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Achieving High Chilled-Water Delta *T*s

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This article recommends practical methods for achieving high, i.e., 15°F (8.3°C) or greater, chilled-water temperature differentials (Delta *T*s) in variable flow hydronic cooling systems. Once high chilled-water Delta *T*s are realized, more Btu's (J/s) of cooling will be accomplished per gallon (liter) of chilled-water distributed (**Figure 1**). Pressure losses and pumping energy will decline considerably in existing hydronic cooling systems and smaller pumps and piping may be installed in new hydronic cooling systems. Also, water chiller capacities will no longer be limited by maximum evaporator flow rates and chilled-water storage tanks will store many more tonhours (MJ) of cooling. This fundamental approach provides significant, enduring improvements in the performance of variable-flow hydronic cooling systems.

Consider a building served by a variable flow hydronic cooling system (with partial storage) that was designed for a 15°F (8.3°C) chilled-water Delta T. The building's design cooling load is 1,500 tons (1119 kJ/s); the design pressure drop in its secondary chilled-water circuit is 150 ft (249 kPa); and it's served by two 75 hp (56 kJ/s) secondary chilled-water distribution pumps (each capable of 1,500 gpm [94 L/s] chilled-water flow [maximum]). The design capacity of its chilled-water storage tank is 500 tons (384 kJ/s) sustained over a 10- hour discharge period—equating to 5,000 ton-hours (63.4 MJ). Two 500 ton (384 kJ/s) water chillers (with three-pass evaporators selected at 7.5 fps [2.2 m/s] tube velocity) cool the building in conjunction with the chilled-water storage tank during the discharge cycle, then recharge the chilled-water storage tank during the 14-hour charge cycle. Consider further that, due to several common shortcomings, the actual chilled-water Delta T is only 10°F (5.6°C) at peak cooling conditions. What will be the effect on peak cooling performance?

- The flow rate in the secondary chilled-water circuit must increase by 50%—from 2,400 gpm (151 L/s) to 3,600 gpm (227 L/s).
- The pressure loss in the secondary chilled-water circuit must increase by 125%—from 150 ft (249 kPa) to 338 ft (561 kPa).
- Assuming 80% pump efficiency, pumping energy in the secondary chilled-water circuit must increase by 237%—from 114 hp (85 kJ/s) to 384 hp (286 kJ/s).
- The additional pumping energy (if available) in the secondary chilled-water circuit will increase the peak cooling load by 4%—from 1,500 tons (1119 kJ/s) to 1,557 tons (1162 kJ/s).
- Chilled-water storage capacity will decrease by 33%—from 5,000 ton-hours (63.4 MJ) to 3,335 tonhours (42.3 MJ)—permitting only 333 tons (256 kJ/s) of cooling to be sustained over a 10-hour discharge period.
- The capacities of the water chillers will be limited to 446 tons (333 kJ/s) each (89% of design capacity) if maximum evaporator tube velocity of 10 fps (3 m/s) is observed.
- The combined capacity of the water chillers and storage tank will be limited to a total of 1,225 tons (914 kJ/s) versus a peak cooling load of 1,557 tons (1162 kJ/ s)—resulting in a 332 ton (248 kJ/s) cooling capacity shortfall (21%).

It's evident that the low/below-design chilled-water Delta *T* will prevent this building's cooling load from being satisfied at peak cooling load conditions. If the results of this example sound extreme, please read the actual case study.[1]

Table 1 : Electrical/cooling analogy

	Electrical Distribution	**Hydr Hydronic onic Cooling**
Symptoms	High Current Demand and High Voltage Losses in Conductors and Transformers	High Flow Demand and High Pressure Losses in pipes and Heat Exchangers
Problems	Low Power Factor	Low Delta *T*
Root	Large, Underloaded	Improperly Designed,

Cause	Induction Motors	Operated and/or Maintained Cooling Loads
Solutions	Power Factor Correction at Inductions Motors	Delta T Correction at Cooling Loads
Benefits	Increased Power Distribution and Lower Current Demand and Lower Voltage Lossed	Increased Cooling Distribution and Lower Flow Demands and Lower Pressure Losses

So, if the adverse consequences of low/below-design chilled-water Delta *T*s are so severe, why are low/belowdesign chilled-water Delta *T*s so common?

- Low, e.g., 10°F (5.6°F), design chilled-water Delta *T*s result when standard rating conditions for waterchilling packages are adopted as the design Delta *T* for variable flow hydronic cooling systems.
- Below-design chilled-water Delta *T*s result because many things must be done correctly to achieve equalto- design or greater-than-design chilled-water Delta *T*s, while only a few errors or omissions will result in belowdesign chilled-water Delta *T*s.

For example, computer room air-conditioning units that have standard four-row, A-frame cooling coils with 10°F (5.6°C) design chilled-water Delta *T*s and that contain standard three-way bypass control valves are frequently furnished and installed in variable flow hydronic cooling systems. The consistent and predictable results are low/below-design chilled-water Delta *T*s in the affected systems, plus associated performance problems. Thorough specifications that require the manufacturer to furnish optional six-row cooling coils and optional two-way control valves will preclude this from happening.

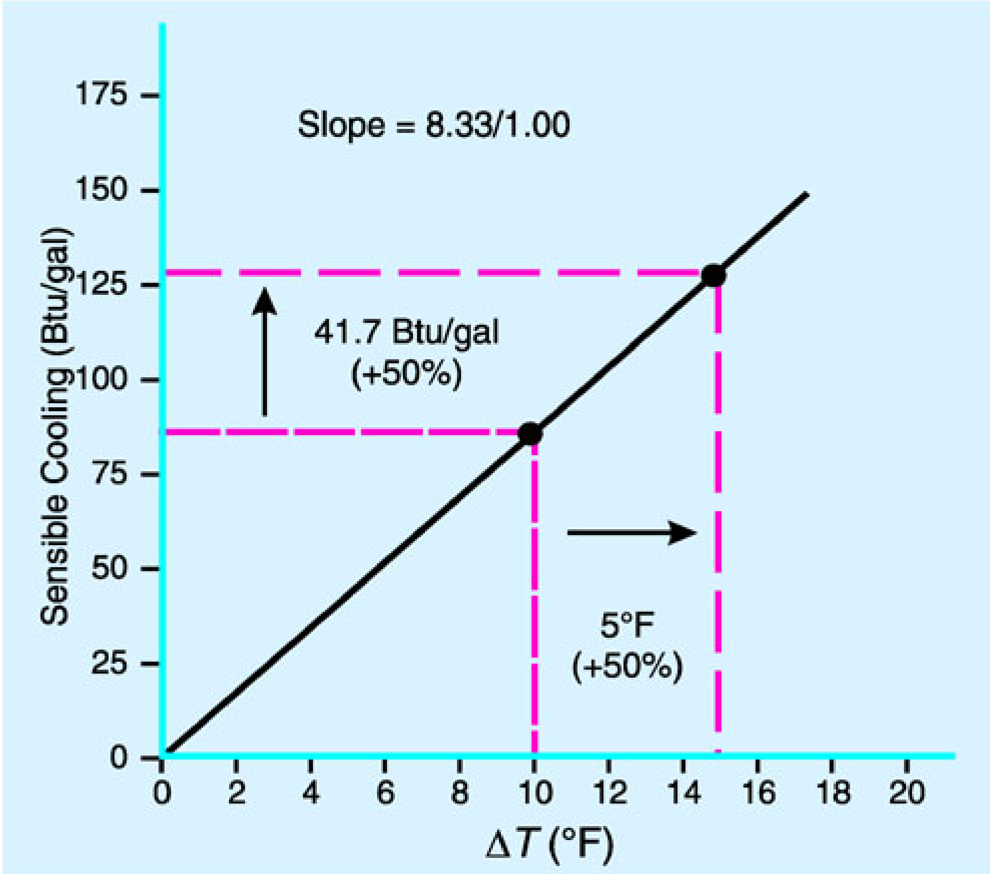

Figure 1 : Hydronic cooling leverage

Analogy: Electrical Distribution Systems

A useful analogy for understanding how to raise Delta *T*s in variable flow hydronic cooling systems is power factor correction in ac electrical distribution systems (**Table 1**). In effect, a low power factor is equivalent to a low Delta *T*. Furthermore, operating an ac electrical distribution system with a low power factor is as problematic as operating a variable-flow hydronic cooling system with a low Delta *T*.

A low power factor (Delta *T*) results from large, underloaded induction motors (poorly-designed, installed, controlled, and/or maintained cooling loads). It's evidenced by high current (flow) demand and high voltage (pressure) losses in conductors (pipes) and transformers (heat exchangers). In addition, the conductors (pipes) and transformers (heat exchangers) can readily distribute additional power (cooling) by simply raising the power factor (Delta *T*). Finally, increasing the power factor (Delta *T*) is best accomplished at the motors (cooling loads).

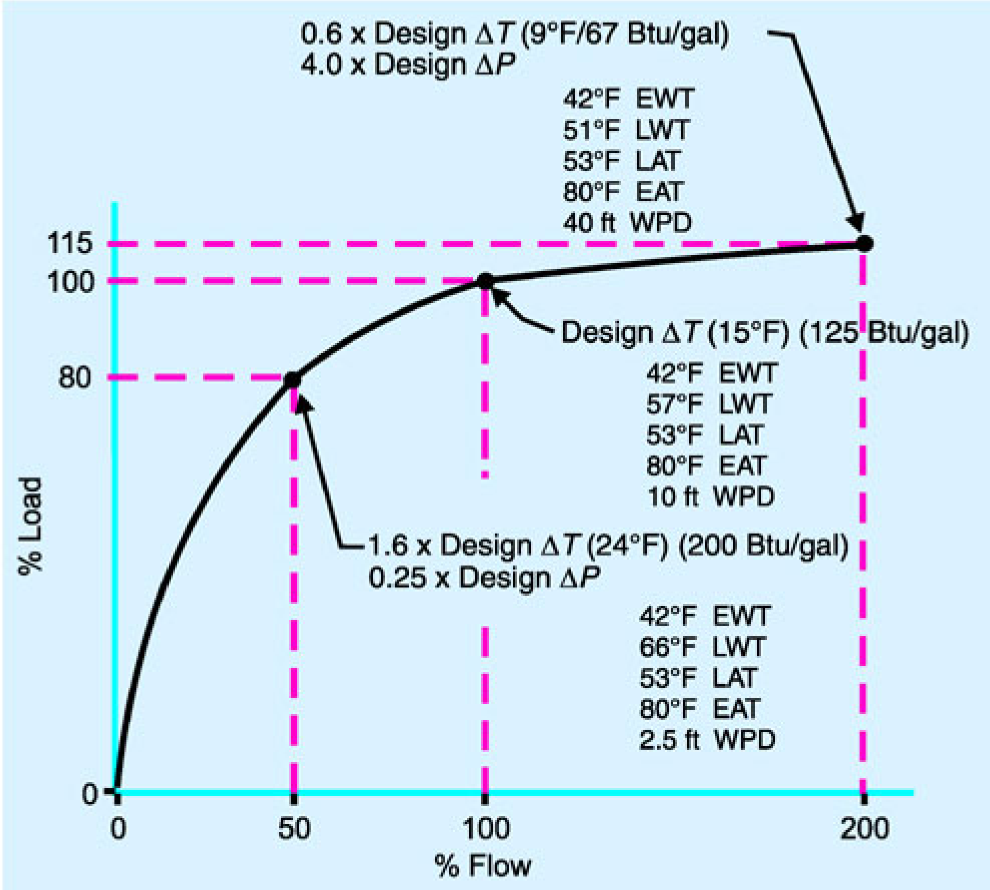

Figure 2 : Cooling coil characteristics

Background

It's obvious that the chilled-water flow rate in a variable flow hydronic cooling system should decrease as the cooling load decreases. What is not obvious— and due to bypassing and such is rarely observed in practice—is that the chilled-water Delta *T* in a variable flow hydronic cooling system should be equal to design at full load and greater than design at part load.

To clarify this important point, a cross-flow cooling coil with 42°F (5.6°C) entering chilled-water temperature, 80°F (26.7°C) mixed entering air temperature, and 53°F (11.7°C) leaving air temperature has a non-linear performance characteristic such that 80% of the design cooling output can be obtained with 50% of the design chilled water flow (**Figure 2**). Thus, the chilled-water Delta *T* in a variable flow hydronic cooling system should be 60% greater-than-design at 80% of the design cooling load.

However, the actual chilled-water Delta *T* is limited by the fact that the leaving chilled-water temperature cannot exceed the entering air temperature. Even so, with a design Delta *T* of 15°F (8.3°C) or 125 Btu/gallon (35.2 kJ/L) at 100% load, a part-load Delta *T* of 24°F (13.3°C) or 200 Btu/gallon (56.2 kJ/L) is achievable at 80% load. This corresponds to a return chilled water temperature of 57°F (13.9°C) at 100% load, increasing to 66°F (19.8°C) at 80% load. Conversely, this corresponds to an approach to the entering air temperature of 23°F (12.8°C) at 100% load, decreasing to 14°F (7.8°C) at 80% load.

Also, it's important to realize that 200% of cooling coil design chilled-water flow will only yield 115% of design cooling coil output (**Figure 2**). Furthermore, this condition will result in 40% lower-than-design cooling coil Delta *T*s. It will also starve other terminal devices of necessary chilled-water flow. So twoposition control valves, as well as modulating control valves that remain 100% open under all load conditions (due to dirty cooling coils/filters, sub-set thermostats, etc.), are especially harmful in variable flow hydronic systems.

Objectives

- Design or above-design chilled-water Delta *T*s are assured from 1% to 100% load conditions in properlydesigned, installed, controlled, and maintained variable flow hydronic cooling systems.
- A design Delta *T* of 15°F (8.3°C) to 18°F (10.0°C) is readily achievable in comfort cooling applications with 42°F (5.6°C) entering chilled-water temperature, 80°F (26.7°C) mixed entering air temperature, and 53°F (11.7°C) leaving air temperature.

These objectives can be achieved by proper application of cooling coils, control valves, control systems, distribution pumps, and piping systems.

Approach

To apply resources most effectively, it's important to realize that chilled-water Delta *T*s are determined by the various terminal devices, a.k.a., cooling loads—and not by the central water-chilling plant. So, to effectively raise chilled-water Delta *T*s, the design, installation, operation, and maintenance of the cooling coils, control valves, and control systems located in equipment rooms throughout facilities must be addressed. This is the fundamental solution that provides high leverage and yields enduring improvements.

After this is understood, it is easy to realize that installing more water chillers, larger chilled-water pumps, and/or larger chilled-water pipelines at the central waterchilling plant will not overcome the poor performance problems associated with low/below-design chilled-water Delta *T*s in variable flow hydronic cooling systems. That symptomatic approach is capital-intensive, energyintensive, and misses the fundamental problem. This is mentioned because the symptoms of low/below- design chilled-water Delta *T*s, i.e., poor cooling performance, are frequently misinterpreted as evidence of inadequate capacity at the central water-chilling plant (refer to the example given at the beginning and the case study cited in the reference section of this article).

Best Practices

Following are 25 "best practices" to achieve high chilled-water Delta *T*s. They range from component selection criteria to distribution system configuration guidelines and are applicable to new installations as well as retrofit projects. All have been successfully implemented at a 6.8 × 106 ft2 (631.7 × 103 m2) semiconductor manufacturing complex in Dallas. Their leveraged, synergistic effect improved that large facility's peak composite Delta *T* by 50%—from 12°F (6.7°C) in 1993 to 18°F (10.0°C) in 1998.

1. Schedule cooling/dehumidifying coils for high Delta *T*s. Schedule cooling/dehumidifying coils to provide a 25% higher chilled-water Delta *T* than the chilled-water storage tank and the evaporators of the water chiller(s) are designed for. Also, schedule cooling/ dehumidifying coils with a chilled-water entering temperature 3°F (1.7°C) higher than the chilled-water storage tank and the evaporators of the water chiller(s) are scheduled to supply. Using an automated cooling coil selection program, start with full-circuit serpentine coils having eight rows of 5/8 in. (17.1 mm) diameter copper tubes spaced 12 per inch (one per 2 mm). Airside velocity should be below 500 fpm (2.5 m/s) and the airside pressure drop should be below 0.8 in. of water (2 kPa) at the design airflow rate. Also, waterside tube velocity should be below 5 fps (1.5 m/s) and waterside pressure drop should be below 10 ft of water (30 kPa) at design chilled-water flow rates. Finally, double-check waterside performance to preclude laminar flow at reduced cooling loads.

2. Specify modulating two-way globe-style control valves with equal percentage plugs. As discussed earlier, a cooling coil provides 80% of design cooling with 50% of design chilled-water flow. An equal-percentage valve plug provides 50% of design chilled-water flow at 80% open. The result of these two characteristics is linear control of cooling coil performance. Specify 50:1 minimum rangeability to assure accurate plug positioning at reduced cooling loads. Also, specify globestyle control valves to assure more precise plug positioning than ball-style control valves. And, because they have quick-opening characteristics, avoid butterflystyle control valves for control of cooling coil output. Most importantly, size chilled-water control valves to waste 100% of excess branch circuit pressure differential when fully-open at the respective cooling coil design chilled-water flow rate. This will ensure adequate control valve "authority."

3. Specify robust control valve actuators. Chilledwater control valve actuators must have shut-off ratings that exceed the highest branch circuit differential pressure that the chilled-water pump(s) can generate. This is especially important for the control valves located in the branch circuits nearest to the chilled-water pumps in linear, direct-return piping systems. Otherwise, valve plugs will be forced off their seats when the branch circuit differential pressure rises above the actuator shut-off rating, allowing chilled-water to leak-by. Also, accurate plug positioning is impossible when the valve actuator does not have enough power to overcome the branch circuit differential pressure. A 50% safety factor is recommended.

4. Specify robust control valve cages, trim, plugs and seals. Chilled-water control valves must be rated for pressure drop service in excess of the highest chilledwater differential pressure that the chilled-water pump(s) can generate. Otherwise, high velocities and throttling will cause valve seats to deteriorate and chilled-water will leak-by when the control valve is closed. A 50% safety factor is recommended.

5. Omit external balancing devices. Variable flow hydronic cooling systems must be designed to be selfbalancing. This means that control valves must be sized to control with authority over the entire load and pressure range. For example, if serving identical cooling coils in a direct-return piping system, control valves in branch circuits nearest to the chilled water pump(s) should be smaller than control valves in branch circuits furthest from the chilled-water pumps. This will enable the former to "waste" the excess differential pressure that will be present in the branch circuits nearest to the chilled water pumps.

Once this is accomplished, external balancing devices are unnecessary. In fact, external balancing devices reduce control valve authority and are counterproductive. Furthermore, they add unnecessary branch circuit pressure drops. The only place external balancing devices are really needed in variable flow hydronic cooling systems are at piping manifolds serving large, stacked coil air-handling units.

6. Specify digital control. A properly functioning digital control system is more accurate than a properly functioning pneumatic control system and will position chilled-water control valves with less offset and drift.

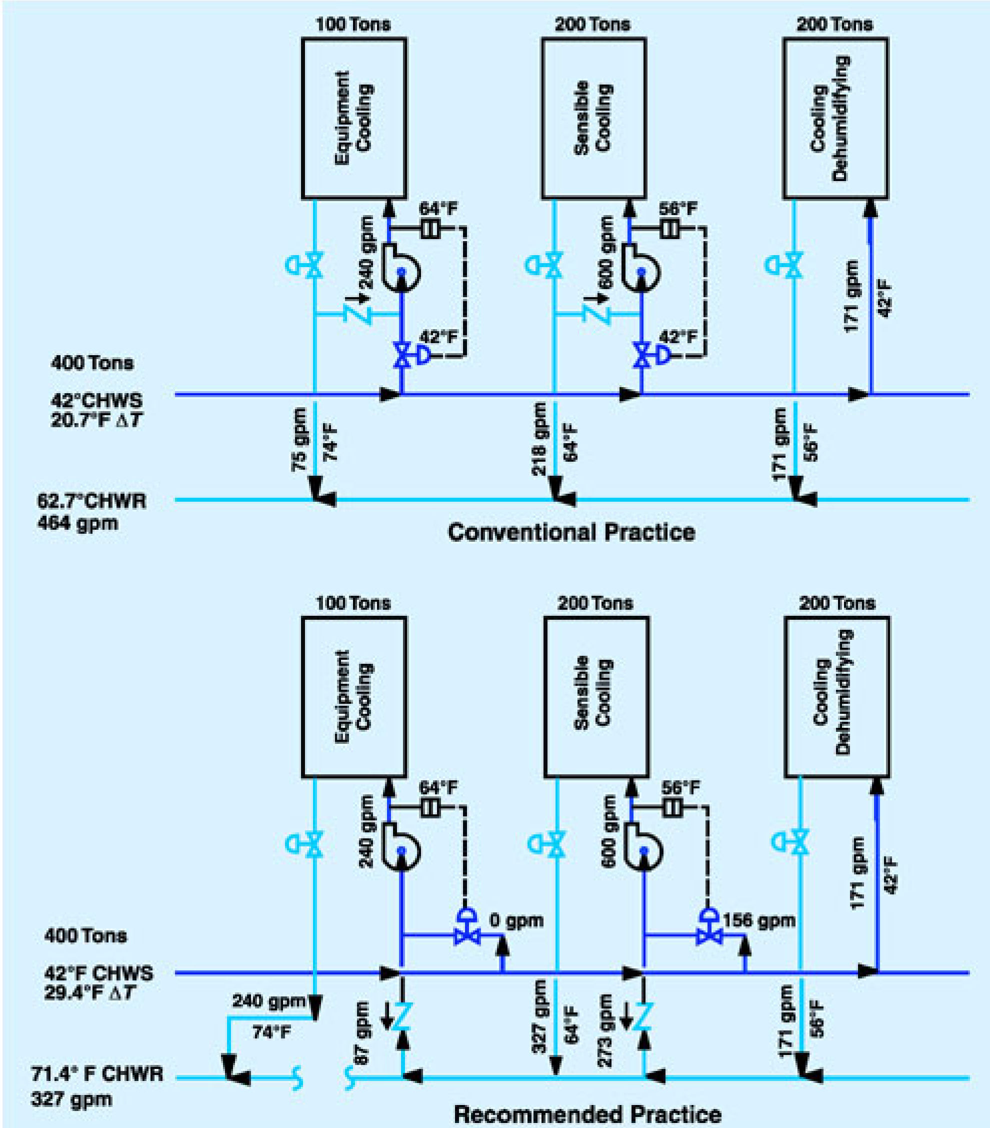

Figure 3 : Cascade cooling

7. Use chilled-water several times before allowing it to return. Chilled-water is not required to return to the water chilling plant after a single pass through one terminal device (**Figure 3**). For example: 1) use the 56°F (13.3°C) chilled-water returning from cooling/ dehumidifying coils as the supply for sensible cooling coils; 2) then use the 64°F (17.9°C) chilled-water returning from the sensible cooling coils as supply for watercooled equipment, e.g., vacuum pumps, air compressors, etc. in order to maximize the chilled-water Delta *T*. However, be sure that the 74°F (23.3°C) chilled-water returning from water-cooled equipment returns directly to the water chilling plant without further recycling.

8. Use chilled-glycol in low temperature/humidity applications. In practice, 5°F (2.8°C) is about the closest approach to the leaving air temperature that a single cooling/dehumidifying coil can provide while still yielding a high chilled-water Delta *T* according to the cooling coil selection criteria recommended earlier.

For a variable flow hydronic cooling system with a 42°F (5.6°C) chilled-water supply temperature, this will favor the use of chilled-glycol in applications requiring 47°F (8.3°C) or lower leaving air temperatures, e.g., makeup air units for clean rooms with 44°F (6.7°C) dew point requirements. This practice prevents one low temperature/humidity application from requiring that the chilled-water temperature be sub-set for the entire facility.

9. Omit unnecessary water-to-water heat exchangers. As a minimum, there will be a 2°F (1.1°C) temperature difference between the hot inlet and cold outlet at a cooling heat exchanger.

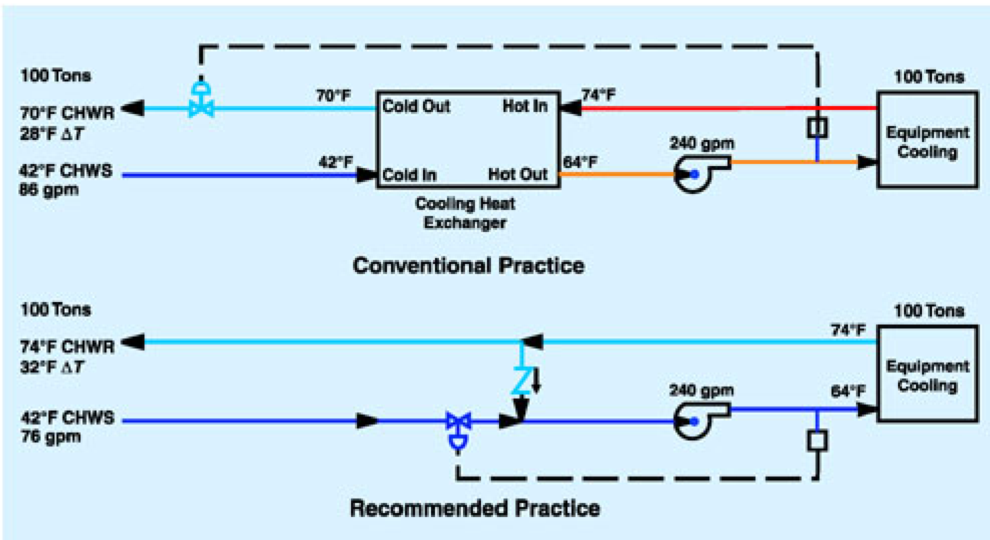

Figure 4 : Blended-water cooling

If the hot and cold pressures are in the same ANSI class and if the hot and cold fluids are identical, it is better to omit the cooling heat exchanger and blend the hot and cold fluids to produce the desired hot outlet temperature (**Figure 4**) and elevate the chilled-water Delta *T*. Configure the piping so that the excess water from the hot outlet, i.e., the warmest water, returns to the water chilling plant.

10. Treat sensible cooling and cooling/ dehumidifying separately. In order to sub-cool the entering air to saturation and remove moisture, a cooling/ dehumidifying coil will generally require a lower entering water temperature than a sensible cooling coil. Therefore, in applications such as clean rooms and computer rooms, it's desirable to install sensible cooling coils to control the temperature of the large volume of re-circulated air and install separate side-stream cooling/ dehumidifying coils to control humidity. This provides the opportunity to supply warmer chilled-water to the sensible cooling coils and realize a higher composite chilled-water Delta *T*.

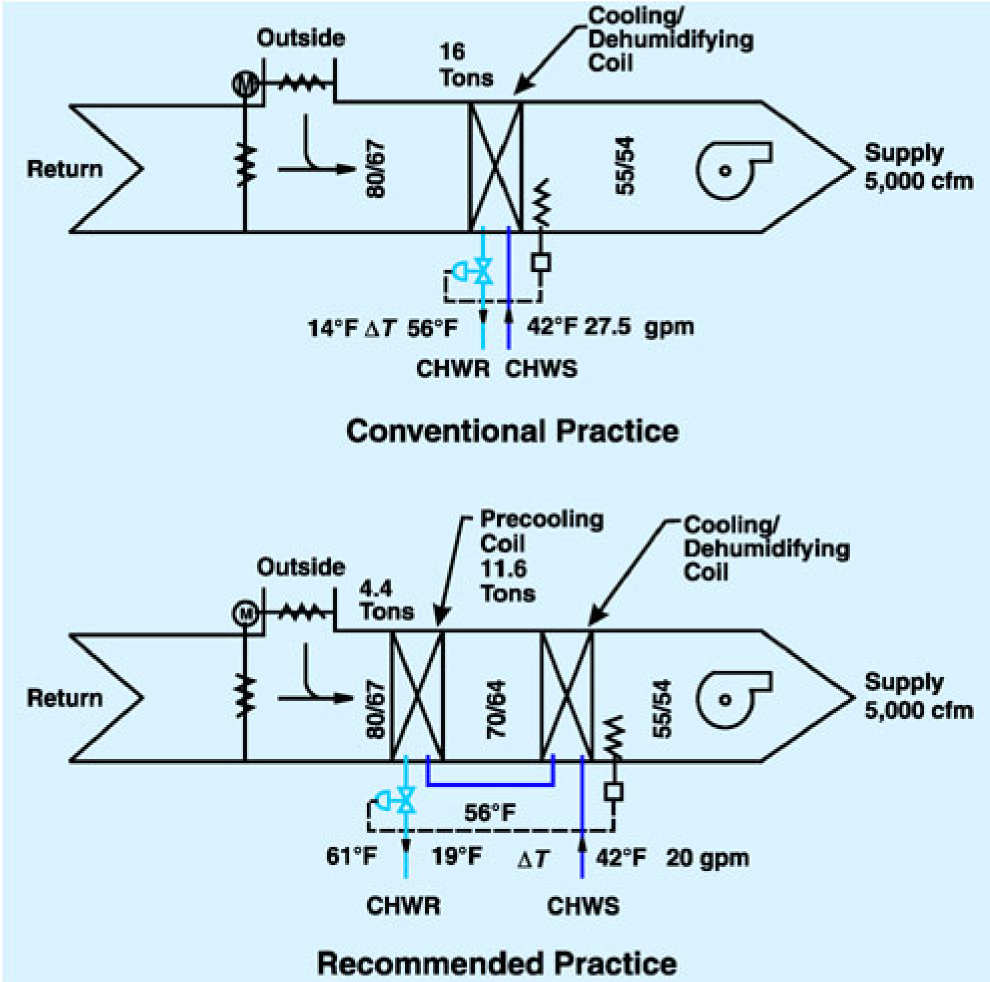

Figure 5 : Series cooling coils

11. Use two cooling coils in series. Two cooling coils in series (**Figure 5**) create counter-flow air-to-water cooling rather than cross-flow air-to-water cooling, resulting in a higher chilled-water Delta *T*. This method is especially effective for 100% outdoor air applications requiring a large airside Delta *T*. However, increase coil face areas in order to reduce air velocities, avoid increased airside pressure drops, and avoid increased fan energy consumption.

12. Use runaround precooling/re-heating coils. Run-around pre-cooling/re-heating coils are an effective method to reduce the load on the cooling/dehumidifying coil and, in turn, increase the chilled-water Delta *T* and reduce flow demand (**Figure 6**). The pre-cooling and reheating is “free” throughout the cooling season. This method is especially effective for 100% outdoor air applications requiring a large airside Delta *T* and reheating.

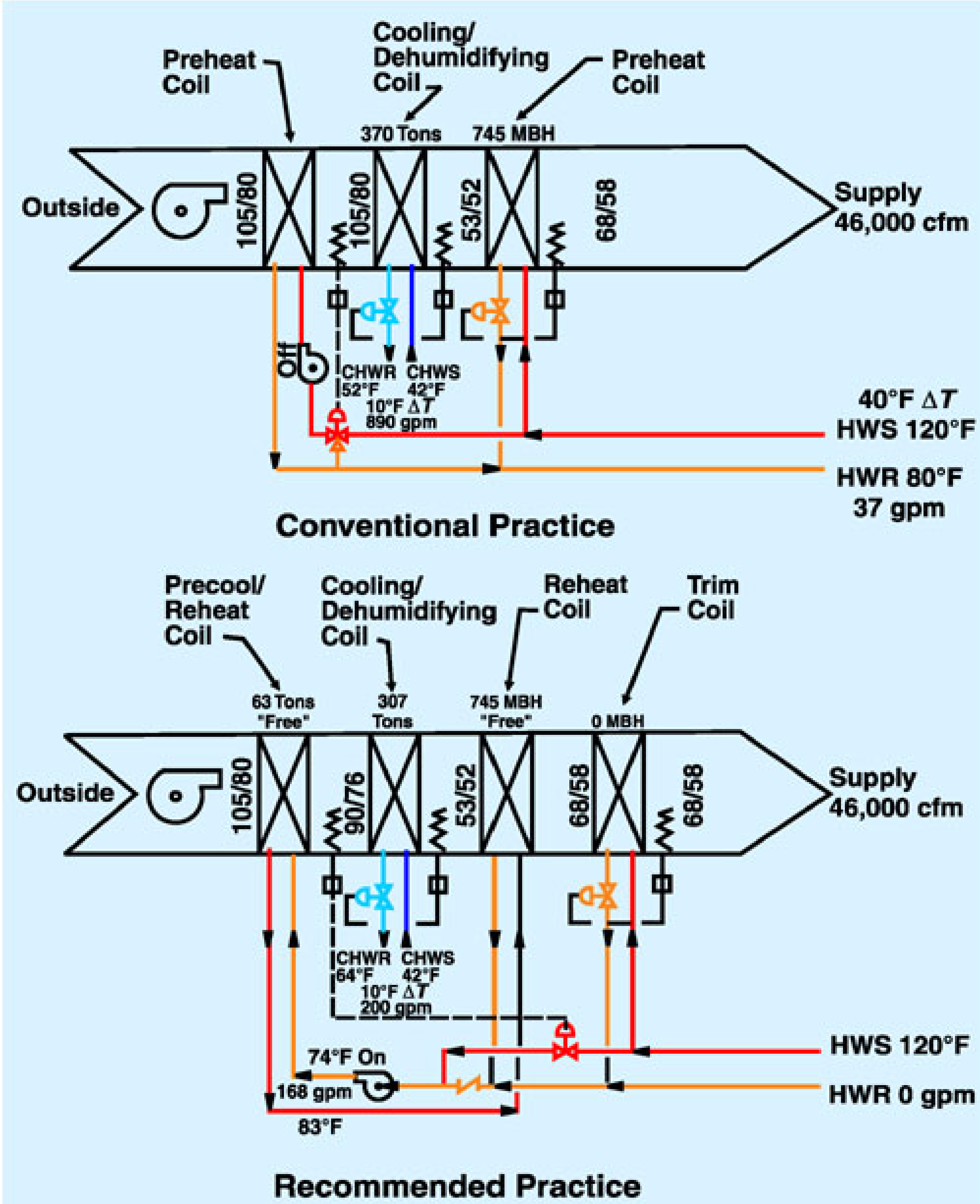

Figure 6 : Runaround coils

However, increase coil face areas in order to reduce air velocities, avoid increased airside pressure drops, and avoid increased fan energy consumption.

13. Replace three-way bypass control valves. As discussed earlier, these are detrimental in variable flow hydronic cooling systems and must be replaced with twoway control valves wherever they are encountered. This is because three-way bypass control valves allow chilledwater to bypass the cooling coil and return to the water chilling plant at supply temperature.

Three-way bypass control valves also permit increased total flow at partial cooling loads. Because three-way bypass control valves are normally furnished with less powerful actuators than two-way control valves, simply plugging the bypass port generally will not be adequate.

14. Close control valves when air-handling unit fans are off. When the fan in an air-handling unit is off, the respective chilled-water control valve must close. Otherwise, chilled-water will return to the water chilling plant at supply temperature. This best practice is so obvious that it's often overlooked. However, it should not preclude opening the chilled-water control valve as a last-ditch freeze protection measure in the event of a pre-heat system malfunction.

15. Calibrate temperature and humidity sensors. If these read high, chilled-water control valves will open further than necessary and Delta *T*s will be reduced. Poor sensor location, e.g., an air temperature sensor exposed to direct sunlight, can have the same effect.

16. Protect temperature and humidity setpoints. The best practice is to enter the setpoints into an appropriate software program that has password protection. Otherwise, occupants or mechanics attempting "quick fixes" to cooling problems are likely to lower the setpoints. The root causes of the cooling problems, e.g., inadequate air distribution, will be difficult to solve if random setpoint changes are permitted, and zone temperature/humidity setpoints will most likely remain sub-set. The case study cited earlier describes how extensive this problem can become.

17. Minimize fouling and airflow restrictions. Inadequate airside flow and internal or external fouling of cooling coils inhibit heat transfer and require more chilled-water flow than necessary, thereby lowering Delta *T*s. Appropriate best practices include use of a fouling factor in cooling coil selections; periodic cleaning of the exterior surfaces of cooling coil tubes/fins; continuous water treatment to preclude corrosion, bio-fouling, and scaling on the interior surfaces of cooling coil tubes; and replacement of air filters as they become clogged.

18. Reduce distribution pump speeds at partial cooling loads. If distribution pump speeds remain constant year-round, chilled-water pressure differentials will rise when chilled-water flow rates decrease at partial cooling loads. This may force chilled-water control valves open if they have inadequate actuators, thereby lowering Delta *T*s. The best practice is to equip distribution pumps with variable speed drives and modulate their speeds to maintain chilled-water pressure differentials in the most remote branch piping circuits at pre-set levels. The control valves will provide better control with constant chilled-water pressure differentials and annual distribution pump energy will be reduced considerably.

19. Use multizone and primary/secondary distribution pumping. Consider a central water chilling plant that serves two buildings. Building 1 requires 30 psi (50 kPa) differential pressure at its service entrance under design cooling conditions and Building 2 requires 50 psi (83 kPa). It is best for the central water chilling plant to maintain a 5 psi (8 kPa) primary differential pressure at both building service entrances and equip each building with an appropriately sized variable speed secondary chilled-water distribution pump station.

This compares favorably to an all-primary distribution system with the water chilling plant maintaining a 50 psi (83 kPa) pressure differential at the service entrance to both buildings.

This approach is particularly effective if Building 1 has a larger cooling load than Building 2, because as much as 20 psi (33 kPa) of excess pressure differential must be continuously wasted by the chilled-water control valves in Building 1.

20. Reduce pressure differentials at partial cooling loads. A cooling coil can provide 80% of design cooling output with 50% of design chilled-water flow. Also, pressure drops at 50% of design chilled-water flow are only 25% of those at 100% of design chilled-water flow. This means that chilled-water differential pressure setpoints can be reduced at partial cooling loads without adverse impact on cooling coil performance. If the distribution pumps are equipped with variable speed drives, this best practice saves considerable distribution pumping energy on an annual basis. It also promotes high Delta *T*s and stable control valve performance. However, be sure that cooling coils and heat exchangers serving constant, year-round cooling loads, e.g., clean rooms and computer rooms, are selected for the lowest expected chilled-water pressure differential.

21. Elevate supply temperatures at reduced cooling loads. When the outdoor air dew point is below the desired indoor air dew point, only sensible cooling is required and the chilled-water supply temperature can be raised without adverse impact on cooling coil performance.

This practice will promote high Delta *T*s and help preclude laminar chilled-water flow conditions inside lightly-loaded cooling coils.

However, be sure that cooling coils and heat exchangers serving constant, year-round cooling loads, e.g., clean rooms and computer rooms, are selected for the highest expected chilledwater supply temperature.

22. Design reverse-return and loop-style distribution systems. Reverse return and loopstyle chilled-water distribution systems provide more uniform branch circuit pressure differentials than linear, direct-return chilled-water distribution systems. This promotes stable control valve performance and high chilled-water Delta *T*s.

23. Eliminate constant speed "booster" pumps. Constant speed chilled-water "booster" pumps are detrimental in variable flow hydronic cooling systems and should be equipped with variable speed drives wherever they are encountered. Otherwise, they "rob" chilled water from adjacent cooling coils and heat exchangers at high flow conditions and generate high chilled-water pressure differentials at low flow conditions. Neither case is conducive to high chilled-water Delta *T*s.

24. Replace marginal cooling coils and heat exchangers. In every variable flow hydronic cooling system, a few marginal cooling coils or heat exchangers exist. These require lower chilledwater supply temperatures and/or higher chilled-water pressure differentials than all of the others due to undersizing, over-loading, control valve too small, etc. As these "bad actors" are replaced, cooling performance will improve and Delta *T*s will increase.

25. Monitor Delta *T*s. If you can't measure it, you can't manage it. So provide chilledwater temperature indicators/ sensors at the inlets and outlets of all cooling coils and heat exchangers, as well as at branch points in distribution systems.

With the information that these devices provide, low/ below-design chilled-water Delta *T*s will become readily evident and can be traced quickly to their source(s).

Delta *T* Techniques echniques

- In existing hydronic cooling systems, high, e.g., 15°F (8.3°C) or greater, Delta *T*s reduce chilledwater flow rates and pressure drops, as well as pumping energy and operating costs.
- In new hydronic cooling systems, high, e.g., 15°F (8.3°C) or greater, Delta *T*s permit smaller pumps and piping to be installed.
- Adopting standard rating conditions for waterchilling packages, i.e, 10°F (5.6°C) chilled-water Delta *T*s, as the design Delta *T* for variable-flow hydronic cooling systems will result in low Delta *T*s.
- One error or omission in project drawings/specifications may result in low/below-design chilled-water Delta *T*s.

- Low/below-design chilled-water Delta *T*s reduce the ability of heat exchangers and pipes to deliver hydronic cooling—just like low power factors reduce the ability of transformers and conductors to deliver electrical power.
- The chilled-water Delta *T* should be equal-to-design at full-load and greater-than-design at part-load in a variable-flow hydronic cooling system.
- Chilled-water Delta *T*s are determined by a building's various terminal devices, a.k.a., cooling loads, not by its central water-chilling plant.
- High chilled-water Delta *T*s result from proper design, installation, operation, and maintenance of cooling coils, control valves, control systems, distribution pumps, and distribution piping.
- More water chillers, larger chilled-water pumps, and/or larger chilled-water piping will not overcome the performance problems resulting from low/ below-design chilled-water Delta *T*s.

Cost of Ownership

So, what are the first cost implications of measures to assure high chilled-water Delta *T*s? In general, eight-row cooling coils will cost more than six-row cooling coils, particularly when enlarged face areas are provided to avoid increased airside pressure drops. Also, industrial-quality control valves with robust actuators will cost more than commercialquality control valves with less robust actuators, but omitting the external balancing devices required for the latter will mitigate the first cost difference.

On the other hand, a hydronic cooling system with a design Delta *T* of 15°F (8.3°C) will require smaller distribution pumps and piping (with a lower first cost) than a hydronic cooling system with a 10°F (5.6°F) design Delta *T*. The system with the higher Delta *T* will also require a smaller chilled-water storage tank (with a lower first cost). These considerations will tend to tilt the scale towards lower first cost for high Delta *T* designs in large hydronic cooling systems having fewer/larger air-handling units, more distribution pumps, greater lengths of distribution piping, and chilled-water storage tanks. Good candidates include district cooling systems that serve large campuses with multiple buildings.

Operating cost considerations clearly favor hydronic cooling systems having higher chilled-water Delta *T*s due to lower chilled-water flow rates and pressure drops yearround.

Also, as mentioned earlier, providing cooling coils with enlarged face areas will preclude increased airside pressure drops (and increased fan energy consumption). Large hydronic cooling systems have the potential to realize the greatest operating cost savings from high Delta *T* designs.

In all cases, specific application conditions will determine first cost and operating cost differences resulting from alternate hydronic cooling system design and operation practices.

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