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Selecting Chilled-Water Coils for

ASHRAE 90.1's New 15°F Delta T Requirement

The 2016 version of ASHRAE Standard 90.1 requires chilled-water cooling coils to be selected for at least a 15°F ΔT.

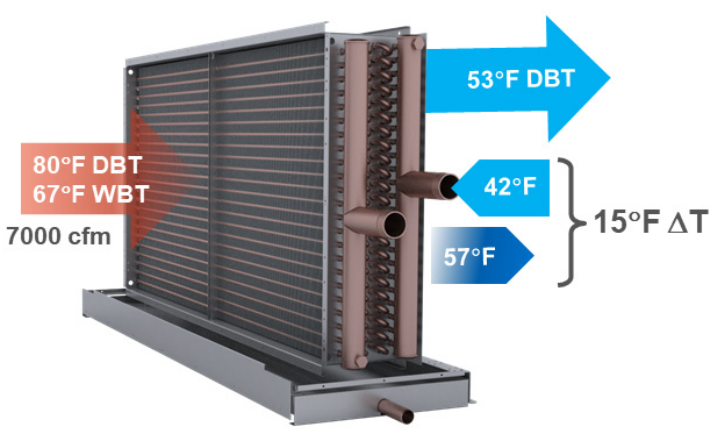
This Engineers Newsletter reviews this new requirement, demonstrates how it affects selection of chilled-water coils, and analyzes the impact of part-load conditions on coil performance.

New ASHRAE 90.1 Requirement

Beginning in its 2016 version, ASHRAE Standard 90.1 now requires chilled-water cooling coils be designed for at least a 15°F temperature difference (ΔT) between the leaving and entering water; and the leaving-water temperature must be no colder than 57°F, at design conditions.[1]

As an example, for the leaving-water temperature to be 57°F, the coil in Figure 1 is selected with a 42°F entering-water temperature. This would comply with the minimum 15°F ΔT requirement.

Figure 1. Chilled-water cooling coil



Committee's intent. Sometimes it is helpful to understand the committee's intent when a new requirement like this is added to Standard 90.1. Other than attending the committee meetings, the best way to understand intent is to read the foreword that is included when an addendum is released for public review.

6.5.4.7 Chilled-Water Coil Selection.

Chilled-water cooling coils shall be selected to provide a 15°F or higher temperature difference between leaving and entering water temperatures and a minimum of 57°F leaving water temperature at design conditions.

Exceptions to Section 6.5.4.7:

1. Chilled-water cooling coils that have an airside pressure drop exceeding 0.70 in. H_2O when rated at 500 fpm face velocity and dry conditions (no condensation)
2. Individual fan-cooling units with a design supply airflow rate ≤ 5000 cfm
3. Constant-air-volume systems
4. Coils selected at the maximum temperature difference allowed by the chiller
5. Passive coils (no mechanically supplied airflow)
6. Coils with design entering chilled-water temperature ≥ 50°F
7. Coils with design entering air dry-bulb temperature ≤ 65°F

As stated in the foreword that accompanied this addendum (see excerpt below), the committee's intent was to encourage the use of coils with more heat transfer surface to achieve this higher ΔT.[2] It cited a life-cycle cost analysis that was published in the *ASHRAE Journal*.[3] This analysis showed that designing a chilled-water system for a higher ΔT reduced first costs—due to smaller valves, piping, and pumps—and also reduced overall system energy cost. The results showed that the increase in fan energy due to the added coil surface was more than offset by the pump energy savings.

> "High ΔT was shown to result in optimum energy efficiency and life cycle costs in analysis by Taylor in Optimizing Design & Control of Chilled Water Plants, Part 3: Pipe Sizing and Optimizing ΔT (ASHRAE Journal, December 2011). The analysis showed that the fan energy increase due to the larger coil was more than offset by the pump energy savings, and net first costs were reduced due to smaller piping and pumps, offsetting higher coil costs. So both first costs and energy costs are reduced by this requirement...The intent of this addendum is to encourage the use of coils with larger heat transfer surface areas to generate a high ΔT."

That same article also showed that using a slightly colder entering-water temperature would allow for a coil selection to achieve the desired higher ΔT with no impact on supply fan energy use—that is, no change in surface area or air pressure drop compared to the 10°F ΔT baseline. In that case, the results showed that while the colder water does increase chiller energy use, this was more than offset by the pump energy savings.

Exceptions. While this is now a prescriptive requirement, like other sections of Standard 90.1, there are several exceptions listed (see excerpt on page 1).

Per exception 2, individual fan-cooling units that supply 5000 cfm or less are exempt. This exempts fan-coils, blower-coils, classroom unit ventilators, and small air handlers. According to the foreword, the committee included this exception because most small fan-coils do not have an option for an eight-row coil. But as demonstrated later in this EN, eight rows is not necessarily required to achieve a 15°F ΔT. It is still a good idea to select smaller units with a higher ΔT. However, Standard 90.1 does not currently require it.

Per exception 3, if airflow across a cooling coil is constant volume, that coil is also exempt from this requirement. For the first analysis in the cited article, the author used a coil selected with more surface area, resulting in a higher air pressure drop. For the VAV system analyzed, the resulting increase in fan energy was more than offset by the pump energy savings. However, as mentioned above, the article also showed that by using a slightly colder entering-water temperature, there was no change in air pressure drop (zero impact on fan energy) compared to the 10°F ΔT baseline. In that analysis, the higher ΔT still resulted in lower overall energy use. So even in a constant-volume system, it is a good idea to design for a higher ΔT, but Standard 90.1 currently does not require it.

Per exception 6, coils that are selected with an entering chilled-water temperature of 50°F or warmer are also exempt. This exempts sensible-only cooling equipment, like radiant panels, chilled beams, and sensible-cooling terminal units, like Trane's CoolSense™.

Affected applications. Some examples of applications where this requirement does apply include:

- Mixed-air, multiple-zone VAV systems, since they have variable airflow and are usually larger than 5000 cfm
- Single-zone VAV air handlers, which might be used in larger zones like auditoriums, arenas, gymnasiums, or manufacturing areas
- Dedicated outdoor-air systems, when designed for variable airflow—which would be the case if demand-controlled ventilation is implemented

Selecting Cooling Coils for ≥ 15°F ΔT

The example chilled-water coil in Figure 1 is in a mixed-air VAV air handler that cools 7000 cfm to 53°F leaving the coil. The entering chilled-water temperature is 42°F, with a 15°F ΔT at design conditions.

Table 1 compares four coils that could be used for this application:

- Coil #1 has six rows of 3/8-inch diameter tubes and turbulators (devices mounted inside the tubes that increase fluid turbulence to improve heat transfer).
- Coils #2 and #3 have six rows of 1/2-inch tubes, one with turbulators and the other without. Notice that turbulators allow coil #3 to provide the required capacity with fewer fins than coil #2. This reduces the air pressure drop, but increases the water pressure drop. (Whether or not this additional water pressure drop impacts the size of pumps, or pump energy use, depends on whether or not this coil is located in the "critical circuit" of the piping system.)
- Coil #4 has six rows of 5/8-inch tubes and turbulators.

For this example, coil #1 is the least expensive option. Coil #4 is the best choice for minimizing both air and water pressure drops, but it costs more than the other options. Coil #2 or #3 might be selected to better balance cost and pressure drops.[4]

Effect of part-load operating conditions. As explained, Standard 90.1 requires a minimum 15°F ΔT at design conditions. This allows for a lower water flow rate (gpm) and a reduction in installed cost due to smaller valves, pipes, and pumps. But the other motivation is to reduce pump energy use, for which part-load operation (off-design conditions) also matters.

For the mixed-air VAV system in this example, the entering-air conditions will change as the outdoor conditions change, and the airflow across the coil will change as the zone-level VAV dampers modulate. Figure 2 shows the resulting water ΔT of these same four coils—each selected to achieve a 15°F ΔT at design conditions—at three different entering-air conditions at which the cooling coil will still be active, and at two different airflows.

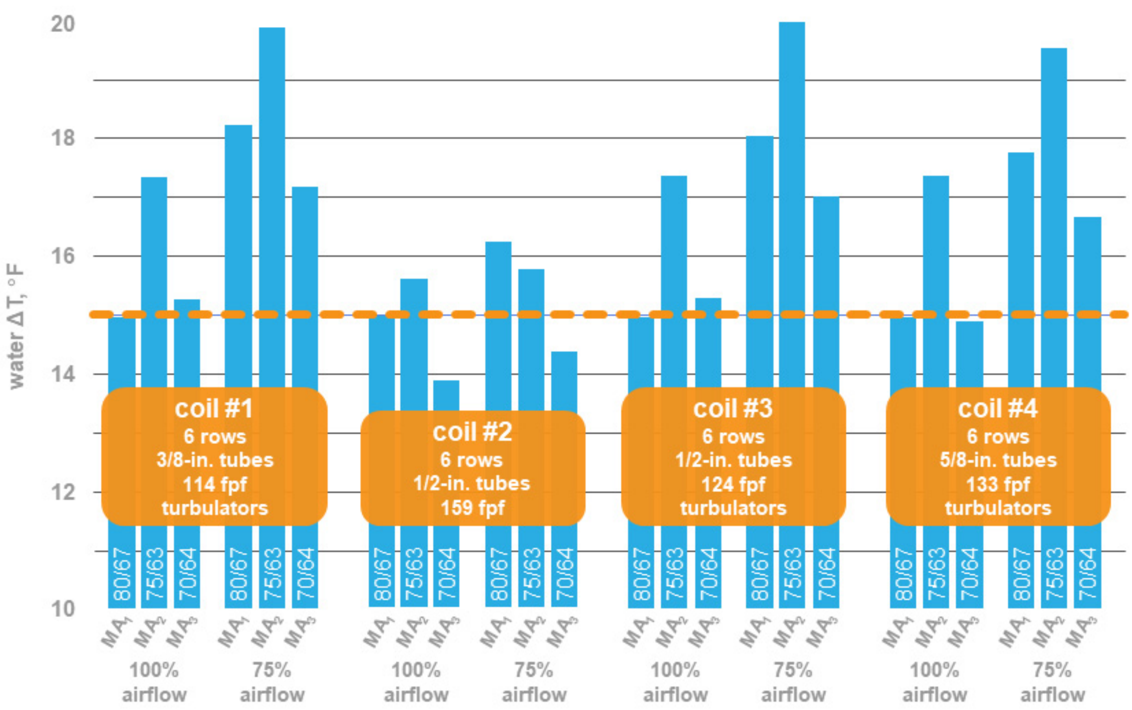
At part-load conditions, the coils with turbulators (#1, #3, and #4) are able to maintain, or even increase, the water ΔT. However, for the coil without turbulators (#2), the water ΔT starts to drop at the third part-load condition. And in all cases, the ΔT is not as high in coil #2 as it is in the coils with turbulators.

Table 1. Example chilled-water coil selections for 15°F ∆T

	coil #1	coil #2	coil #3	coil #4
entering-water temperature, °F	42	42	42	42
leaving-water temperature, °F	57	57	57	57
water ∆T, °F	15	15	15	15
tube diameter, in.	3/8	1/2	1/2	5/8
coil rows	6	6	6	6
fin density, fins/ft	114	159	124	133
fin design	high capacity	high capacity	high efficiency	high efficiency
turbulators	yes	no	yes	yes
water flow rate, gpm	40	40	40	40
water velocity, ft/s	2.7	2.8	2.8	2.1
water pressure drop, ft. H_2O	11.2	4.7	11.1	5.2
air pressure drop, in. H_2O	0.81	0.95	0.71	0.71
cost of coil	base - 30%	base	base + 8%	base + 15%

Coil selections from Trane Official Product Selection Software (TOPPS™), based on a size 14 Performance Climate Changer™ air handler with coils constructed of copper tubes and aluminum fins.

Figure 2. Water ∆T at part-load conditions



Entering-air conditions: MA_1 = 80°F dry bulb / 67°F wet bulb, MA_2 = 75°F dry bulb / 63°F wet bulb, MA_3 = 70°F dry bulb / 64°F wet bulb
Coil airflows: 100% airflow = 7000 cfm, 75% airflow = 5250 cfm

Turbulators increase fluid turbulence, which improves heat transfer (Figure 3). This allows a coil to provide the required capacity with a lower water flow rate (higher ΔT), leading to reduced pumping energy at part-load conditions. After analyzing many coil configurations, this is a consistent trend regarding turbulators and ΔT.

Impact of laminar flow. The ASHRAE Handbook suggests that chilled-water coils are best selected with water velocity between 2 to 4 ft/sec, at design conditions.[5] This recommended range is intended to provide a good balance between coil size and minimizing both air and water pressure drops.

But water velocity is also important because it is one of the key factors for determining flow turbulence, depicted by the Reynolds Number. As the turbulence of a moving fluid increases, so does its ability to transfer heat from the tube wall to the fluid.

Some in the HVAC industry express concern that coil heat transfer will deteriorate rapidly if the Reynolds Number falls into the laminar flow region. The performance prediction methodology prescribed by AHRI Standard 410 was refined in 2001, allowing coil performance to be accurately predicted well into the laminar flow region, without fear of large discrepancies between predicted and actual performance.[6]

To demonstrate, Figure 4 plots coil capacity versus water flow rate and Reynolds Number, using the current AHRI 410 performance prediction model. While this does show a slight dip in capacity that is less linear than the rest of operating range, it is certainly not drastic. In summary, laminar flow does not cause a severe drop-off in capacity. And the AHRI prediction methods allow coils to be rated accurately well into the transitional and laminar flow regions.

Warmer versus colder water temperatures. This new requirement in Standard 90.1 requires the leaving-water temperature be no colder than 57°F, allowing it to be warmer.

Table 2 compares coils selected for a 15°F ΔT, but with different entering-water temperatures. Coils #2 and #3 are from the previous example, with 42°F entering water. Coils #5 and #6 are selected with a 45°F entering-water temperature.

The coils selected with warmer water (#5 and #6) require eight rows of tubes to provide the necessary capacity. This results in much higher air and water pressure drops than the six-row coils selected with colder water (#2 and #3). And not only will the coil be more expensive due to these additional rows, the air-handling unit will likely need to be longer, which increases the cost of the casing as well.

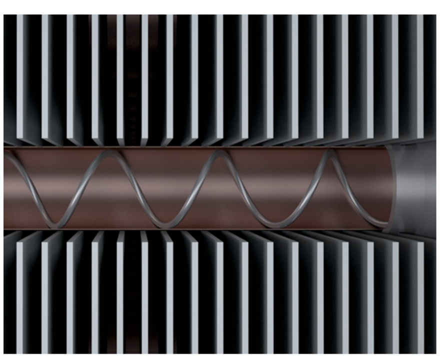
Even though the chiller has to work a little harder to make the 42°F water versus 45°F in this example, this is typically more efficient than making the fans and pumps both work harder to overcome these higher pressure drops.[3,7,8,9]

Water ΔT higher than 15°F. Note that Standard 90.1 requires the water ΔT to be 15°F or higher. There are many in the industry who recommend ΔTs even higher than this.[3,7,8,9]

In Table 3, the first column is one of the six-row coils from the previous example (coil #3), selected with 42°F entering water and a 15°F ΔT. The coil in the second column (coil #7) also uses 42°F entering water, but with a 20°F ΔT. And the coil in the third column (coil #8) is selected with 40°F entering water and a 25°F ΔT.

The larger water ΔTs reduce the water flow rate even further—from 40 gpm down to 30 gpm or 24 gpm—and also reduce the water pressure drop. This significantly lowers pump energy use. However, in this example, the higher ΔTs require more coil surface area, so air pressure drop does increase.

Figure 3. Turbulators



Trane has over 50 years of successful field experience with turbulators, with performance certified by AHRI.

Figure 4. Impact of laminar flow on coil performance

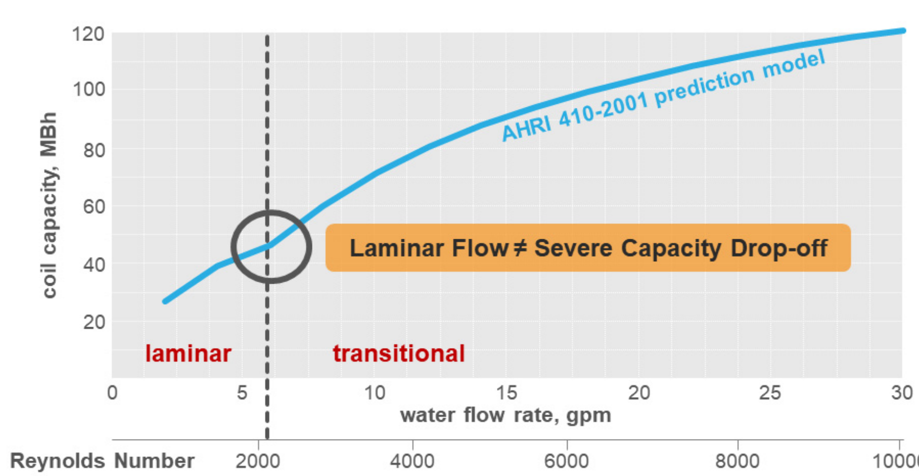


Table 2. Colder versus warmer water temperatures

	coil #2	coil #3	coil #5	coil #6
entering-water temperature, °F	**42**	**42**	**45**	**45**
leaving-water temperature, °F	57	57	60	60
water ΔT, °F	15	15	15	15
tube diameter, in.	1/2	1/2	1/2	1/2
coil rows	6	6	8	8
fin density, fins/ft	159	124	153	113
fin design	high capacity	high efficiency	high capacity	high capacity
turbulators	no	yes	no	yes
water flow rate, gpm	40	40	40	40
water velocity, ft/s	2.8	2.8	2.8	2.8
water pressure drop, ft. H_2O	4.7	11.1	5.8	15.1
air pressure drop, in. H_2O	0.95	0.71	1.2	1.0
cost of coil	base	base + 8%	base + 30%	base + 35%

Table 3. 15°F versus 20°F versus 25°F ΔT

	coil #3	coil #7	coil #8
entering-water temperature, °F	42	42	40
leaving-water temperature, °F	57	62	65
water ΔT, °F	**15**	**20**	**25**
tube diameter, in.	1/2	1/2	1/2
coil rows	6	8	8
fin density, fins/ft	124	114	135
fin design	high efficiency	high efficiency	high capacity
turbulators	yes	yes	yes
water flow rate, gpm	40	30	24
water velocity, ft/s	2.8	2.1	1.6
water pressure drop, ft. H_2O	11.1	8.4	5.8
air pressure drop, in. H_2O	0.71	0.88	0.92
cost of coil	base	base + 30%	base + 35%

ASHRAE 62.1 Limit on Air Pressure Drop

As mentioned, the Standard 90.1 committee stated in the foreword to this addendum that their intent was to encourage the use of coils with more heat transfer surface to achieve this higher ΔT. In some cases, this might result in a higher air pressure drop.

ASHRAE Standard 62.1 includes a requirement intended to ensure that coils can be properly cleaned.[10] Deeper coils with more rows, and coils with a higher density of fins, can be more challenging to clean.

The Standard 62.1 committee addressed this issue by prescribing a limit on coil air pressure drop, as a surrogate measure for clean-ability. In other words, coils with higher air pressure drops are, in general, more difficult to clean properly.

> **5.11.2 Finned-Tube Coil Selection for Cleaning.** Individual finned-tube coils or multiple finned-tube coils in series without intervening access spaces of at least 18 in. shall be selected to result in no more than 0.75 in. H_2O combined dry-coil pressure drop at 500 fpm face velocity.

This section of the standard requires that the air pressure drop of a finned-tube coil cannot exceed 0.75 in. H_2O. But notice that this is at a specific air velocity (500 fpm), and this limit is based on the air pressure drop when the coil is dry (not dehumidifying).

For the example in this EN, the entering-air conditions are 80°F dry bulb and 67°F wet bulb, which equates to a 60°F dew point. The air is being cooled to 53°F, which means that water vapor will be condensing out of the air and onto the coil surface. Therefore, the air pressure drops listed are for a wet coil, not dry.

To ensure that a selected coil complies with this requirement, use the manufacturer's selection program to re-run the performance of the coil, but change the entering-air conditions so that the coil will be dry, with no condensation. In this example, by lowering the entering wet bulb from 67°F to 55°F, the entering dew point drops to 30°F—well below the coil surface temperature, so the coil will operate dry.

The first column in Table 4 shows coil #2 from the previous example. The air pressure drop is 0.95 in. H_2O, but this is when the coil is wet. The second column shows the same coil, with the entering wet bulb changed to 55°F, so the coil will operate dry. (Note that the airflow was also changed slightly, so that the air velocity is exactly 500 fpm.) Under these dry conditions, at the prescribed air velocity, the air pressure drop is 0.70 in. H_2O, so this coil does comply with the Standard 62.1 limit on air pressure drop.

Table 4. Air pressure drop of a wet versus dry coil

	coil #2 (wet)	coil #2 (dry)
coil airflow, cfm	7000	6820
coil face velocity, fpm	513	**500**
entering dry-bulb temperature, °F	80	80
entering wet-bulb temperature, °F	67	**55**
entering dew point temperature, °F	60	**30**
leaving dry-bulb temperature, °F	53	53
tube diameter, in.	1/2	1/2
coil rows	6	6
fin density, fins/ft	159	159
fin design style	high capacity	high capacity
turbulators	no	no
air pressure drop, in. H_2O	0.95	0.70

Conclusion

ASHRAE Standard 90.1-2016 now requires chilled-water coils to be selected for a minimum 15°F ΔT, but there are exceptions. This higher ΔT reduces the water flow rate (gpm), which allows installation of smaller valves, pipes, and pumps, and also reduces pumping energy.

To achieve the 15°F minimum ΔT, some designers may choose to select the coil with more fins. This will increase the air pressure drop. Other designers may choose to select the coil with turbulators. This will increase the water pressure drop, but results in higher water ΔTs at part-load conditions, which leads to pump energy savings. And designing the system with a slightly lower entering-water temperature can allow the coils to be selected with little or no impact on air pressure drop and fan energy use.

Finally, selecting cooling coils for a higher ΔT has an impact on the design and operation of the chiller plant. These issues are addressed in other *Engineers Newsletters*.[8,11,12]

By John Murphy, Trane. To subscribe or view previous issues of the *Engineers Newsletter* visit trane.com/EN. Send comments to ENL@trane.com.

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