Engineers Newsletter

Chilled-Water Systems Design Issues

Learning from past mistakes

As with anything in life, you can learn HVAC design the hard way, from painful and costly mistakes, or the easy way, from veteran engineers who can help you avoid those mistakes. The purpose of this EN is to help new engineers with the latter.

Background

The following three cases are composite scenarios of various chilled-water system design issues we have witnessed over the years. In each scenario, the design issues are highlighted and followed by possible solutions.

By sharing the hard-earned HVAC design wisdom that comes from trial and error, we hope to shorten the learning curve for new engineers.

Case 1: New Variable Primary Flow

A new system has been designed using an increased system chilled-water temperature difference as recommended in the ASHRAE GreenGuide[1] and by Taylor[2]. The chillers are selected at low design evaporator pressure drops (5 ft of head [15 kPa]) per chiller to minimize pump power and energy. To reduce installed cost and operating costs, the system is designed as variable primary flow (VPF). The bypass line (sized the same as the supply manifold) and valve that allow minimum chiller flow rate are located in the mechanical room, as are the air handlers. A differential pressure sensor across the evaporator is used to determine flow rate and to control the bypass valve.(Figure 1).

See any red flags? Let's take a look at the problems that occurred on the job and how to avoid making the same mistakes.
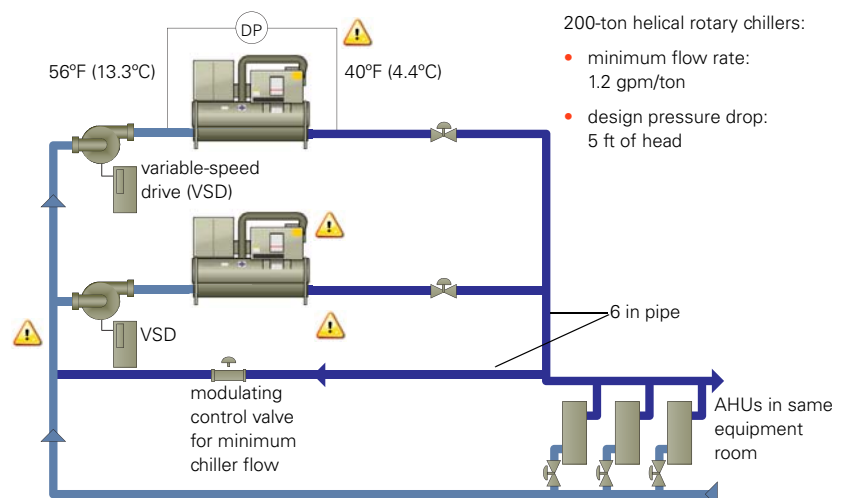
Issue 1: Flow sensing device is not accurate enough at the minimum flow rate. As mentioned, the design evaporator pressure drop is 5 ft of head [15 kPa]. At the minimum flow rate of 1.2 gpm per ton [0.022 L/s per kWr], the pressure drop will be about 3 ft of head [9.0 kPa]. Even if the differential pressure sensor is of high quality, it will be difficult to select one that has the required precision to accurately measure such a small difference between design flow and minimum flow pressure drops.

Solution 1: Select the proper flow sensing technology. Flow meters are generally more expensive, require more calibration, and must be installed per the manufacturer's requirements with a specific amount of straight piping up- and down-stream. Despite these limitations, they're better suited to monitor flow for this system.

Issue 2: Loop volume does not meet the chiller unit control requirement. Chiller unit controls often require a certain loop volume to maintain good chilled-water

Figure 1. Case 1: New variable primary flow (VPF)

200-ton helical rotary chillers:
- minimum flow rate: 1.2 gpm/ton
- design pressure drop: 5 ft of head

temperature control and ensure that compressors are not cycled unnecessarily. The loop volume includes the fluid in the evaporator bundle, pipes, and coils. In many close-coupled systems—for example where the air handlers and chillers are in the same equipment room—the loop volume will not be adequate. The required loop volume varies by chiller manufacturer, chiller type, and unit control capabilities. Check with the manufacturer for each chiller specified. In this case, the required loop time was two minutes. So the system water volume required is the loop time multiplied by the water flow:

2 min. x 300 gpm = 600 gal
[2 min. x 60 sec. per min. x 18.9 L/s = 2270 liters]

Solution 2: Increase the loop volume to the required level. If the loop volume is close to that required, increasing the piping manifolds may be adequate. Otherwise add a buffer tank, preferably on the return side of the system, and on the chiller side of the bypass line. Ensure the system volume with the tank meets the required loop volume. To attain smoother system control, consider making the tank a little larger than the absolute minimum requirement. The additional cost and space will likely be small.

Issue 3: Oversized bypass line and valve do not provide adequate control. The bypass line is the same size as the pipe manifold. Given this fact, if a large valve is installed in this line, control at the minimum flow rate may be a challenge, since initially opening the valve a little allows significant flow to be bypassed.

Solution 3: Size the bypass line and valve properly. In a variable primary flow system, size the bypass line for the largest minimum flow rate. In this case:

200 tons x 1.2 gpm per ton = 240 gpm [15 L/s]

Many designers will use 4 in [10 cm] line for this flow rate. Then select a valve to give proper bypass control. While not required, some consulting engineers specify a pressure independent valve for bypass.

Issue 4: Flow can't be significantly reduced. Chiller design flow rate is 1.5 gpm per ton [0.027 L/s per kWr] and the minimum is 1.2 gpm per ton [0.022 L/s per kWr]. This means the flow rate can only be reduced 20 percent before reaching minimum. That's not much for a system that is supposed to be *variable* primary flow.

Solution 4: Put the chillers in series!

With chillers in series (Figure 2), the design flow rate per chiller is about 3.0 gpm per ton [0.054 L/s per kWr], allowing significant turndown. Design evaporator pressure drop increases since water must flow through both evaporator bundles. This increases design pump power. However, in many systems the flow rate is less than design flow much of the year, so the resulting pump energy increase may be small.

To allow for servicing, engineers often design manual bypass lines into the system.

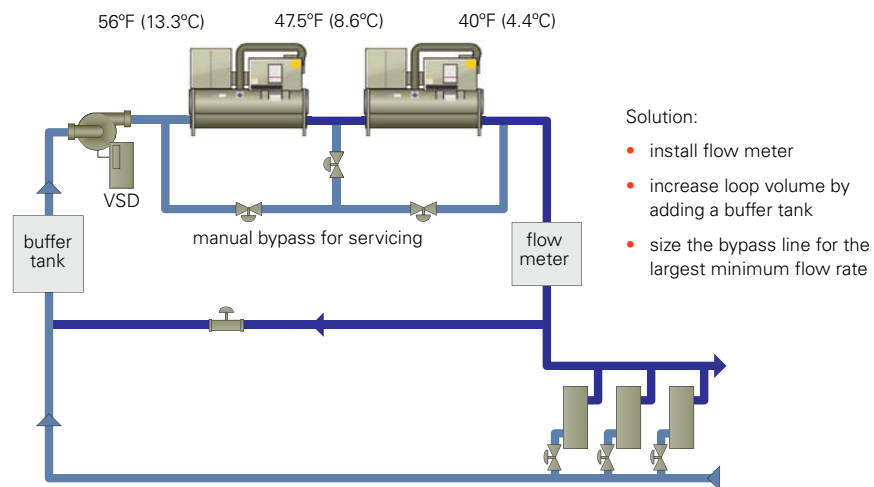The series chiller layout has two significant benefits:

• The upstream chiller operates at an elevated temperature and increased efficiency. Very often this compensates for any increase in pump energy.

• When chillers are piped in parallel in a VPF system, there is a significant flow rate change in the operating chiller when the second chiller is added. When the chillers are piped in series, there is *no* flow rate transition when the second chiller is enabled. This can greatly simplify system control.

It may be beneficial to pipe the chillers in series if the system

• is designed for variable primary flow,

• has a chilled-water design temperature difference of 14°F [7.8°C] or larger,

• has two chillers, and

• is not likely to require a future expansion.

If the system ΔT is 10°F [5.6**°**C], consider increasing it to 14°F [7.8°C] or larger.

Figure 2. New variable primary flow (VPF) series chillers

56°F (13.3°C)   47.5°F (8.6°C)   40°F (4.4°C)

VSD

buffer tank

manual bypass for servicing

flow meter

Solution:
• install flow meter
• increase loop volume by adding a buffer tank
• size the bypass line for the largest minimum flow rate

Case 2: Conversion from Primary Secondary to Variable Primary Flow

The system includes two 500-ton chillers in a primary-secondary configuration. The owner wants to convert the system to variable primary flow (VPF) and add a small (30-ton) heat recovery chiller.

This job is "value engineered" to:

- use the existing secondary pumps (which can handle the flow and pressure drop requirements),

- switch out the high quality differential pressure sensor, which costs $1500, to a $100 sensor, and

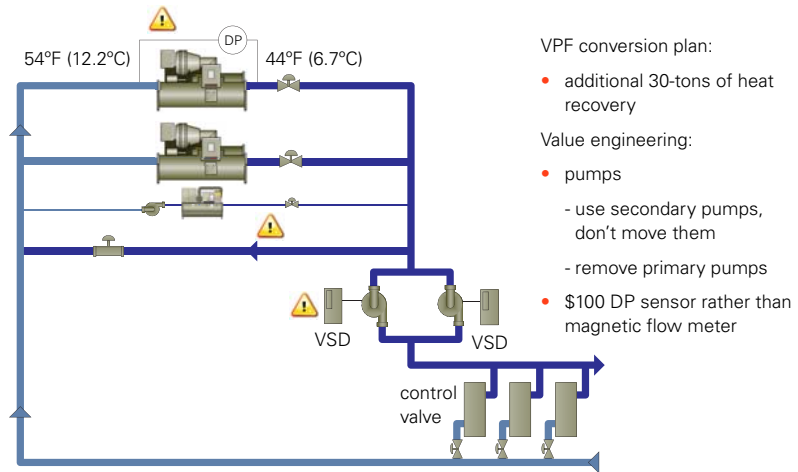- connect the heat recovery chiller in parallel with the existing chillers (Figure 3).

Issue 1: The pumps are positioned incorrectly to allow bypass for minimum flow. Water cannot flow from right-to-left in the bypass line, so minimum flow cannot be maintained.

Solution 1: Rather than converting to full variable primary flow, consider converting to variable primary/variable secondary. As its name implies, a variable primary/variable secondary system (Figure 4) employs variable primary (chiller) flow as well as variable secondary (coil) flow. In retrofitting a system, the use of the existing pumps already installed is generally simple. In contrast, converting to primary secondary requires significant piping changes, and the present secondary pumps must be moved and possibly increased in size.

To convert to variable primary/variable secondary:

- Contact the chiller provider to ensure that chiller unit controls can tolerate variable evaporator water flow. If not, update the unit controls.

- Add variable-speed drives (VSDs) to the primary pumps (make sure the pump motors are compatible).

- Add a high quality differential pressure sensor across each evaporator.

Figure 3. Case 2: After value-engineered conversion to VPF

54°F (12.2°C)   DP   44°F (6.7°C)

VSD   VSD

control valve

VPF conversion plan:

- additional 30-tons of heat recovery

Value engineering:

- pumps
  - use secondary pumps, don't move them
  - remove primary pumps
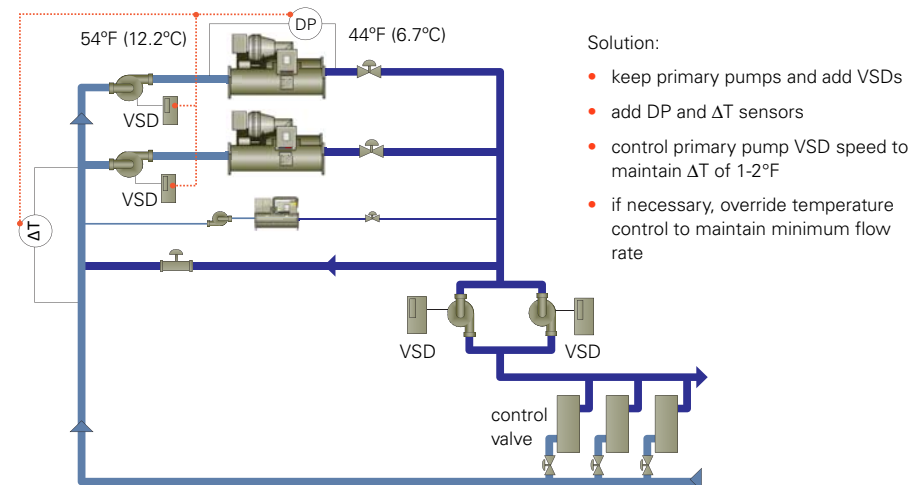- $100 DP sensor rather than magnetic flow meter

- Add a matched pair of temperature sensors to monitor the system return-water temperature and the chiller return-water temperature.

- Control the primary pump VSDs to maintain the chiller return-water temperature a degree or two lower than the system return-water temperature. This ensures there is always a little more chilled water being produced than demanded.

- If the temperature control results in chiller flow dropping below its minimum flow rate, increase the pump speed to maintain the required minimum chiller flow rate.

Issue 2: It is likely the flow-sensing device is not accurate enough. A $100 differential pressure sensor is very unlikely to provide the accuracy, reliability and repeatability required to operate a variable primary flow system. Unfortunately this seems to occur often. We suggest you resist the temptation to reduce costs when value will be lost. A VPF system operates on flow rate, so accuracy is critical.

Solution 2: Specify and install an accurate, reliable and repeatable flow-sensing device. It's very important that the flow-sensing device is accurate, reliable, and repeatable. Another benefit of a higher quality sensor is that, in general, it requires less calibration. The cost of a proper flow-sensing device will often be closer to $1000 than $100. Informative Appendix E of ASHRAE Guideline 22-2008[3] provides an example of flow measurement accuracy and precision specification language.

Figure 4. Convert to variable primary-variable secondary

54°F (12.2°C)   DP   44°F (6.7°C)

VSD

VSD

ΔT

VSD   VSD

control valve

Solution:

- keep primary pumps and add VSDs
- add DP and ΔT sensors
- control primary pump VSD speed to maintain ΔT of 1-2°F
- if necessary, override temperature control to maintain minimum flow rate

Issue 3: Poor heat recovery chiller location.   Flow and control issues can result when a smaller chiller is piped in parallel with larger chillers. When evaporators are piped in parallel, the chillers are loaded to equal percentages. This does not allow the heat recovery chiller to be preferentially loaded or to deliver the desired hot water temperature and quantity.

Solution 3: Pipe the heat recovery chiller in the sidestream position. The sidestream configuration (Figure 5) -allows the heat recovery chiller to:

- Be preferentially loaded, since it receives the warmest return water temperature.

- Operate as a "heater" to make the desired leaving condenser water temperature. The chilled water produced reduces the load on the central chillers.

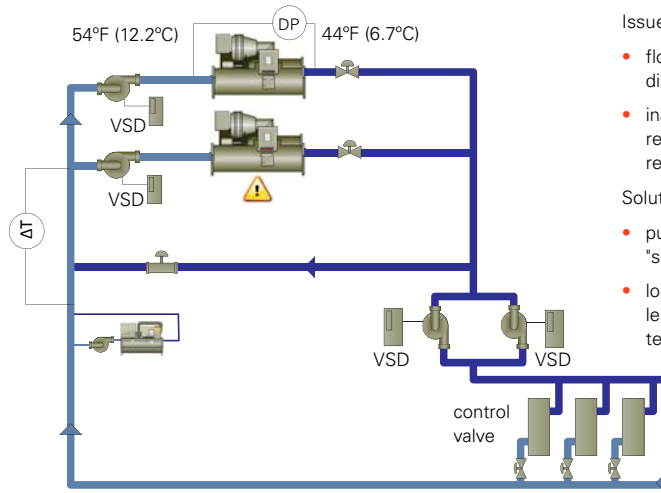- Meet its needed evaporator flow requirements independent of other chiller operation.

Case 3: Conversion of a Parallel-piped System to Series, and from "Conventional" to Increased ΔT.

This 400-ton system includes two water-cooled chillers piped in parallel. The chillers and air-handling coils were both selected with water temperatures of 54°–44°F. The system serves a school, which does not have a trained plant operator on site—simplicity is beneficial.

The school officials would prefer air-cooled chillers to eliminate cooling tower maintenance requirements. They have decided to use 50 percent glycol to keep the fluid from freezing in the winter. No changes are to be made to the airside of the system. In addition, energy usage reductions are desired.

For this scenario, let's examine some common misconceptions that might foil a new engineer.

Figure 5. Value-engineered conversion heat recovery

54°F (12.2°C)   DP   44°F (6.7°C)

VSD

VSD

ΔT

VSD   VSD

control valve

Issues:

- flow balance between dissimilar chillers

- inability to fully load the heat recovery chiller due to mixed return water temperature

Solution:

- put heat recovery chiller in "sidestream' position

- load using either condenser-leaving or chilled-water temperatures

Misconception 1: Existing coils cannot be reselected at higher temperature differences.

Solution 1: Coils are heat transfer media that can be selected at many conditions.   Table 1 shows a coil originally selected for a 10°F [5.6°C] chilled water ΔT reselected to *produce the same cooling capacity*. This can be done if the entering water temperature is *lower* and results in a *lower* flow rate and *higher* chilled-water ΔT. This change has no effect on the airside of the system, but a substantial impact on the waterside.

The advantages of series chillers (Figure 6) to produce the higher chilled-water ΔT was discussed earlier in Case 1.

When making supply water temperature and flow rate changes to existing systems, consider the following factors:

- Many coils in air-handling units have adequate heat transfer area to perform similarly to the data in Table 1. On the other hand, if a coil has limited heat transfer capability (e.g., small, two-row fan-coils), reselecting at a higher ΔT is unlikely to work well.

- What is the coldest (or lowest) chilled-water supply temperature possible before condensation forms within the pipe insulation? Often with existing pipe insulation, supply water temperature reduced to 40°F [4.4°C] can be successful, but perhaps not 36°F [2.2°C].

- If coil control valves were oversized at a 10°F [5.6°C] ΔT, they are even more oversized at a higher ΔT and lower flow rate. Valves may need to be changed to allow good control.

Misconception 2: Use of three-way valves as the method of bypass can use more pump energy.

Solution 2: Yes, but this is mitigated by a number of factors.

- First, the system flow rate dropped by nearly 38 percent because of the wider ΔT across the coils and reduced water flow rate. This lower flow rate results in lower system pressure drop and pump power.

- In many constant flow systems, it's difficult to retrofit a separate

Table 1. Air-handler coil reselection using low entering-water temperature

	original selection	reselection
capacity (MBh) [kW]	504 [148]	504 [148]
entering water temperature (°F) [°C]	44 [6.7]	41 [5.0]
flow rate (gpm) [L/s]	101 [6.37]	63 [4.0]
leaving water temperature (°F) [°C]	54 [12.2]	57 [13.9]
water ΔT (°F) [°C]	10 [5.5]	16 [8.0]

bypass, control valve and controller into the system. Leaving enough three-way valves to allow the required minimum chiller flow rate can be a simple, cost-effective way to convert a system from constant to variable flow.

- Minimum pump speed can be set to attain the minimum flow rate with all converted two-way valves closed.

Using three-way valves as a method of bypass is a good compromise between simplicity and performance.

Misconception 3: More glycol is better. When antifreeze is added to a system, fluid viscosity and pump power are often considered. However, these impacts may be small compared to the effect on chiller and coil heat transfer capability and capacity—which can be degraded by 30 percent or more.
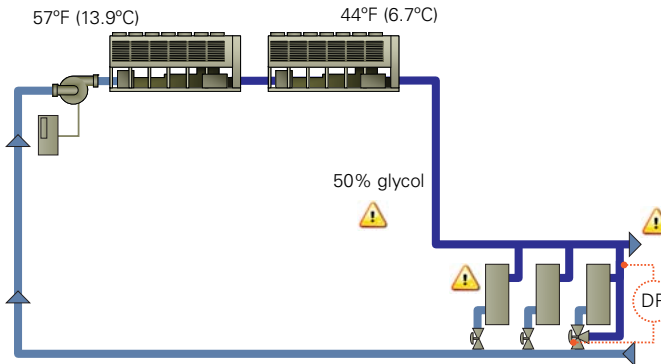
Solution 3: Add only the level of antifreeze necessary. In general, use the smallest amount of antifreeze to adequately protect the system. This minimizes first cost of the antifreeze itself, plus minimizes the first cost and operating cost impact on the heat transfer surfaces and pumps. When adding antifreeze to an existing system, ensure the chiller and coil capacities are adequate given the level of antifreeze.

Final Thoughts

While it's often more memorable to experience issues in the field, it's cheaper and much less painful to learn from others!

We can all listen to those with experience, ask probing questions, and attempt to learn. By doing so the industry can be elevated, deliver higher performing systems that result in more satisfied building owners and occupants.

Figure 6. Retrofit VPF series chillers

Retrofit changes:

- new chillers in series
- send 41°F water to existing coils to increase ΔT
- new pump with VFD
- leave 3-way valves equal to chiller minimum flow rate
- add differential pressure sensor at remote AHU to control pump speed
- add 50 percent glycol to the system for freeze protection

Freeze vs. Burst Protection

As the temperature drops below the glycol solution's freeze point, ice crystals begin to form. Because the water freezes first, the remaining glycol solution is further concentrated and remains a fluid. The resulting ice crystals and fluid combination make up a flowable slush. The fluid volume increases as this slush forms and flows into available expansion volume.

Freeze protection indicates the concentration of antifreeze required to prevent ice crystals from forming at the given temperature. Burst protection indicates the concentration required to prevent damage to equipment (e.g., coil tubes bursting). Burst protection requires a lower concentration of glycol, which results in less degradation of heat transfer.

For a chilled-water VAV system, since the cooling coil is typically shut off during sub-freezing weather, burst protection is usually sufficient. Freeze protection is mandatory in those cases where no ice crystals can be permitted to form (such as a coil loop that operates during very cold weather).

When an air-cooled chiller is used, an alternative approach is to use a packaged condensing unit (condenser and compressor) located outdoors, with a remote evaporator barrel located in an indoor equipment room. The two components are connected with field-installed refrigerant piping. This configuration locates the part of the system that is susceptible to freezing (evaporator) indoors and still uses an outdoor air-cooled condenser.

Concentration required for freeze protection vs. burst protection

temperature °F (°C)	ethylene glycol concentration volume (% volume)		propylene glycol concentration volume (% volume)	
	freeze protection	burst protection	freeze protection	burst protection
20 (-7)	16	11	18	12
10 (12)	25	17	29	20
0 (-18)	33	22	36	24
-10 (-23)	39	26	42	28
-20 (-29)	44	30	46	30
-30 (-34)	48	30	50	33
-40 (-40)	52	30	54	35
-50 (-46)	56	30	57	35
-60 (-51)	60	30	60	35

Source: Dow Chemical Company. 2008. HVAC Application Guide: Heat Transfer Fluids for HVAC and Refrigeration Systems www.dow.com/heattrans

References

[1] American Society of Heating, Refrigerating and Air-Conditioning Engineers. 2010. *ASHRAE Green Guide: The Design, Construction, and Operation of Sustainable Buildings*, 3rd ed. Atlanta, GA: ASHRAE.

[2] Taylor, S. 2011. "Optimizing Design & Control of Chilled Water Plants; Part 3: Pipe Sizing and Optimizing ΔT." *ASHRAE Journal.* 53(12):22-34.

[3] American Society of Heating, Refrigerating and Air-Conditioning Engineers. 2008. *ASHRAE Guideline 22: Instrumentation for Monitoring Central Chilled Water Plant Efficiency*, Appendix E. Atlanta, GA: ASHRAE.

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(SYS-APM008-EN, updated May 2012)

Water-Source and Ground-Source Heat Pump Systems examines chilled-water-system components, configurations, options, and control strategies. The goal is to provide system designers with options they can use to satisfy the building owners' desires. (SYS-APM010-EN, updated November 2013)