many reheat boxes,

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FIGURE 5 Two-story office building using a direct-return piping arrangement per floor.

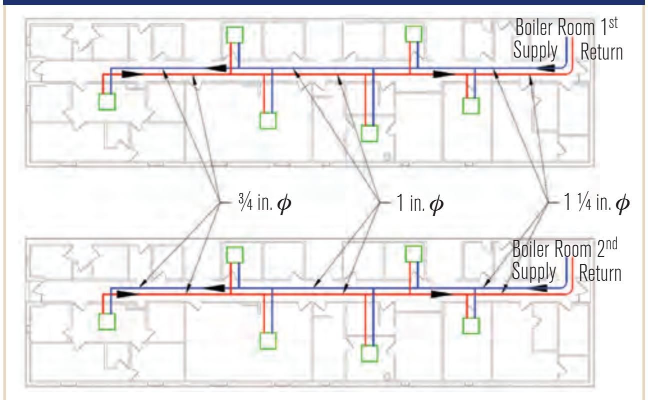


FIGURE 6 Two-story office building using a single reverse-return piping network.

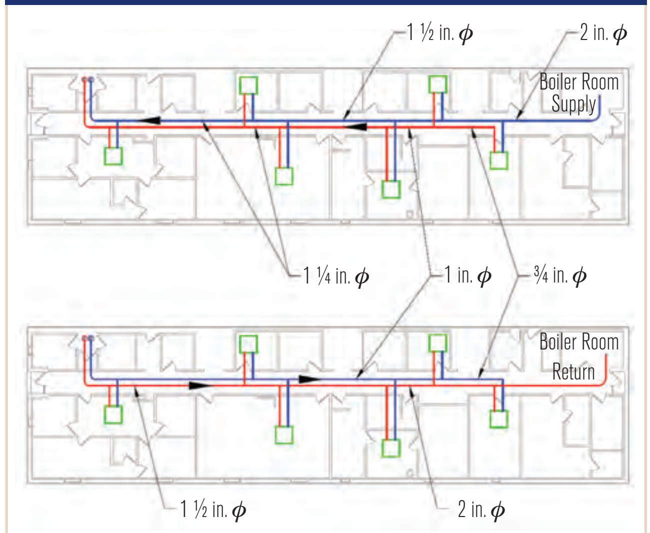
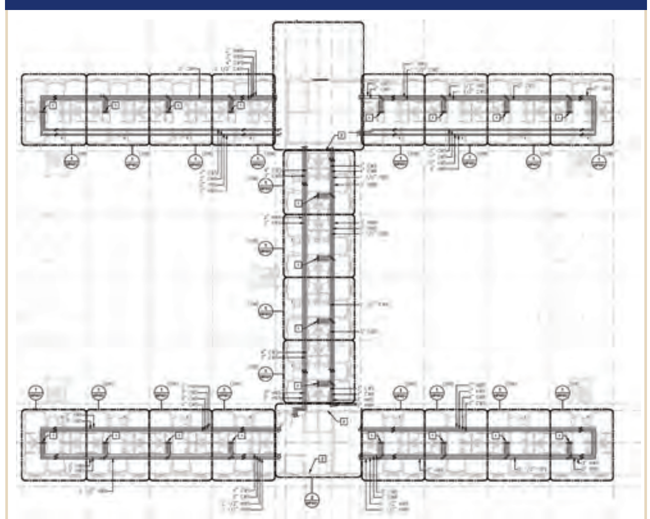


FIGURE 7 Three-story H-shaped building poorly suited for reverse-return.



adding, subtracting, and relocating VAV reheat boxes in a reverse-return system does not require manual rebalance of the hydronic piping.

Building owners are reluctant to send technicians climbing above ceilings in occupied portions of the building not otherwise impacted by renovations, so a balanced direct-return system may not stay that way; whereas a nearly self-balanced reverse-return system will remain nearly so after even several renovations.

For a two-pipe water-source heat pump system, with both heating and cooling inputs to the loop, a reverse-return system allows the heating and cooling source to be widely separated. This can be useful with cooling towers on the roof and boilers in the basement, in which case direct-return doesn't work. Both the heating and cooling sources need to be upstream of the supply loop in direct-return. The heating and cooling sources can be separated at two extremes of the distribution loop if they are connected by the reverse-return leg.

Finally, there is a misconception that reverse-return adds total system head pressure because of the long return line, but a closer examination will show that to be untrue. Using *Figure 1* as an example, in the direct-return system, the pump must push the water to the farthest terminal and all the way back to the source again. Each of the other branches to the other terminals must include some artificial means of increasing the pressure drop to simulate the losses of the longest run.

In the reverse-return system, the longest run (or, any run through any of the terminals) is no longer than the longest run in the direct-return system. In fact, reverse-return usually results in somewhat lower pump head than direct-return. This is because (a) the reverse-return main is a large pipe 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; and (b) because even a fully open balance device has some pressure drop, which can be eliminated with reverse-return.

Case 1: Reverse-Return is Advantageous

If, for example, the piping system is designed to make a complete loop around the inside perimeter of one floor of a building, starting and ending at a mechanical room or similar base point, the difference between direct and reverse-return piping quantities is zero to negligibly small. To illustrate, see *Figure 2* and *Figure 3* (Page 60). It is envisioned in these figures that the piping is a hot water heating system and the terminals depicted are VAV

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reheat boxes, but they could easily be finned-tube convectors, or fan-coil units, or even chilled beams in a chilled water system.

It is also envisioned that the center core is "impenetrable" by piping (e.g., an open atrium, or courtyard, or banks of elevators and stairwells, etc.), forcing the hydronic piping to route around the perimeter.

Compare *Figure 3* with *Figure 2* and notice that overall system pressure drop (and pump head) is reduced with reverse-return in this case. The worst hydraulic path—the path that determines overall system pressure drop and pump head—in *Figure 2* (assuming all coils and branches are the same) is the last one in the string. Supply water flows about three-fourths of the circumference of building to the coil, then returns three-fourths of the circumference of the building back again, for a total of 1.5 times the building circumference. In *Figure 3*, all possible hydraulic path pipe lengths equal about one circumference of the building. So reverse-return will reduce total friction, reduce pumping energy, and eliminate balance valves, all while not increasing piping cost.

Case 2: It's a Toss-up—Further Study Needed

This author has actually seen *Figure 2* applied many times when clearly *Figure 3* would have been advantageous. But there is a third option, as the astute reader may have already anticipated—splitting the full circumference of the building into two halves (*Figure 4*, Page 60) and using two separate direct-return circuits. Now the reverse-return of *Figure 3* is not quite as favorable because the mains start and end larger than those of *Figure 4*. The reverse-return of *Figure 3* still handles load diversity better than *Figure 4* and still offers the advantage of eliminating balance valves and balance labor, both present and future.

In another example, a long, narrow two-story building with a mechanical room at one end can use reverse-return by traveling outbound on the first floor and inbound on the second floor with only a small difference in piping cost (*Figures 5* and *6*, Page 62). In smaller piping sizes, the cost of labor to install the piping, insulation, hangers, and so forth tends to dominate and lessen the differences in pipe material cost by size, so the net cost after deducting balancing may be similar. Both of these examples—*Figure 3* versus *Figure 4* and *Figure 5* versus *Figure 6*—merit a closer look and neither should be dismissed out of hand.

Case 3: Direct-Return is Advantageous

Reverse-return piping is generally not a good choice when the distribution system dead-ends, such as in an elongated narrow (or "long-and-skinny") single-story system. In that case, the reverse-return piping requires a third pipe to carry the full system flow from the far end back to the mechanical room or starting point. While this arrangement still provides the self-balancing benefit, the addition of a third pipe running the entire length of the system may add significant cost. On the other hand, if that building is two stories, reverse-return may still be applied cost-effectively, as discussed in Case 2.

In some cases, reverse-return results in a significant increase in piping cost and would then be difficult to justify. An example of a hopelessly misapplied reverse-return system in this author's recent project experience is that of the H-shaped three-story building (*Figure 7*). The request for proposal (RFP) on that project required a four-pipe fan-coil system room-by-room, and it insisted on reverse-return. Making matters worse, the RFP required balance valves at every fan-coil unit even though reverse-return was specified, eliminating the one potential saving grace of reverse-return.

The routing of the reverse-return lines was particularly convoluted due to the H-shape and the three stories, rendering the four-pipe system to become essentially a "six-pipe" system: (1) outbound hot water supply, (2) outbound hot water return, (3) inbound hot water return, (4) outbound chilled water supply, (5) outbound chilled water return, and (6) inbound chilled water return in each "arm" of the H-shaped building. The RFP's author could not be swayed otherwise, but clearly this was a poor application for reverse-return.

Case 4: A Hybrid is Advantageous

Not to be overlooked is the case of a multistory building. Revisit *Figures 2* and *3* and now imagine they represent a 10-story building consisting of a similar floor plan on each level. Instead of the boiler room found in *Figures 2* and *3*, imagine that location represents a vertical pipe shaft with a boiler either in the basement below or on the roof above. In this case, the most advantageous design may be a hybrid, with reverse-return on each individual floor as depicted in *Figure 3*, but with direct-return on the main vertical riser.

Because the piping makes a complete loop around the inside perimeter, almost no difference exists between direct- and reverse-return piping quantities on the floor. But reverse-return on the main vertical riser would require a large additional return pipe to take flow from the 10th floor to the basement, so that portion of the system remains direct-return.

This author has used this hybrid design many times, using a single balance valve in only one location per floor (near the shaft) to balance the flow to that floor, while allowing the self-balancing nature of reverse-return to eliminate all the balance valves at individual coils. If each floor has, say, 40 hydronic coils, this 10-story hybrid system reduces the number of balance valves from 400 to 10,* while still retaining most of the advantage mentioned earlier in terms of future renovations.

Conclusions

Reverse-return piping is a beneficial design option in some cases. Don't automatically assume it adds first cost; it can be cost-neutral in many applications and less expensive than some balancing options such as pressure-independent control valves. And don't fall into the trap of thinking it adds system pressure drop and/or increases pump head; pump head will be the same or lower. Engineers and designers laying out hydronic systems should be familiar with reverse-return and consider applying it when appropriate, as well as understand its limitations and recognize applications where it is not beneficial.

Acknowledgments

The author thanks Cole Stirewalt, Student Member ASHRAE and an undergraduate in mechanical engineering at the University of Missouri. He prepared *Figures 2* through *6* as part of a summer internship with the firm employing this author.

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2. Taylor, S., J. Stein. 2002. "Balancing variable flow hydronic systems." *ASHRAE Journal* 44(10).
3. *2016 ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 13, Figure 21. ■

* On more than one occasion, contractors eager for a change order pointed out to me that I "forgot" all those balance valves at the coils. They were disappointed to learn that I did so on purpose and there would be no change order.

"I saw an article by Mick Schwedler, P.E., Fellow ASHRAE, about waterside economizers in data centers, and the idea came to my mind: 'I wonder if the performance required by ASHRAE Standard 90.1 can actually be achieved?'. So, I set up the whole project, load calculations in various cities, coil selections, cooling tower selections, and discovered that, very likely, most waterside economizers do not meet Standard 90.1 performance requirements."
~Daniel H. Nall, P.E. Originally published in August 2014


Daniel H. Nall

Waterside Economizers & 90.1

BY DANIEL H. NALL, P.E., BEMP, HBDP, FAIA, LIFE MEMBER ASHRAE

Most engineers probably feel they are safely code compliant with their waterside economizers if their system is configured per Steve Taylor's June column:[1] as an integrated economizer with the heat exchanger in series with, and upstream of, the chillers, if their cooling tower is intelligently selected for their peak load condition, and if they provide the appropriate controls to initiate economizer mode during the appropriate exterior conditions. But, they would likely be wrong. It's not only about the cooling tower.

A number of factors determine whether ASHRAE/IES Standard 90.1-2013's requirements can be met,* some of which are completely out of the engineer's control. The fraction of room sensible and total design load experienced by the airside and the waterside systems of the building during those conditions is one of the most important variables that determines whether or not the waterside economizer can meet the entire cooling load at the 50°F dry bulb/45°F wet bulb (10°C/7.2°C) condition. Even in a new, fully code compliant building, these fractions can vary dramatically.

Supply Air Temperature Reset

Re-resetting the supply air temperature upwards during the 50°F/45°F (10°C/7.2°C) condition is almost a prerequisite for achieving waterside economizer code compliance for a variable air volume system. With an air temperature setpoint off the coil of 53°F (11.7°C), necessary to control space humidity for design weather conditions in many locations, there is an inadequate system approach temperature, given that the cooling tower, heat exchanger and cooling coil each has approach limitations.

Resetting the supply air temperature requires that it be able to meet the sensible load in all zones, including interior zones not affected by the exterior temperature. Therefore, terminals in all interior zones must be upsized so loads can be met with the higher supply air temperature, as required by Standard 90.1-2013: "Zones that are expected to experience relatively constant loads, such as electronic equipment rooms, shall be designed for the fully reset supply temperature."[2]

TABLE 1 Space sensible, airflow and total coil part load fractions and coil leaving air setpoints for various cities at 50°F dry bulb/45°F wet bulb (10°F/7.2°F).*

CITY	ZONE	SPACE SENSIBLE PART LOAD ÷ DESIGN LOAD	AIR OFF COIL SETPOINT (°F)	AIRFLOW PART LOAD ÷ DESIGN LOAD	COIL LOAD PART LOAD ÷ DESIGN LOAD
Atlanta	3A	0.72	58.0	0.96	0.42
Los Angeles	3C	0.79	57.1	0.99	0.56
Bakersfield	3B	0.67	58.0	0.89	0.40
New York City	4A	0.96	53.0	53.0	0.64
Seattle	4C	1.03	53.0	1.03	0.79
Chicago	5A	0.98	53.0	0.98	0.65
Denver	5B	0.97	53.0	0.97	0.78

*Design loads calculated with ASHRAE outdoor air design conditions and August solar geometry. Part loads calculated at 50°F dry bulb/45°F wet bulb (10°F/7.2°F) with October solar geometry.

* Standard 90.1-2013 states, "Water economizer systems shall be capable of cooling supply air by indirect evaporation and providing up to 100% of the expected system cooling load at outdoor air temperatures of 50°F dry bulb/45°F wet bulb and below."

TABLE 2 Cooling coil selections and cooling tower supply temperature required to meet waterside economizer loads at 50°F/45°F (10°C/7.2°C) condition.

	COOLING COIL*							
	DESIGN FACE VELOCITY (FPM)	ROWS	FPI	50/45 FACE VELOCITY (FPM)	DBT OFF COIL (°F)	REQ. CHWT (°F)	HX APPROACH (°F)	CT SUPPLY TEMP. (°F)
Atlanta	546	6	14	523	58.0	55.1	3.0	52.1
	400	8	10	383	58.0	56.1	3.0	53.1
Los Angeles	546	6	14	541	57.0	53.3	3.0	50.3
	400	8	10	397	58.0	54.3	3.0	51.3
Bakersfield	546	6	13	487	58.0	55.1	3.0	52.1
	400	8	10	356	58.0	56.3	3.0	53.3
New York City	546	6	14	525	53.0	48.6	2.0	46.6
	400	8	10	385	53.0	50.1	2.0	48.1
Seattle	546	6	12	563	53.0	45.8	2.0	43.8
	400	8	10	412	53.0	48.5	2.0	46.5
Chicago	546	6	14	536	53.0	48.5	2.0	46.5
	400	8	10	392	53.0	50.0	2.0	48.0
Denver	546	5	14	528	53.0	45.5	2.0	43.5
	400	8	10	387	53.0	49.4	2.0	47.4

*All coil and cooling tower selections made using standard air (not elevation corrected).

Of course, systems and components are usually sized with a safety factor, but it only takes one fully loaded interior zone to prevent successful supply air temperature reset. Upsizing terminals must be done carefully because the minimum supply air fractions on today's pressure independent terminals may cause overcooling or reheat activation in underloaded interior zones.

Having dealt with the troublesome interior zones, the engineer seeking compliance for his/her waterside economizer has to deal with another factor, the coincident internal sensible load fraction for the entire building. While internal loads do not vary with external temperature, and solar loads will vary with the changing relationship of the building geometry to seasonal solar geometry, conduction loads across the building skin will vary with the temperature difference between inside and outside.

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In Standard 90.1-2013, however, envelope performance requirements change significantly from Climate Zone 3 to Climate Zone 4, markedly changing the perimeter zone load dependence on conduction gains and losses and solar heat gain. *Table 1* (Page 66) shows the relationship between design conditions and the 50°F/45°F (10°C/7.2°C) economizer condition for room sensible loads, supply air reset opportunities and total coil loads for several cities in Climate Zones 3, 4 and 5.

Bakersfield, Calif., was chosen as an example of a moderately hot and dry climate. These loads are calculated simplistically using an instantaneous calculation of a simplified, minimally code compliant rectangular single floor in a high-rise building. While Standard 90.1-2013 does not specify solar conditions coincident with the 50°F dry bulb/45°F wet bulb (10°C/7.2°C) outdoor air conditions, this study assumed clear sky conditions and August solar heat gain coefficient (SHGC) for the design condition and October SHGCs for the 50°F/45°F (10°C/7.2°C) condition. Addition of mass walls, solar shading devices, roofs and architectural complexity would certainly change the results somewhat, but this calculation identifies the importance of the issue.

While resizing a few terminals and duct connections to enable servicing interior zones with higher temperature supply air is a small cost, resizing the entire airside system to enable such reset will not be feasible for most projects. Projects in Climate Zones 4 and above may not be able to reset supply air temperature to facilitate waterside economizer operation at the required outdoor conditions.

Cooling Coil and Heat Exchanger Performance

The cooling coil supply air approach to entering cooling water temperature is an important component of the total approach of supply air temperature to outdoor wet-bulb temperature. Coil selection balances a number of factors, including construction economy, energy efficiency and space requirements. Reducing coil area decreases costs, but increases pressure drop across the coil, increasing energy cost and also reducing thermal coupling between the coil and the airstream. The latter characteristic will have an influence on the coil approach temperature at economizer conditions. *Table 2* (Page 67) shows coil selections for a number of cities and

TABLE 3 Range and cooling tower supply temperature at 45°F (7.2°C) wet bulb for various cooling tower selections. (Based on 50°F/45°F [10°C/7.2°C] part load fractions in different cities.)

City Design WB Zone	Design Range and Approach Criteria	10°F* 7°F	14°F 7°F	10°F* 4.5°F	14°F 4.5°F
Atlanta 78°F 3A	Range (°F)	5.08	5.08	5.08	5.08
	Supply Temp (°F)	50.5	52.2	48.8	49.9
Los Angeles 70°F 3C	Range (°F)	6.76	6.76	6.76	6.76
	Supply Temp (°F)	51.2	52.1	49.0	49.8
Bakersfield 74°F 3B	Range (°F)	4.88	4.88	4.88	4.88
	Supply Temp (°F)	51.2	52.1	49.0	49.8
New York City 78°F 4A	Range (°F)	7.83	7.83	7.83	7.83
	Supply Temp (°F)	53.0	55.5	50.6	52.3
Seattle 66°F 4C	Range (°F)	9.65	9.65	9.65	9.65
	Supply Temp (°F)	51.7	53.7	49.3	50.7
Chicago 78°F 5A	Range (°F)	7.92	7.92	7.92	7.92
	Supply Temp (°F)	53.1	55.6	50.7	52.4
Denver 65°F 5B	Range (°F)	9.52	9.52	9.52	9.52
	Supply Temp (°F)	51.7	53.6	49.2	50.7

*Water flow for 10°F (5.6°C) cooling tower selections reduced to 71% at 50°F/45°F (10°C/7.2°C) condition to optimize heat exchanger flow.

the entering water temperature required to meet the supply air temperature setpoint established in *Table 1*.

Coils typically are sized at a face velocity of approximately 550 fpm (6.25 m/s). Coils selected by these criteria to meet the loads are standard selections, with 5 or 6 rows, and 12 to 14 fins per inch (fpi). These coils were selected with a 9°F (5.0°C) approach. Note that the approach temperature at economizer conditions is significantly lower, primarily because of the lower enthalpy of the incoming mixed air.

Selecting a coil with much more heat transfer surface, 8 rows, 10 fins, and at a much lower face velocity, 400 fpm (4.5 m/s), results in a slight reduction in static pressure drop across the coil, but also a significant decrease in coil approach temperature at economizer conditions. Heat transfer area density, however, is constrained by ASHRAE Standard 62.1, which limits dry coil

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pressure drop at a face velocity of 500 fpm (2.54 m/s) to 0.75 in. w.c. (187 Pa) as a surrogate measure to ensure cleanability.[3]

Considering just design conditions, selection of a more robust coil might offer the opportunity for reduction in the chilled water flow rate, due to increased chilled water temperature rise through the coil. The designer should verify that the reduced maximum chilled water flow through the coil will provide a sufficiently low approach temperature so that the required supply air temperature goal can be met during economizer conditions. For many buildings, meeting the energy code economizer requirement will require a more robust coil selection than is required to meet standard design conditions.

Plate and frame heat exchanger approach is another variable in the total approach from outdoor air wet-bulb temperature to supply air temperature. According to Taylor's recent column, the most cost-effective selections for plate and frame heat exchangers is a 3°F (1.7°C).[1] As will be shown later, this selection may be possible in some climates, and may represent a cost trade-off with the cooling tower selection. In other climates, where further reduction in cooling tower approach is space or cost prohibitive, a 2°F (1.1°C) selection may be required.

Cooling Tower Performance

While total interior sensible load has a great impact on the operation of the airside system, total coil load fraction has a great impact on cooling tower approach. As has been reported in several recent *ASHRAE Journal* articles, cooling tower approach at low wet-bulb temperatures increases significantly above that at design conditions, depending on the heat rejection loads on the tower.[4,5] The total coil load is a function of both the interior sensible load and the latent and sensible load of the required outdoor air fraction.

At the 50°F/45°F (10°C/7.2°C) economizer setpoint, the coil is likely dry, because the available water from the cooling tower is warmer than the mixed air dewpoint temperature. The reduction of coil total load for 50°F/45°F (10°C/7.2°C) conditions compared with design conditions is, therefore, a function of the reduction in interior sensible load caused by envelope heat transfer at the lower outdoor air temperature and by the reduced enthalpy of the mixed air onto the cooling coil.

Cooling tower selection at design conditions for range and approach temperature will have a significant impact on the temperature of the water off the cooling tower at the 50°F/45°F (10°C/7.2°C) condition. *Table 3* (Page 68) shows the performance of some selections that were made with Cooling Technology Institute compliant software at the 50°F/45°F (10°C/7.2°C) condition.

Cooling towers were selected at the approximate design wet-bulb temperature for the location, and, arbitrarily, at 4.5°F and 7°F (2.5°C and 3.9°C) approaches and 10°F and 14°F (5.6°C and 7.8°C) ranges. No attempt was made to optimize selections for life-cycle costs or minimum energy costs.

For this study, water flow for the towers selected at a 10°F (5.6°C) range (2.8 gpm/ton [3.01 L/s·kW]) was reduced to 2.0 gpm/ton (2.15 L/s·kW) for economizer operation for the cooling tower water flow to be more equal to chilled water flow through the heat exchanger for economizer operation. Experimenting with the software indicated that reduction in water flow with increased range, maintaining the same load, has little impact on cooling tower approach until the chilled water temperature range drops below about 5ºF or so.

From the standpoint of systems design, the impact of selecting the cooling tower for a reduced approach temperature is much less than resizing the entire condenser water system to accommodate a reduced design range. Reducing the design approach temperature, furthermore, is more effective at reducing the cooling tower approach temperature at economizer conditions than is reducing the design range.

Comparison of the required cooling tower supply temperature shown in *Table 2* with the cooling tower supply temperature from several cooling tower selections in *Table 3* gives rise to several conclusions:

1. In Climate Zone 3, the design cooling coil often is adequate to meet the waterside economizer requirement.

2. In Climate Zone 3, trade-offs between cooling coil robustness and cooling tower size can provide an optimized solution.

3. In Climate Zones 4 and 5, no combination of cooling coil and cooling tower meets the waterside economizer requirement.

Non-compliance in Climate Zones 4 and 5 is primarily the result of the unchanged sensible indoor load at the economizer condition, lower envelope conduction losses due to lower U-value requirements, and increased solar loads due to higher SHGC limits (compared with Climate Zone 3), not only preventing supply air temperature reset, but also raising the cooling tower load and approach at economizer conditions.

Conclusions

Achieving compliance with Standard 90.1-2013 with respect to waterside economizers is not the sure thing that many have assumed it is. Achieving and documenting compliance requires a process that provides information for the selection of a number of components that impact waterside economizer performance.

Clearly, the first step is to establish the room sensible part-load fraction at waterside economizer conditions. Reduced window solar control requirements and increased envelope thermal performance requirements in colder climate zones will tend to increase that part-load fraction. While the simple building load calculation used for this analysis demonstrated very high part-load fractions for cities in Climate Zones 4 and 5, this outcome may not be the case for all projects and the calculation should be made for every project.

After room sensible part-load fraction has been established, along with the potential for supply air temperature reset, the total coil part-load fraction can be calculated. Note that locations with relatively mild design conditions, such as Los Angeles and Seattle, will be challenged because their part-load fraction will tend to be high. Decreasing the part-load fraction helps achieve economizer compliance in two ways; a lower space sensible load fraction enables supply air temperature reset to increase the difference between the ambient wet-bulb temperature and the supply air temperature, and a lower total coil load fraction enables a closer approach of the leaving cooling tower water temperature to the ambient wet-bulb temperature.

Selection of the cooling coil comes next, and this selection can be done in concert with the heat exchanger selection and the cooling tower selection. The strategy for the cooling coil is to increase the thermal coupling between the coil and the airstream, minimizing the approach of the leaving supply air temperature to the entering cooling water temperature. Using 8 row, 10 fpi coils to minimize supply air temperature approach to the entering chilled water temperature is a powerful strategy.[1] For the heat exchanger, the strategy is merely to determine what is the largest approach temperature that your other components will let you get away with. For the cooling tower, the strategy is to define the most cost effective way of achieving the required cooling water temperature to the cooling coil. Cost effectiveness analyses for various options for each of these components can lead the designer to the optimal means of achieving compliance.

The Standard 90.1-2013 outdoor condition requirement for meeting the entire cooling load with waterside economizer is arbitrary. It does not vary with climate zones for buildings (for data centers it does vary since data center loads are relatively independent of wet bulb), and, thus, is not related to the frequency distribution of wet-bulb temperatures in any location. However, an effective economizer function is a powerful tool in reducing energy consumption and cost. The above discussion relates to how to obtain code-mandated economizer performance, but could be used to evaluate economizer function at other conditions. Conditions for initiation of economizer function could be changed to reflect the actual wet-bulb frequency distribution in any climate in pursuit of the optimal solution in that climate. The issues and the techniques would be the same no matter the target conditions.

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"There are many simple things engineers and owners sometimes overlook to improve performance of central chilled water distribution systems. I have spent much of my career troubleshooting and correcting design and operational issues to optimize performance in these systems. I focus on how to improve chilled water ΔT, select effective pumping strategies, and how to connect buildings to these large distribution systems to avoid common problems." ~Kent W. Peterson, P.E. Originally published in January 2014

Kent W. Peterson

Improving Performance Of Large Chilled Water Plants

BY KENT W. PETERSON, P.E., PRESIDENTIAL MEMBER/FELLOW ASHRAE

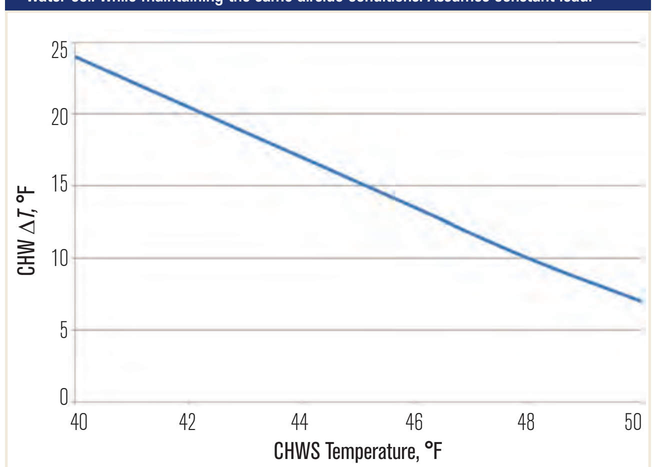
Although large campus central chilled water plants can be designed to be energy efficient, the most impact on the overall system performance often is how the connected building systems are designed to interface with the control plant.

Improving Chilled Water ΔT

Many large central chilled water systems depend on high chilled water temperature differential, ΔT, to minimize pumping energy and optimize chilled water thermal storage capacity. Buildings directly connected to central chilled water distribution systems should be designed to minimize pumping energy and maximize return chilled water return temperature to the central plant. High ΔT is achieved with proper coil and control valve selection, piping and pumping design and supply water control.

Maximizing the ΔT between the chilled water supply and return will maximize the cooling load that can be met with a given chilled water flow rate. Chilled water ΔT is primarily determined by cooling coil effectiveness at the loads and is not something that can be achieved with controls or control sequences at the central chiller plant.

FIGURE 1 Impact on LWT from varying EWT for an original 45°F EWT/55°F LWT chilled water coil while maintaining the same airside conditions. Assumes constant load.



Cooling Coils

Maximizing cooling coil performance is crucial for the entire chilled water system operation. Chilled water ΔT will be determined by how well the terminal devices perform. Cooling coils should be selected to satisfy the load, considering the expected supply water temperature delivered to the coil.[1] Temperature gain in the distribution system as well as heat exchangers should be considered. Chilled water temperature at the cooling coil inlet can sometimes be several degrees higher than the supply temperature leaving the central plant. The return water temperature and leaving air condition at each coil depends on coil configuration, airflow across the coil, entering air enthalpy and entering water temperature.

When designing new buildings to connect to an existing central plant, it is many times best to use an 8 row/10 fins per inch coil.[2]

When evaluating the potential for connecting an existing building to a high ΔT central chilled water system, careful evaluation of the existing coils is prudent when considering a potential lower chilled water supply temperature and the coils ability to meet the system ΔT requirements. Generic AHRI-certified rating and selection programs (available from several

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coil manufacturers) can be used to model existing coil conditions/ construction. The coil construction can be matched and the impacts of the different chilled water supply temperature can then be modeled for the existing coils. Many times the existing coils may not need to be changed out when the higher ΔT central plant has a lower supply water temperature than the original coils within the existing building being connected to the plant.

Coil performance is generally based on mean temperature differential of the supply and return water temperatures at the coil. For a given coil selection and load, the warmer the supply water, the more water the coil needs to meet the load, resulting in a lower return water temperature. *Figure 1* shows the effect on ΔT for a cooling coil at different entering water conditions with a constant load. ΔT will also degrade as the entering air temperature approaches the design return water temperature during part load conditions.

Coil Control Valves

In maximizing ΔT two-way control valves are necessary for all cooling coils; do not use three-way valves at any coil. Pressure drop at the rated flow rate can vary, depending on where the cooling coil is located in the system hydraulic gradient curve. Several easy-to-use hydraulic modeling programs are available for modeling hydraulic performance and optimizing pipe and valve sizing within the building.

To accomplish high ΔT through a range of load conditions, the coil control valves and actuators must[3]:

- Be selected and sized in the hydraulic gradient so that the valve uses its full stroke;
- Have a high rangeability for controlled operation at very low flow;
- Be able to shut off against the highest anticipated differential pressure;
- Close when the air handler or terminal unit is off; and
- Be controlled to maintain leaving air design conditions.

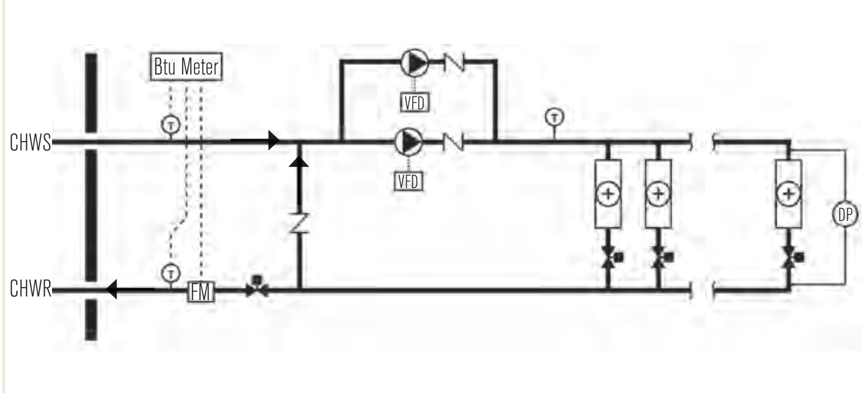
High ΔT can be accomplished with the proper design, construction and operation of coils, control valves and control sequences. There are other operational causes for low ΔT e.g., low entering air temperature during airside economizer, which can never be eliminated. Therefore, the chilled water plant design and chiller selections should account for the range of chilled water ΔT anticipated throughout the year.

Chilled Water Building Connections

It is critical to understand the central plant distribution system pumping control scheme prior to designing the building connection. This includes understanding the potential range of differential pressure the building connection would encounter throughout the year.

The most common building connection in campus-type chilled water systems is a direct connection since the same entity owns the central plant, distribution and buildings. Direct connections are suitable in a system with low-rise buildings where the static head in the distribution system can be kept low. District cooling systems tend to use an indirect connection with heat exchangers to isolate the customer's chilled water system from the

FIGURE 2 Decoupled direct building connection.



district chilled water system. This column focuses on direct connection options.

Direct connections are either decoupled or non-decoupled. A decoupled building connection as shown in *Figure 2* is typically configured with a crossover bridge, building pump and building return water temperature control valve.

The crossover bridge allows the building return water to blend with the supply water and is intended to hydraulically decouple the building from the chilled water distribution system. This was common practice prior to variable frequency drives and networked control systems.

Decoupled connections also have been used when connecting a building that was designed for a higher chilled water supply temperature than the central chilled water system. This technique was used to blend up the chilled water supply temperature to the building to provide the original higher cooling coil design temperature. To optimize ΔT and maintain good temperature and humidity control, it is best to use the colder supply temperature at the coils. Existing buildings should changeout

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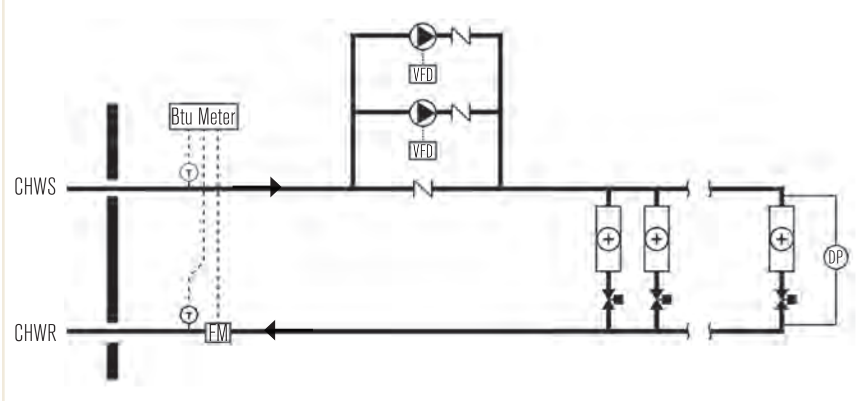
the control valves for the lower flow required to meet the cooling requirements. Blending to raise supply temperature will always increase pump energy and (perhaps counter-intuitively) decreases return water temperature and thus decreases overall plant Δ*T* as shown in *Figure 1.*

FIGURE 3 Non-decoupled direct building connection.



In my experience these practices typically lead to problems relative to building pump and fan energy, control instability, capacity and comfort as well as possible loss in building latent cooling capacity. Therefore, decoupled connections are unnecessary and should be avoided.

Three-way control valves and bypasses should be eliminated in buildings connected to large chilled water systems if trying to maintain high Δ*T*. In rare circumstances when chilled water supply temperature degradation due to long residency in the chilled water system is a concern, a small bypass at the end of the distribution system may be provided and controlled to maintain cold supply water temperature.

Where sufficient distribution differential pressure is available, a non-decoupled direct connection may require no more than a supply and return pipe with appropriate energy metering. When the existing differential pressure

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at the building connection is excessive for the control valve requirements, a differential pressure control valve could be used at the building connection to reduce the differential pressure to that required for the building coil control valves to operate properly. This should be avoided when designing the complete distribution system by choosing a pumping strategy that minimizes the differential pressure gradient in the chilled water distribution system.

The non-decoupled approach can be modified where differential pressure at the building connection will not be adequate during all of the year by adding a building pump that is installed with a parallel bypass and check valve so the pump is only operated when it is required to increase differential pressure for the building as shown in *Figure 3*. Parallel pumps can also be used to provide redundancy. The bypass check valve pressure loss should be less than 1 psi (7 kPa) to prevent water flowing through the building pump when it is off. Many times, the building pump only will be required to operate a fraction of the year when higher cooling loads are experienced, resulting in higher required building loop pressure drop.

Control of the building pump can be provided with a differential pressure transducer in the building chilled water loop. The differential pressure setpoint should be reset based on most demanding coil chilled water valve position. The building pump only should be used when the central plant chilled water distribution system cannot meet the required differential pressure setpoint.

The non-decoupled approach maximizes ΔT while providing stable temperature and humidity control.

Concluding Remarks

The methods in which buildings, air handlers and terminal devices are designed to work with chilled water systems frequently have the greatest impact in optimizing chiller water system ΔT. Hopefully, these tips can help designers and chilled water plant operators improve chilled water system performance.

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